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CENTRAL RECEIVER SOLAR THERMAL POWER SYSTEM
PHASE 1. CDRL ITEM 2

Pilot Plant Preliminary Design Report
Volume 5. Thermal Storage Subsystem

By
Raymon W. Hallet, Jr.
Robert L. Gervais

Date Published—November 1977

Work Performed Under Contract No. EY-76-C-03-1108

McDonnell Douglas Astronautics Company
Huntington Beach, California

U.S. Department of Energy

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VOLUME V
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PREFACE

This report is submitted by the McDonnell Douglas Astronautics Company to the Department of Energy under Contract EY-76-C-03-1108 as the final documentation of CDRL Item 2. This Preliminary Design Report summarizes the analyses, design, test, production, planning, and cost efforts performed between 1 July 1975 and 1 May 1977. The report is submitted in seven volumes, as follows:

Volume I, Executive Overview

Volume II, System Description and System Analysis

Volume III, Book 1, Collector Subsystem

Book 2, Collector Subsystem

Volume IV, Receiver Subsystem

Volume V, Thermal Storage Subsystem

Volume VI, Electrical Power Generation/Master Control Subsystems and Balance of Plant

Volume VII, Book 1, Pilot Plant Cost and Commercial Plant Cost and Performance

Book 2, Pilot Plant Cost and Commercial Plant Cost and Performance

Specific efforts performed by the members of the MDAC team were as follows:

- McDonnell Douglas Astronautics Company
  Commercial System Summary
  System Integration
  Collector Subsystem Analysis and Design
  Thermal Storage Subsystem Integration

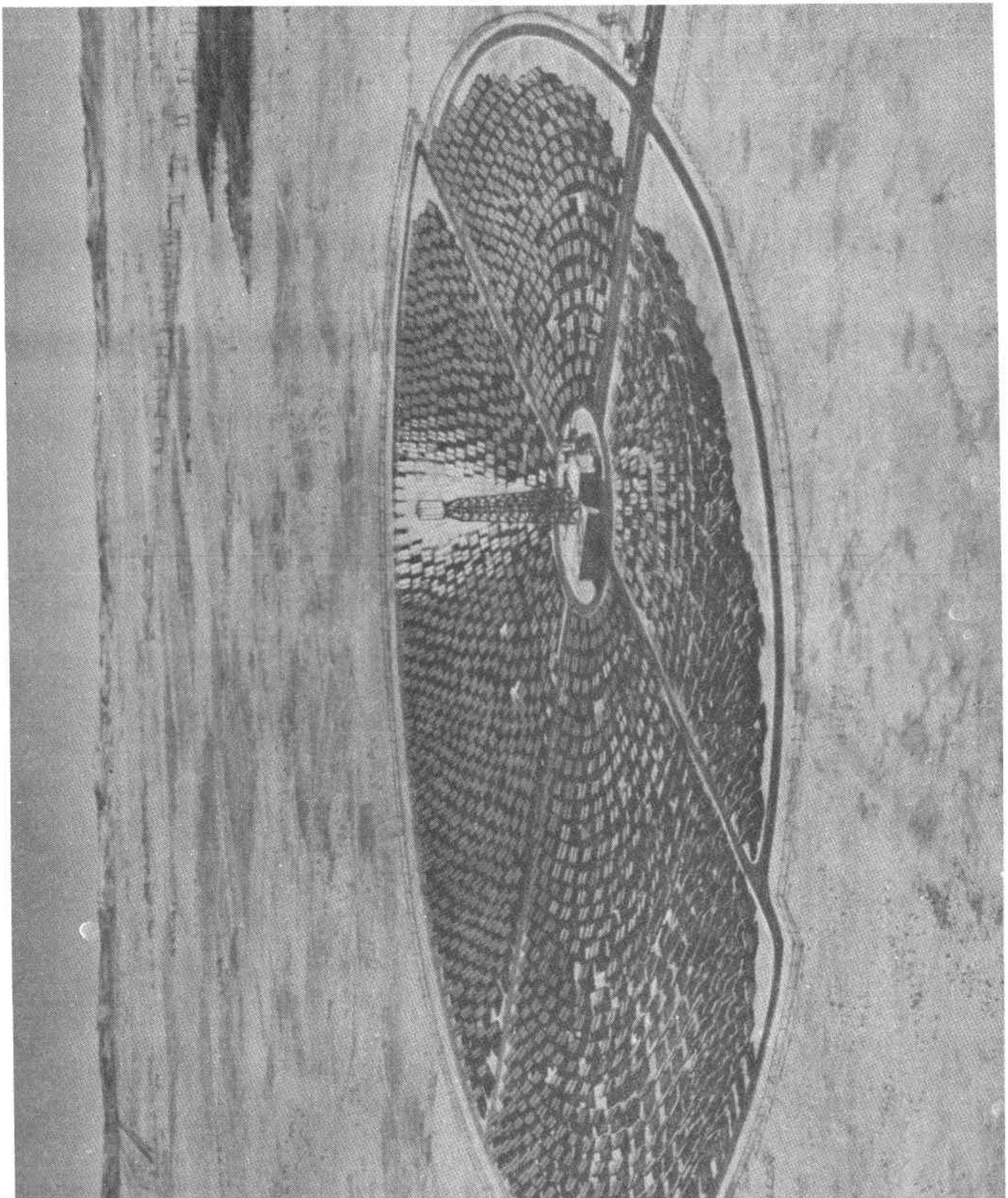
- Rocketdyne Division of Rockwell International
  Receiver Assembly Analysis and Design
  Thermal Storage Unit Analysis and Design

- Stearns-Roger, Inc.
  Tower and Riser/Downcomer Analysis and Design
  Electrical Power Generation Subsystem Analysis and Design

- University of Houston
  Collector Field Optimization

- Sheldahl, Inc.
  Heliostat Reflective Surface Development

- West Associates
  Utility Consultation on Pilot Plant and Commercial System Concepts



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Section 1
INTRODUCTION AND SUMMARY

1.1 THERMAL STORAGE CONCEPT RATIONALE

1.1.1 Power Plant Needs for Thermal Storage

Solar thermal power plants will be a significant future element of the world's energy supply system. A thermal storage subsystem (TSS) will have a vital role in the effective operation to each of those solar power plants. As a buffer between the solar portion of the plant and the electrical generating portion, the TSS protects the turbine from rapid variations in inlet conditions caused by rapid and short term changes in insolation caused by clouds passing over the collector field. In addition, the TSS extends the solar plant's generating capacity into periods with little or no insolation. Providing generating capacity for this period allows the solar plant to displace additional fossil fueled capacity.

Another significant function of the TSS is to assist in the matching of annual fluctuations in insolation to the sizing of the electrical generating portions of the plant. If the electrical subsystem is sized for the peak summer noon insolation, its full capacity is unused for most of the year. If it is sized much smaller, a great deal of solar energy is lost unless it can be stored. Trade studies show that 6 hr of storage provide approximately the optimum capacity in a Commercial Plant in the southwestern United States.

Energy storage concepts that first require electric generation, e.g., pumped water, compressed air, flywheel or battery storage, do not fulfill the basic requirement of buffering the turbine from solar insolation variations. Thermal storage has an additional advantage by directly providing efficient turbine seal heating during periods of turbine inactivity. Therefore, thermal energy storage is preferable for this solar energy application.

1.1.2 Thermal Storage Alternatives

Thermal storage concepts can be classified into three categories: sensible-heat, latent-heat (phase change), and thermochemical (reversible chemical reactions). Of these, sensible-heat systems are clearly within the current state-of-the-art and present the least technical risk and thus the least cost and development risk. Both latent-heat and thermochemical storage systems will require considerable additional development before they can be evaluated in detail and considered for implementation on a large scale.

The primary focus in latent-heat systems has been on liquid-solid phase changes, making use of the heat of fusion. Three of the major problem areas in such systems are: (1) designing to accommodate the volume change upon phase transition, typically of the order of 20%, (2) ensuring that the salt system selected will maintain a clean transition between solid and liquid without long-term changes in the salt structure and composition, and (3) preventing solid buildup on heat-transfer surfaces, which rapidly drops the heat-transfer rates. The third of these problems has been particularly difficult to solve in a fashion practical for large-scale, inexpensive, and reliable 30-yr operation.

Considerable research effort is being expended in thermochemical storage concepts. Some of the major problem areas to be solved are: (1) all stored reactants and reaction products must be in condensed phases (no gases) to provide reasonable storage densities, (2) product separation usually is difficult unless one of the products is a gas, (3) high reaction entropy and enthalpy are difficult to achieve in reactions that do not produce a gaseous product, (4) reaction rates must be sufficiently rapid under practical conditions, (5) heat-transfer rates through solid products (if present) are low, just as for phase change storage, and (6) there must not be changes in the materials and cycle during the required 30-yr life (about 10,000 cycles). Chemical reaction storage concepts may be attractive in the future, but are not yet sufficiently developed for application in this program.

As a result of many studies and for the reasons summarized very briefly above, sensible-heat thermal storage was selected for this application.

1.1.3 Sensible-Heat Thermal Storage Choices

The solar thermal power application of interest required the use of steam as the working fluid in the receiver. It would be uneconomical to store the steam produced by the receiver directly in the quantities required for thermal storage, or to allow it to directly heat another storage medium, such as a rock bed, because of the very large, high-pressure tank which would be required. Therefore, some other fluid is required to accept energy from the steam, and later to supply energy to produce steam for extended operation from storage. A variety of organic and inorganic liquids were considered. The large volumes of thermal storage fluid required in a Commercial Plant make fluid cost a driving parameter. The media cost for an all-liquid thermal storage subsystem is a substantial fraction of the cost of the entire subsystem.

The conventional pebble-bed heater, using air or another gas to heat a rock bed which is often not stationary, was unattractive for this solar power plant application because of excessive blower power, heat exchanger sizes, temperature degradation, and low thermal efficiency. The thermal storage concept finally devised and selected uses a low-cost stationary solid bed to store most of the energy, with a suitable liquid to transfer energy into and out of the bed (and to store part of the energy directly). This dual-medium type of system (patent pending to Rocketdyne and MDAC) combines advantages of a low-cost solid with the flexibility, low pumping power, and moderate heat-exchanger requirements of a liquid energy storage system.

Conceptually, in its simplest form, the system uses a bed (shown in the center of Figure 1-1) of an inexpensive solid (e.g., rock, ore, metal scraps, etc). An appropriate high-temperature liquid fills the voids in the bed and circulates through the bed to deposit or withdraw energy.

In the cyclical operation, heating of the bed (charging) is achieved by removing lower temperature fluid from the bottom of the bed, heating it in a heat exchanger with steam from the receiver, and returning the fluid to the top of the tank. A fairly sharp, temperature transition (a thermocline) is maintained naturally between hot and cold fluid because of the lower density of hot fluid. This thermocline moves downward through the bed during charging and upward during extraction. When the storage unit is completely

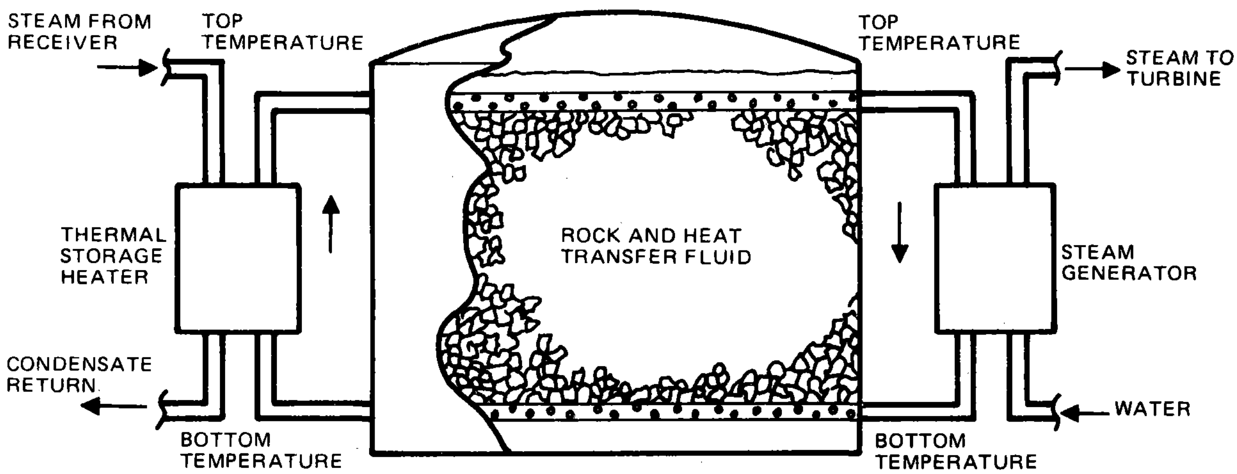

Figure 1-1. Dual-Medium Thermal Storage Concept

charged, all of the bed and the fluid are at the maximum temperature and the thermocline does not exist. The extraction loop uses the fluid to remove energy from the storage unit and produces steam for power plant operation or other plant functions such as equipment heating.

The large cost savings for this type of thermal storage results principally from two factors: (1) replacement of about 75% of the expensive storage liquid with inexpensive rock, and (2) use of the thermocline principle to reduce the tankage volume substantially, compared to a system with separate tanks for hot and cold storage.

There are many variations and improvements on the basic concept outlined above. These include: (1) choice of solid material (e.g., various types of rocks, ore, metal scraps, blocks, bricks, ceramics, etc), (2) size distributions and consequent void fractions of solid bed, (3) method of bed placement, (4) choice of liquid (e.g., water, various petroleum products, heat-transfer fluids, molten salts, liquid metals), (5) single or multiple thermal storage unit (TSU) tanks, (6) combinations of various liquids and solids in series tanks to achieve maximum high-temperature performance at minimum cost, and (7) use of immiscible liquids. These types of variations and design choices were considered at various stages of the design and development work on the thermal storage subsystem.

One of the important decisions which required considerable analysis was selection of the maximum storage temperature, which fixes, within a small range, the maximum temperature of steam produced from thermal storage.

Commercially available organic heat-transfer fluids which are liquid at ambient temperature have maximum operating temperatures of about 316°C (600°F). Storage temperatures above this region require alternatives which introduce some negative features. The principal alternatives are: (1) organic fluids which are solid at ambient temperatures (with higher costs and the problems of initial startup and avoiding solidification during operation), (2) fluids with higher vapor pressures, e.g., Dowtherm A (with higher fluid costs and much higher storage container costs), (3) inorganic salts and salt mixtures (with problems of initial startup and avoiding solidification during operations, plus usually higher medium costs).

The improvements in plant efficiency and energy output which can be obtained with higher storage temperatures must be enough to justify the greater costs necessary to provide the higher storage temperature. Design analyses showed that the baseline system we used for this program (i.e., dual-medium with low-cost organic fluid and rock and storage at about 316°C [600°F]) provided lower cost energy than any of the higher temperature alternatives. The only exception to this conclusion was the addition of a higher temperature dual-medium stage in addition to the baseline dual-medium stage; however, this option would require some additional technology development and demonstration.

1.2 VOLUME V OVERVIEW

This section parallels the body of the report (Sections 2 through 6) and summarizes key information and results from each part of the TSS work accomplished during this program. The thermal storage concept described in Section 1.1 is the concept used in all facets of the program, e.g., preliminary designs and plans for Commercial and Pilot Plant subsystems, and SRE tests.

Section 2 of this report volume contains tabular summaries of design characteristics for the three TSS categories: Pilot Plant, Commercial Plant, and SRE, arranged for convenient cross-reference between the three subsystems. Sections 3 through 6 of the report present details for each of the subsystems, as summarized briefly in the following subsections.

1.2.1 Summary of Commercial TSS Definition

The eventual result of the effort during this program will be successful commercialization of the solar power plant. Although Commercial Plants will be built with various capacities, the first Commercial Plant is envisioned as a 100-MWe plant. The Department of Energy 10-MWe Pilot Plant is intended to provide a firm basis for design and successful operation of the first Commercial Plant. Consequently, a preliminary design of a Commercial Plant TSS was made to establish requirements for the Pilot Plant design.

1.2.1.1 Subsystem Requirements

The TSS buffers the electrical power generating subsystem (EPGS) from excessive variations in insolation, and extends the plant's capacity into periods with low or no insolation.

The general requirement on the TSS is that it provide a means of transferring to stored thermal energy a portion of the thermal output from the receiver subsystem and, subsequently, transferring stored thermal energy to steam in a form suitable for generating electrical power with the conventional turbine-generator in the EPGS.

The TSS is required to have an extractable storage capacity of at least 1857 MWHt, which is comprised of 100 MWHt to provide a turbine hot start and 1757 MWht to permit the turbine-generator to supply 70 MWe net (76 MWe gross) for 6 hr following turbine start-up. The required charging rates are 12.5 MWt (rated steam operation) to 255 MWt (derated steam operation). Required discharging rates are 31.4 to 285 MWt. The maximum allowable heat loss is 2% of extractable capacity in 24 hr, starting in a fully charged condition. The subsystem is required also to provide nighttime seal steam at a temperature of at least 135°C (275°F) and at a rate of 0.42 MWt for approximately 12 hr.

There are five fluid streams crossing the boundaries of the TSS, all water or steam flows. A major requirement is to accept steam from the receiver at 360/510°C (680/950°F) and 10.1 MPa (1465 psia), where the two sets of temperatures correspond to derated and rated steam operation, respectively. Another major requirement is to supply steam from the TSS steam generator for the turbine at 299°C (570°F) and 2.72 MPa (395 psia).

Additional details of requirements on performance, interfaces, environmental, structural, and safety aspects are given in Section 3.1.

The Commercial Plant TSS 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, extractable energy.

A large number of design analyses and trade studies were made in the process of developing the preliminary design for the TSS. These analyses are described in Section 3.2.

Figure 1-2 is a schematic diagram of the TSS, showing all major components, lines, and major control concepts. Process flow conditions are shown at

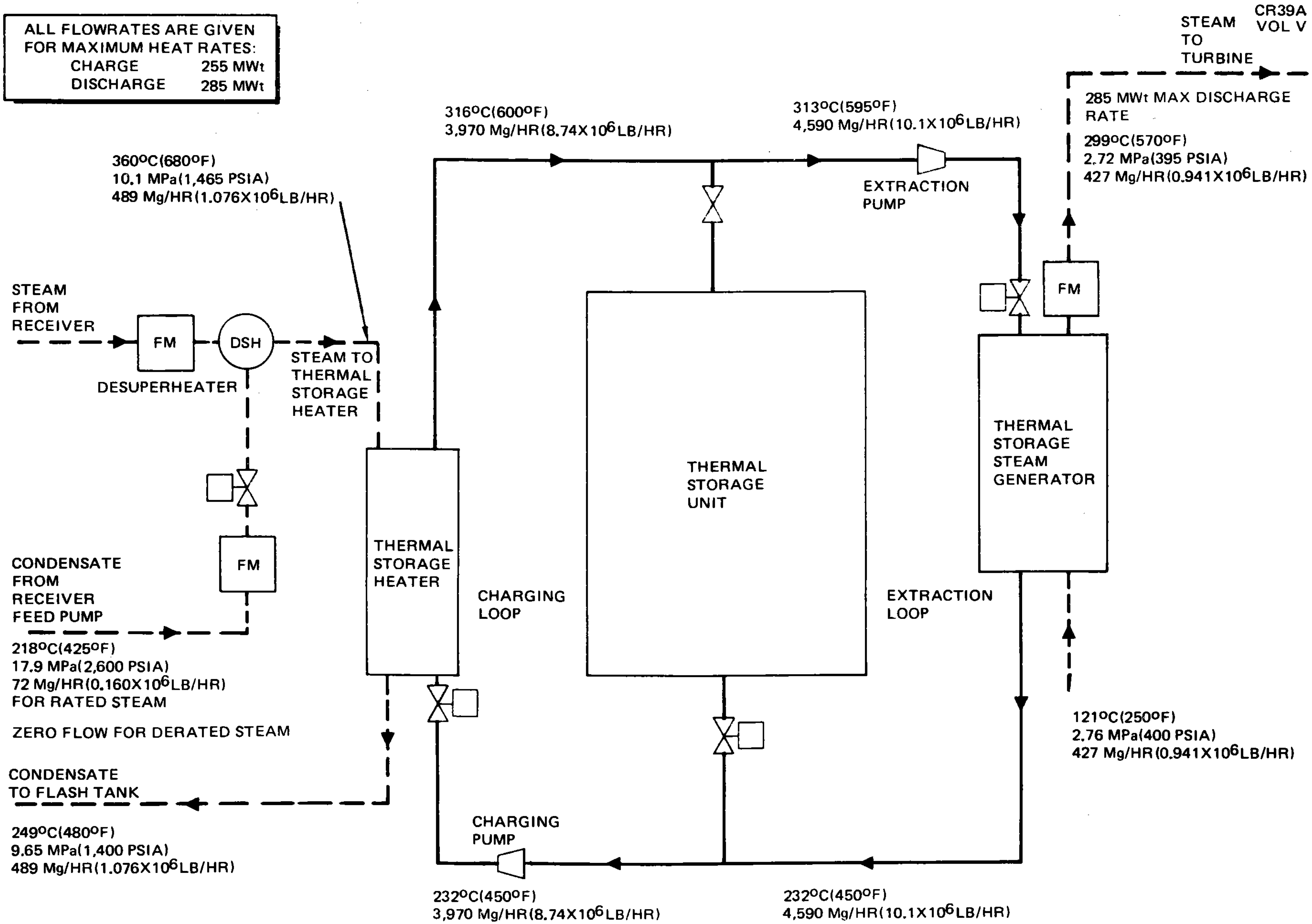

Figure 1-2. Schematic Flow Diagram of Thermal Storage Subsystem for 100-MWe Commercial Plant

various points in the subsystem. Table 1-1 summarizes the principle characteristics of the subsystem and major components. As shown in Figure 1-2, the subsystem can be considered in three major parts: (1) the central TSU's, (2) the thermal charging loop, and (3) the heat-extraction loop. In the charging loop, energy is removed from the receiver steam and stored in the thermal storage tanks. A commercial heat-transfer fluid, Caloria HT43, is used to permit economical ambient pressure storage. The extraction loop uses the fluid to remove energy from the storage units and produce steam for either turbine operation or equipment heating. Additional details are given in Section 3.2.

1.2.1.3 Subsystem Fabrication/Installation

Almost all of the components in the TSS are conventional, commercial items. Their fabrication and installation introduce no new requirements different from standard practices in the chemical processing and petroleum industries. A summary Commercial Plant TSS development schedule is given in Figure 1-3.

The thermal storage tanks are conventional, using standard carbon steel structural materials. The tank fabrication and installation are combined, following standard practice, by field welding various plate sections previously rolled to the proper radius of curvature. The charging heat exchangers, steam generator units, pumps, piping, valves, controls, and instrumentation are all commercial items. Their fabrication will follow standard vendor practices, and their installation will be standard.

The ullage maintenance unit (UMU) and fluid maintenance unit are comprised of standard commercial components: tanks, valves, piping, compressors, filters, and a heat-transfer-fluid distillation unit. Although the combinations of components are unique to perform the particular functions of these two subsystem elements, the fabrication and installation of the individual components introduces nothing new to the construction industry.

Table 1-1 (Page 1 of 2)

COMMERCIAL PLANT THERMAL STORAGE SUBSYSTEM DESCRIPTION

Assembly	Description
Thermal storage unit	Four identical units; each a cylindrical tank, axis vertical, installed above ground, 27.6m (90.5 ft) diameter by 18.3m (60.0 ft) high; 10,900m^3 (386,000 ft^3, 2,890,000 gal) volume; each containing 20.3 x 10^6 kg (22,300 ton) of granite rock and coarse silica sand (approximately 2:1 rock:sand by volume) and 2.2 x 10^6 liters (583,000 gal) of Caloria HT43 heat-transfer fluid. Fluid temperature range: 232 to 316°C (450 to 600°F). Fabricated of ASTM A537 Class 2 structural steel by field-welded construction.
Ullage maintenance unit	Storage and control of ullage gas with compressed gas storage at 1.20 MPa (175 psia); tank pressure control, venting, inert gas (nitrogen) control, volatile vapor recovery and control.
Fluid maintenance unit	Full-flow, continuous filtration with dual 80-mesh filters in main fluid line upstream of pump; periodic distillation with vacuum distillation unit in side-stream to remove polymerized materials; periodic fluid makeup.
Desuperheater	Direct-contact mixing chamber with water injected through multiple atomizing nozzles into superheated steam; single unit; ten nozzles.
Thermal storage heater	Five identical heat exchangers in parallel; each is TEMA type DFU, with removable U-tube bundle, 2 shell passes, 6 tube passes; steam/water on tube side; 1672m^2 (18,000 ft^2) heat-transfer area per heat exchanger; carbon steel.
Steam generator	Five modules in parallel. Each module consists of three separate stages in series consisting of feedwater preheater, boiler, and superheater; steam/water on shell side; carbon steel. Preheater is straight tube, floating head, counterflow exchanger with 435m^2 (4,684 ft^2) heat-transfer area per exchanger. Boiler is horizontal U-tube kettle boiler with 1204m^2 (12,948 ft^2) heat-transfer area per exchanger. Superheater is horizontal U-tube, crossflow exchanger with 594m^2 (6,389 ft^2) heat-transfer area per exchanger.

Table 1-1 (Page 2 of 2)

COMMERCIAL PLANT THERMAL STORAGE SUBSYSTEM DESCRIPTION

Assembly	Description
Fluid charging loop pump	Five identical pumps in parallel; centrifugal, high-temperature type, with single-speed electric motors; each pump has flow of 220 kg/s (490 lb/s); and 0.19 MWe (250 hp) motor input at maximum charging rate (51 MWt).
Fluid extraction loop pump	Five identical pumps in parallel; centrifugal, high-temperature type, with single-speed electric motors; each pump has flow of 260 kg/s (580 lb/s), 0.19 MWe (260 hp)* motor input at maximum extraction rate (57 MWt).

*Required input motor power; not full motor capacity

1.2.1.4 Commercial Subsystem Operational Characteristics

The TSS was carefully designed to permit safe, flexible operation in a variety of modes, with easy transition between operating modes. Providing such flexibility was particularly important for this subsystem since one of its major functions is to accommodate variations and problems introduced by the weather and/or other subsystems, while protecting the turbine from excessive variations in inlet steam conditions and flow rates.

The major modes are: startup (both cold and hot), charging, discharging, intermittent cloud operation, emergency, shutdown, and standby (including equipment heating elsewhere in the plant). These modes and transitions are discussed in detail in Section 3.4.

1.2.2 Summary of Pilot Plant TSS Definition

Since the Pilot Plant objective is to prove the validity of the Commercial Plant design, the same basic design concept was used for both Commercial and Pilot Plant subsystems. The approximately tenfold difference in capacity led to designs making use of modular components, with fewer numbers of components in the Pilot Plant, often of similar size to the more numerous modules in the Commercial Plant subsystem.

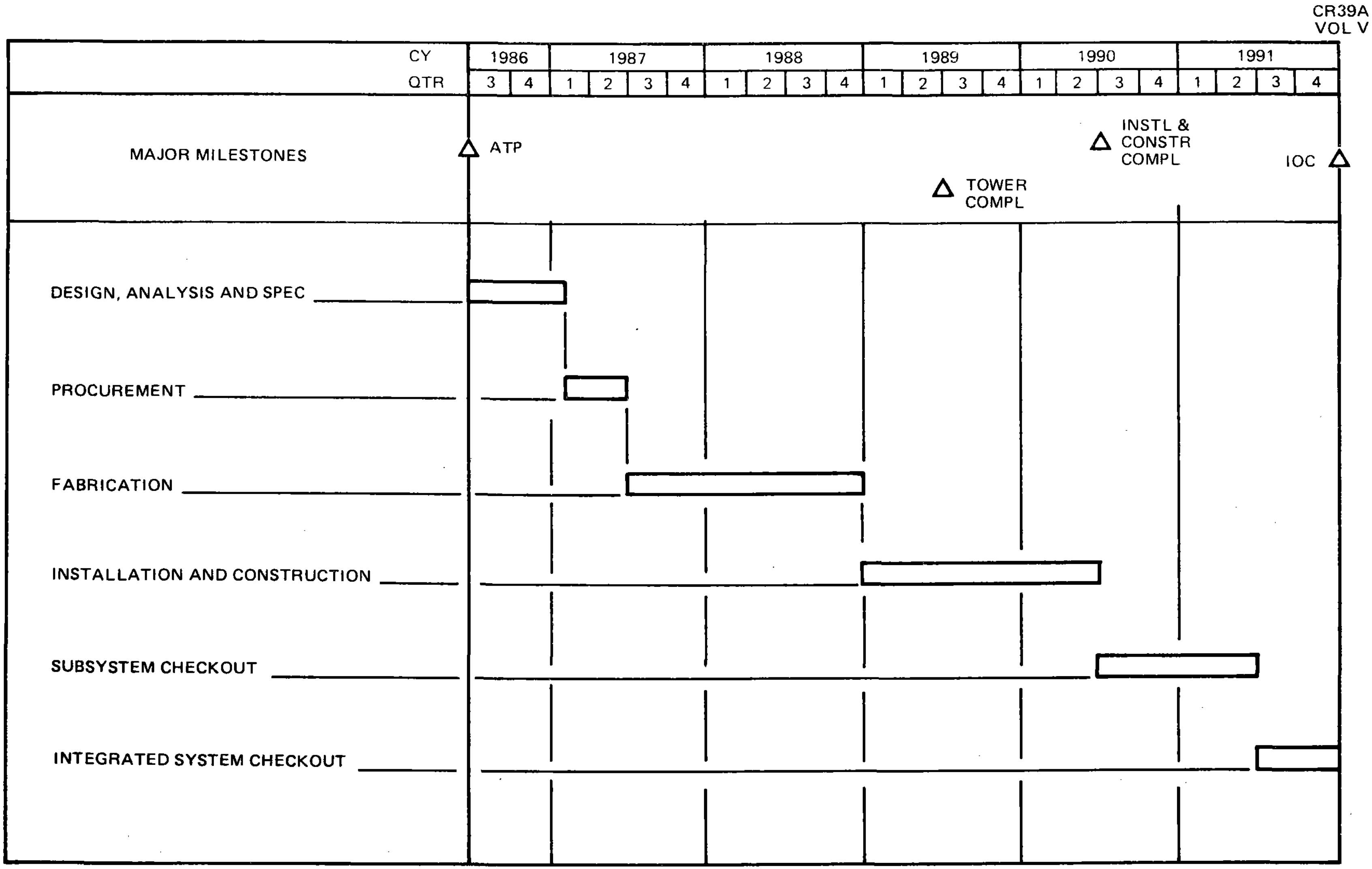

Figure 1-3. Thermal Storage Subsystem – First Commercial Plant

1.2.2.1 Subsystem Requirements

Then Pilot Plant TSS has the same general requirements as in a Commercial Plant; i.e., to buffer the electrical power generating subsystem from excessive variations in insolation, and ext end the pilot plant's generating capacity into periods with low or no insolation.

The TSS is required to have an extractable storage capacity of at least 103.8 MWHt, which is comprised of 7.5 MWHt to provide a turbine hot start and 96.3 MWHt to permit the turbine-generator to supply 7 MWe net (7.8 MWe gross) for 3 hr following turbine startup. The required charging rates are 1.5 MWt to 30 MWt. Required discharge rates are 3.1 MWt to 32.1 MWt. The maximum allowable heat loss is 3% of extractable capacity in 24 hr, starting in a fully charged condition. The subsystem is also required to provide nighttime seal steam at a temperature of at least 135°C (275°F) and at a rate of 0.33 MWt for approximately 16 hr.

A major requirement is to accept steam from the Pilot Plant receiver at 343/510°C (650/950°F) and 10.1 MPa (1465 psia), where the two sets of temperatures correspond to derated and rated steam operation, respectively. Another major requirement is to supply steam from the TSS steam generator for the turbine at 277°C (530°F) and 2.76 MPa (400 psia). Additional details of requirements on performance, interfaces, environmental, structural and safety aspects are given in Section 4.2.

1.2.2.2 Pilot Plant Design Summary

The 10-MWe Pilot Plant TSS employs a sensible-heat concept using dual liquid and solid media for heat storage in a single tank, with the thermocline principle applied to provide high-temperature, extractable energy.

A large number of design analyses and trade studies were made in the process of developing the preliminary design for the TSS. These analyses are described in Section 4.2.

Figure 1-3 is a schematic diagram of the TSS, showing all major components, lines, and major control concepts. Process flow conditions are shown at various points in the subsystem. Table 1-2 summarizes the principal characteristics of the subsystem and major components. As shown in Figure 1-4, the subsystem can be considered in three major parts: (1) the central TSU, (2) the thermal-charging loop, and (3) the heat-extraction loop. In the charging loop, energy is removed from the receiver steam and stored in the TSU tank. A commercial heat-transfer fluid, Caloria HT43, is used to permit economical ambient pressure storage in the tank. The extraction loop uses the fluid to remove energy from the storage unit and produce steam for either turbine operation or equipment heating. Additional details are given in Section 4.2.

1.2.2.3 Pilot Plant Subsystem Operational Characteristics

The TSS was carefully designed to permit safe, flexible operation in a variety of modes, with easy transition between operating modes. Providing such flexibility was particularly important for this subsystem since one of its major functions is to accommodate variations and problems introduced by the weather and/or other subsystems, while protecting the turbine from excessive variations in inlet steam conditions and flow rates.

The major modes are: startup (both cold and hot), charging, discharging, intermittent cloud operation, emergency, shutdown, and standby (including equipment heating elsewhere in the plant). These modes and transitions are discussed in detail in Section 4.4.

1.3 PILOT PLANT SUBSYSTEM PLANS

Figure 1-5 is a summary schedule for the subsystem detail design, production, installation, and checkout. Details of the plans are given in Section 5.

Almost all of the components in the TSS are conventional, commercial items. Their fabrication and installation introduce no new requirements different from standard practices in the chemical processing and petroleum industries.

The thermal storage tank is conventional, made with standard carbon steel structural materials. The tank fabrication and installation are combined, following standard practice, by field welding various plate sections previously

Table 1-2 (Page 1 of 2)

PILOT PLANT THERMAL STORAGE SUBSYSTEM DESCRIPTION

Assembly	Description
Thermal storage unit	Single cylindrical tank, axis vertical, installed above ground, 15.2m (50.0 ft) diameter by 13.4m (44.0 ft) high; 2450m^3 (86,400 ft^3, 646,000 gal) volume; contains 4.53 x 10^6 kg (4990 ton) of granite rock and coarse silica sand (approximately 2:1 rock:sand by volume) and 525,000 liters (139,000 gal) of Caloria HT43 heat-transfer fluid. Fluid temperature range: 218 to 302°C (425 to 575°F). Fabricated of ASTM A537 Class 2 structural steel by field-welded construction.
Ullage maintenance unit	Storage and control of ullage gas with compressed gas storage at 1.20 MPa (175 psia); tank pressure control, venting, inert gas (nitrogen) control, volatile vapor recovery and control.
Fluid maintenance unit	Full-flow, continuous filtration with dual 80-mesh filters upstream of pumps; periodic distillation with vacuum distillation unit in side-stream to remove polymerized materials; periodic fluid makeup.
Desuperheater	Direct contact mixing chamber with water injected through multiple atomizing nozzles into superheated steam; single unit; three nozzles.
Thermal storage heater	Two identical heat exchangers in parallel; each is TEMA type DFU, with removable U-tube bundle, 2 shell passes, 6 tube passes; steam/water on tube side; 464m^2 (5000 ft^2) heat-transfer area per heat exchanger; carbon steel.
Steam generator	Three-stage (series) modules each with separate feed-water preheater, boiler, and superheater; 2 modules in parallel; steam/water on shell side; carbon steel: Preheater is straight tube, floating head, counterflow heat exchanger with 196m^2 (2, 106 ft^2) heat-transfer area per exchanger. Boiler is horizontal U-tube kettle boiler with 791m^2 (8,513 ft^2) heat-transfer area per exchanger. Superheater is horizontal U-tube, crossflow heat exchanger with 84m^2 (904 ft^2) heat-transfer area per heat exchanger
Fluid charging loop pump	Two identical pumps in parallel; centrifugal, high-temperature type, with dual-speed electric motors; each pump has flow of 64 kg/s (141 lb/s), and 0.060 MWe (80 hp)* motor input at maximum charging rate (15 MWt).

*Required input power; not full motor capacity

Table 1-2 (Page 2 of 2)
PILOT PLANT THERMAL STORAGE SUBSYSTEM DESCRIPTION

Assembly	Description
Fluid extraction loop pump	Two identical pumps in parallel; centrifugal, high-temperature type, with single-speed electric motors; each pump has flow of 70 kg/s (155 lb/s), and 0.052 MWe (70 hp)* motor input at maximum extraction rate (16.1 MWt).

*Required input power; not full motor capacity

rolled to the proper radius of curvature. The charging heat exchangers, steam generator units, pumps, piping, valves, controls, and instrumentation are all commercial items. Their fabrication will follow standard vendor practices, and their installation will be standard.

The ullage maintenance and fluid maintenance units are comprised of standard commercial components; tanks, valves, piping, compressors, filters, and a heat-transfer-fluid distillation unit. Although the combinations of components are unique to perform the particular functions of these two subsystem elements, the fabrication and installation of the individual components introduces nothing new to the construction industry.

1.3.1 Subsystem Research Experiments (SRE)

The SRE consisted of tests to provide data needed to establish a firm engineering basis for design of the Pilot and Commercial Plants. The major test objectives were:

- Evaluate heat-transfer-fluid thermal stability, compatibility, and surface fouling characteristics.
- Evaluate charging and extraction capabilities of a scalable TSU.
- Obtain performance of the TSU over all ranges of equivalent operating conditions in the Pilot Plant.
- Demonstrate stable operation for high, low, intermittent, and no insolation conditions.
- Evaluate the heat-transfer fluid under operational conditions.
- Determine the strain from rock/tank wall interactions.

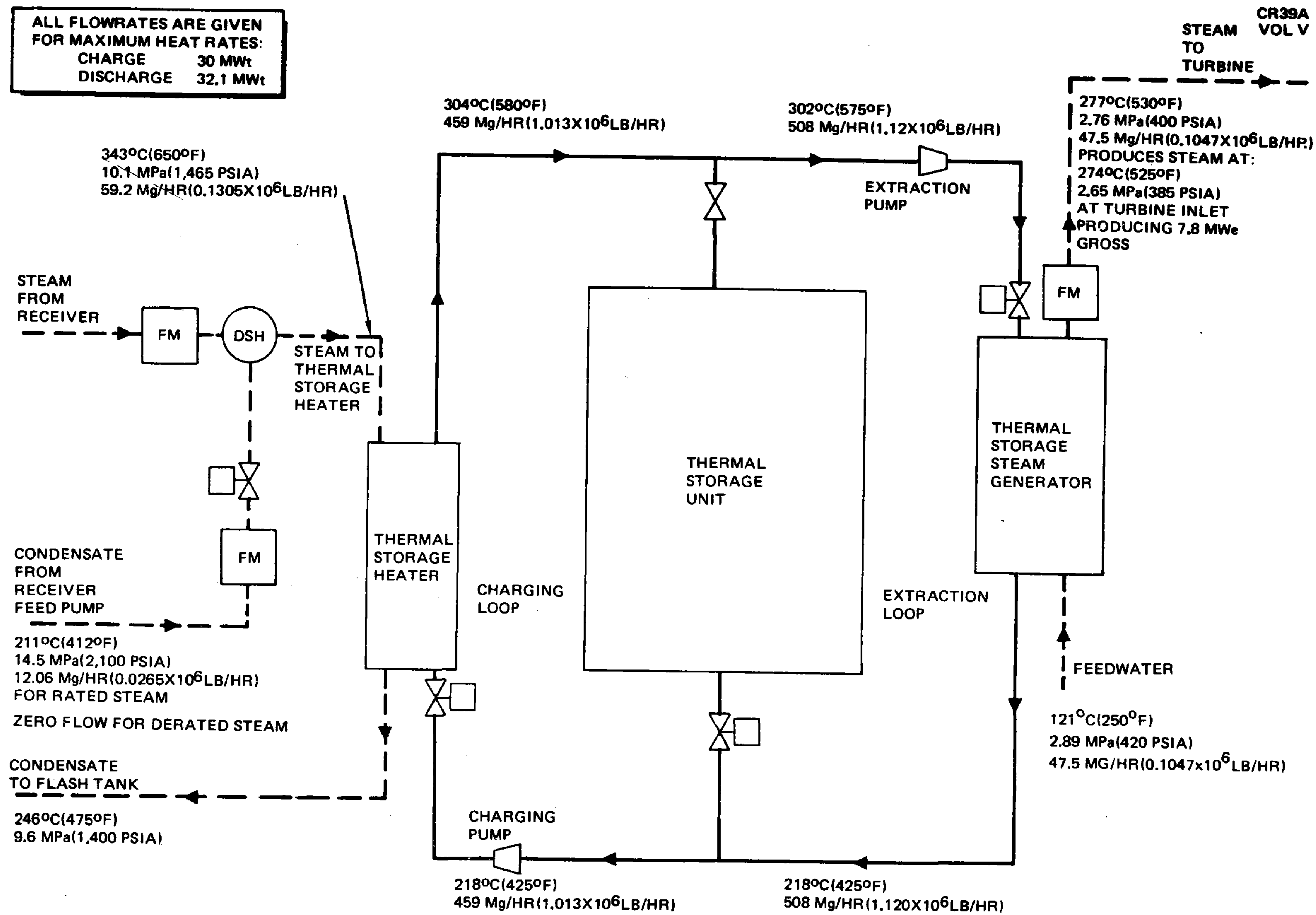

Figure 1-4. Schematic Flow Diagram of Thermal Storage Subsystem for 10-MWe Pilot Plant

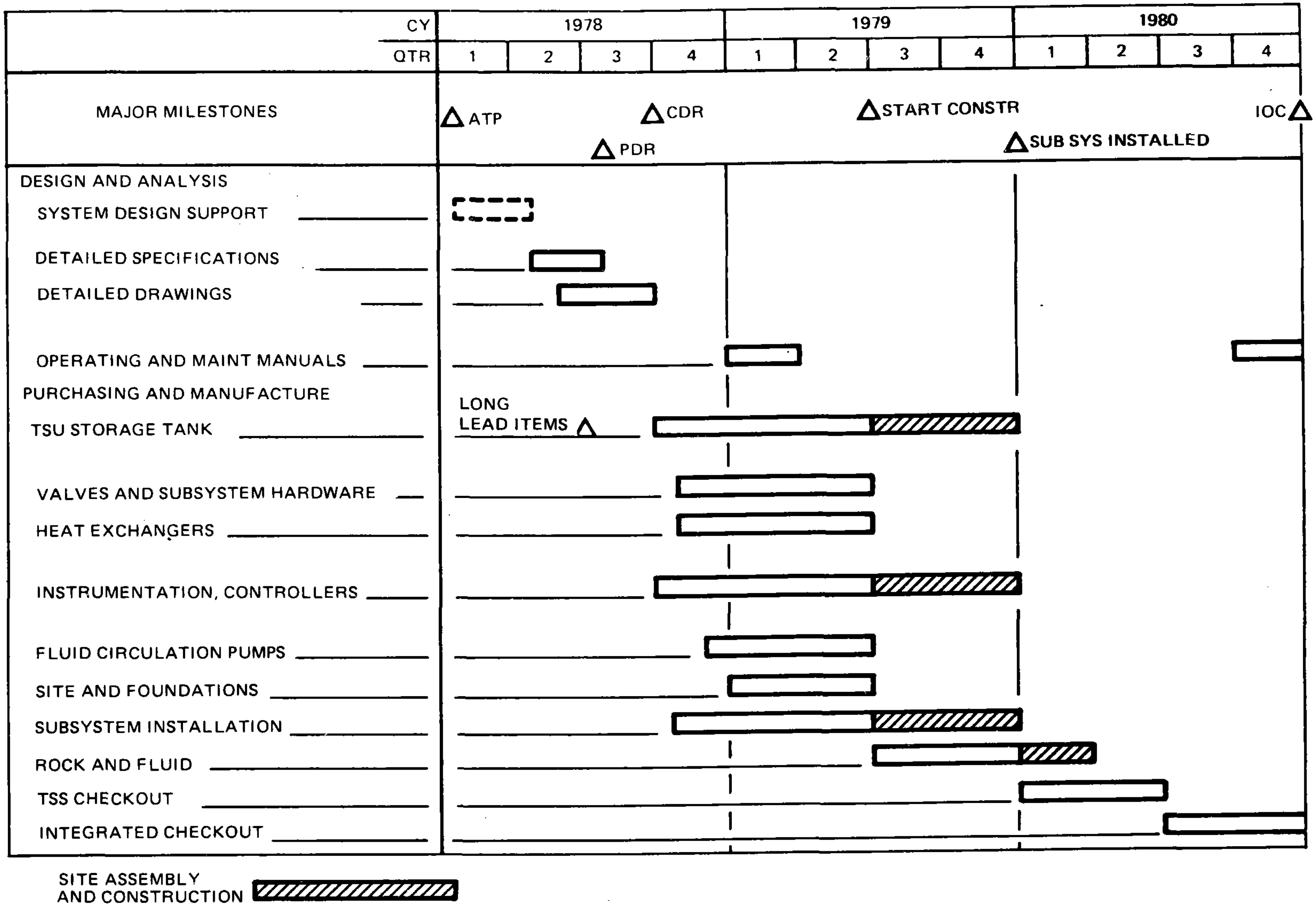

Figure 1-5. Thermal Storage Subsystem, Pilot Plant — Design, Fabrication, and Checkout Summary Schedule

All tests were completed successfully, and all test objectives were met.

Characteristics of candidate heat-transfer fluids were evaluated in a series of fluid prequalification and life tests conducted on three commercial fluids: Exxon Caloria HT43, Monsanto Therminol 55, and Monsanto Therminol 66. These tests further supported the choice of Caloria HT43, and demonstrated that it can fulfill the requirements of this solar power application. This fluid was found to have excellent stability and compatibility with rocks and materials of construction in tests up to 316°C (600°F) and for durations equivalent to about 4 yr of Pilot Plant operation. In addition, during these tests, Caloria HT43 was found to develop no problems in fouling of heat-transfer surfaces.

Test objectives relating to the subsystem and operational characteristics were dealt with in a series of SRE subsystem tests. All test objectives were met or exceeded in a 3-month test program completed in December 1976. A large variety of tests were conducted with thermal charging and discharging rates from 0.1 to 2 MWt and with hold periods up to 144 hr. The performance of the subsystem was excellent. Sharp, stable thermoclines were present, even during partial charging and discharging, with variable rates and under repeated cycling. The performance of the TSU was even better than expected for a unit of this size. The unit delivered over 5.1 MWHt of energy, which exceeded the design goal of a minimum of 4 MWHt.

Other conclusions from the SRE subsystem tests are:

1. The practicality and high performance of the dual-medium thermal storage concept have been demonstrated on a significantly large scale. Scale-up to the 10-MWe Pilot Plant can now be made with high confidence.
2. The development and vertical movement of a sharp thermal gradient or thermocline in a dual-medium (liquid-solid) system is a predictable and reproducible phenomenon.
3. Thermocline integrity and stability are adequate to provide high-energy recovery performance for daily operation.
4. Partial charging and extraction, and repeated cycling, do not significantly degrade the thermocline.

5. Fluid flow and temperature uniformity are very high across the thermal storage unit cross-section. Flow channeling and "rat-holing" are insignificant in the SRE unit.
6. Heat loss from the TSU is not severe. With larger units and improved insulation, very little energy would be lost over typical hold periods.
7. The use of low-cost, commercial, river-bed gravel provides adequate performance. Conventional filters incorporated in the charging circuit are adequate to rapidly remove dust from the rock bed.
8. For the test time accumulated, tank wall stresses were far below design values (i.e., passive load design), and there was no indication of high stresses induced from rock settling and packing.
9. Initial removal of water from the bed occurs readily and controllably during initial bed heating. No special design features other than adequate vapor venting capacity are necessary for water removal.

Details of the SRE tests and results are given in Section 6 of this report.

Section 2
THERMAL STORAGE DATA LIST

This section contains tabular summaries of design characteristics and design discussion features, following the list and order of information given in Reference 2-1.

2.1 DESIGN CHARACTERISTICS DATA LIST

Table 2-1 contains tabular summaries of design characteristics for the three TSS categories: Pilot Plant, Commercial Plant, and SRE. The entries and their order of presentation are those given in Reference 2-1.

Further details supporting the table entries, with additional information and discussion of the design work and decisions, are given in Section 2.2 and other sections of this report. Section 1.1 discusses the rationale for the thermal storage concept selected for this solar application. Sections 1.1, 4.3.1, and 6.2 discuss the rationale for selection of the thermal storage media.

The Pilot Plant subsystem design is summarized in Section 4.6; expanded details are given in Section 4.3, with subsections for each major component. Control features and operational characteristics of the Pilot Plant are discussed in Sections 4.3.9 and 4.4. Safety is discussed in detail in Section 4.5, as well as many portions of other sections. Plans for fabrication, installation, checkout, quality assurance, and maintenance are given in Section 5.

Details of the 100-MWe Commercial Plant subsystem design are given in Section 3.2, with subsections for each major component. Control features and operational characteristics of the Commercial Plant are discussed in Sections 3.2.7 and 3.4. The Commercial Plant subsystem plans for fabrication and installation are given in Section 3.3.

The details of the SRE subsystem are given in Section 6.3.2.

Table 2-1 (Page 1 of 7)

THERMAL STORAGE SUBSYSTEM DESIGN CHARACTERISTICS

	SI Units			English Units		
	Pilot Plant	Commercial Plant	SRE	Pilot Plant	Commercial Plant	SRE

1a. Storage Materials

FLUID: Caloria HT43 Heat-Transfer Fluid
ROCK: 1-in. River Gravel (primarily granite)
SAND: Silica Sand No. 6

1b. Properties of Storage Materials:

CALORIA HT43

Specific Heat (Fig. 2-1)

Temperature °C	C_p J/kg°C	Temperature °F	C_p Btu/lb°F
25	1950	77	0.44
100	2110	200	0.50
200	2490	400	0.60
250	2690	500	0.65
300	2890	600	0.70

Density (Fig. 2-1)

Temperature °C	Density 10^3 kg/m^3	Temperature °F	Density lb/ft^3
25	0.851	77	53.1
100	0.801	200	50.3
200	0.733	400	45.6
250	0.697	500	43.0
300	0.657	600	40.3

Viscosity (Fig. 2-2)

Temperature °C	Viscosity 10^{-4} N-s/m^2	Temperature °F	Viscosity lbm/hr-ft
25	580	77	140
100	43	200	12.0
200	10.7	400	2.50
250	6.9	500	1.54
300	4.8	600	1.03

ROCK

Specific Heat

Temperature °C	C_p J/kg°C	Temperature °F	C_p Btu/lb°F
25	828	77	0.198
100	882	200	0.210
200	956	400	0.230
250	995	500	0.240
300	1035	600	0.250

Table 2-1 (Page 2 of 7)

THERMAL STORAGE SUBSYSTEM DESIGN CHARACTERISTICS

	SI Units			English Units		
	Pilot Plant	Commercial Plant	SRE	Pilot Plant	Commercial Plant	SRE

1b. Properties of Storage Materials: (Cont)

ROCK (Cont)

Viscosity (Fig. 2-2)

Temperature °C	Density 10^3 kg/m^3	Temperature °F	Density lb/ft^3
25	2.670	77	166.7
100	2.665	200	166.4
200	2.659	400	166.0
250	2.656	500	165.8
300	2.652	600	165.5

SAND

Specific Heat

Temperature °C	C_p J/kg°C	Temperature °F	C_p Btu/lb°F
25	828	77	0.198
100	882	200	0.210
200	956	400	0.230
250	995	500	0.240
300	1035	600	0.250

Density

Temperature °C	Density 10^3 kg/m^3	Temperature °F	Density lb/ft^3
25	2.600	77	162.3
100	2.595	200	162.0
200	2.589	400	161.6
250	2.586	500	161.5
300	2.582	600	161.2

	SI Units				English Units			
	Units	Pilot Plant	Commercial Plant	SRE	Units	Pilot Plant	Commercial Plant	SRE
2. Quantity of Storage Materials per Stage								
Caloria HT43	megagram	447	7,500	20.7	ton	494	8,225	22.8
	m^3(1)	525	8,800	24.3	1000 gal(1)	139	2,330	6.4
Rock and Sand (2:1 Mixture)	megagram	4,530	81,000	210	ton	4,990	89,300	231
	m^3(1)	2,280	40,800	107	1000 ft^3(1)	80.5	1,441	3.73

(1) Volume at 21°C (70°F)

Table 2-1 (Page 3 of 7)

THERMAL STORAGE SUBSYSTEM DESIGN CHARACTERISTICS

		SI Units				English Units			
		Units	Pilot Plant	Commercial Plant	SRE	Units	Pilot Plant	Commercial Plant	SRE
3.	Tankage								
	Type		Cylindrical, axis vertical, installed above ground				Cylindrical, axis vertical, installed above ground		
	Quantity		1	4	1		1	4	1
	Diameter	m	15.25	27.6	3.2	ft	50	90.5	10.5
	Height	m	13.4	18.3	13.3	ft	44	60	43.7
4.	Tankage Volume	m^3	2,450	43,700	107	1,000 ft^3	86.4	1,444	3.778
5.	Design Capacity for Each Stage	MWHt	103.8	1,857	4	10^6 Btu	354.3	6,338	13.652
6.	Charging Rates for Each Stage	MWt				10^6 Btu/hr			
	Max		30	255	1.2		102.4	870.0	4.096
	Min		1.5	12.5	0.1		5.120	42.66	0.341
7.	Discharging Rates for Each Stage	MWt				10^6 Btu/hr			
	Max		32.1	285	2.0		109.6	972.7	6.826
	Min		3.1	31.36	0.1		10.58	107	0.341
8.	Thermal Storage Material Temperatures When Charged and Discharged for Each Stage	°C				°F			
	Charged		302	316	316		575	600	600
	Discharged		218	232	232		425	450	450
9.	Pinch Points for Each Stage	°C				°F			
	Charging		17.7	10	na(2)		32	18	na(2)
	Discharging		6.7	18.3	na		12	33	na
10.	Number of Heat Exchangers(3)	--							
	Heater		2	5	1			same	
	Preheater		2	5	1				
	Boiler		2	5	1				
	Superheater		2	5	na				
11.	Outside Dimensions of Each Heat Exchanger (diameter x length)	m				ft			
	Heater		1.04 x 9.75	1.19 x 12.2			3.4 x 32.0	4.0 x 40	
	Preheater		0.64 x 9.15	1.14 x 7.93			2.1 x 30.0	3.75 x 26	
	Boiler		1.67 x 10.8	2.29 x 9.15			5.5 x 35.5	7.5 x 30	
	Superheater		0.64 x 3.65	1.4 x 8.2			2.1 x 12	3.75 x 27	

(2) Not applicable
(3) Each parallel steam generator module consists of a feedwater preheater, a boiler, and a superheater

Table 2-1 (Page 4 of 7)

THERMAL STORAGE SUBSYSTEM DESIGN CHARACTERISTICS

	SI Units				English Units			
	Units	Pilot Plant	Commercial Plant	SRE	Units	Pilot Plant	Commercial Plant	SRE
12. Weight of Each Heat Exchanger	Kg				lb			
Heater (6)		24,970	45,632			55,000	100,500	
Preheater (7)		4,105	9,518			9,000	21,000	
Boiler (7)		19,300	30,700			42,500	67,600	
Superheater (7)		1,737	12,250			3,822	27,000	
13. Number of Tubes for Each Heat Exchanger	--							
Heater (6)		450	1,200	3				
Preheater (7)		431	800	(4)		same		
Boiler (7)		650	1,232	8				
Superheater (7)		225	720	na				
14. Heat Exchanger Tube Dimensions (ID/OD)	mm				inch			
Heater		15.7/19	15.7/19	35.1/42.2		0.620/0.75	0.620/0.75	1.38/1.66
Preheater		14.8/19	21.2/25.4	(4)		0.584/0.75	0.834/1.00	(4)
Boiler		14.8/19	14.8/19	35.1/42.2		0.584/0.75	0.584/0.75	1.38/1.66
Superheater		14.8/19	14.8/19	na		0.584/0.75	0.584/0.75	na
15. Heat Exchanger Tube Length for Each Heat Exchanger	m				ft			
Heater (6)		17.2	21.8	46.6		56.6	72.0	153
Preheater (7)		8.54	7.47	(4)		28	24.5	(4)
Boiler (5)(7)		22.9	18.4	32.3		75	60.5	106
Superheater (5)(7)		7.0	15.5	na		23	51	na
16. Total Heat-Transfer Area for Each Heat Exchanger (based on tube mean diameter)	m^2				ft^2			
Heater (6)		464.5	1,670	16.7		5,000	18,000	180
Preheater (7)		196	435	(4)		2,110	4,680	(4)
Boiler (7)		791	1,204	33.6		8,513	12,950	362
Superheater (7)		84	594	na		904	6,390	na
17a. Heat-Exchanger Capacity Rate for Each Heat Exchanger (based on fluid ΔT)	MWt/°C				10^6 Btu/hr°F			
Heater (6)		0.17	0.61			0.33	1.16	
Preheater (7)		0.18	0.66			0.35	1.25	
Boiler (7)		0.20	0.70			0.37	1.32	
Superheater (7)		0.33	0.25			0.63	0.47	

(4) Combined with boiler
(5) Average length of U-tubes
(6) Pilot Plant assembly consists of 2 heat exchangers (HEX); 10/1 turndown ratio for each Heat Exchanger; assembly turndown ratio = 20/1
Commercial Plant assembly consists of 5 heat exchangers (HEX); 4/1 turndown ratio for each Heat Exchanger; assembly turndown ratio = 20/1
(7) Pilot Plant assembly consists of 2 HEX; 5/1 turndown ratio for each HEX; assembly turndown ratio = 10/1
Commercial Plant assembly consists of 5 HEX; 2/1 turndown ratio for each HEX; assembly turndown ratio = 10/1

Table 2-1 (Page 5 of 7)

THERMAL STORAGE SUBSYSTEM DESIGN CHARACTERISTICS

	SI Units				English Units			
	Units	Pilot Plant	Commercial Plant	SRE	Units	Pilot Plant	Commercial Plant	SRE
17b. Heat-Exchanger Effectiveness	--							
Heater		0.777	0.869	variable				
Preheater		0.944	0.857	(4)		same		
Boiler		0.907	0.765	variable				
Superheater		0.657	0.831	na				
17c. Number of Transfer Units for Each Heat Exchanger	--							
Heater		3.12	2.81	variable				
Preheater		3.32	2.11	(4)		same		
Boiler		2.37	1.45	variable				
Superheater		1.23	2.93	na				
17d. Overall Heat-Transfer Coefficient for each Heat Exchanger	w/m^2-°C				Btu/hr-ft^2-°F			
Heater		1,033	909	variable		182.0	160	variable
Preheater		494	465	(4)		87	82	(4)
Boiler		585	829	variable		103	146	variable
Superheater		253	306	na		44.6	54	na
18. LMTD for Each Heat Exchanger	°C				°F			
Heater		33	28.3	na		59.4	51.0	na
Preheater		33.8	51.5	variable		60.8	93	variable
Boiler		27.7	41	variable		49.8	74	variable
Superheater		33.6	32.8	na		60.4	59	na
19. Heat Exchanged for Each Heat Exchanger Unit (min/max)	MWt				10^6 Btu/hr			
Heater(6)		1.0/15	12.5/51	0.1/1.2		3.41/51.2	42.7/174	0.342/4.10
Preheater(7)		0.605/3.125	6.02/10.94			2.06/10.7	20.55/37.34	
Boiler(7)		2.30/11.95	22.83/41.5			7.84/40.8	77.92/141.6	
Superheater(7)		0.18/0.96	2.50/4.56			0.61/3.28	8.53/15.56	
20. Storage Fluid Temperature for Each Heat Exchanger (inlet/outlet)	°C				°F			
Heater		218/304	232/316	232/316		425/580	450/600	450/600
Preheater		236/218	249/232	(4)		456/425	480/450	(4)
Boiler		297/236	308/249	316/232		567/456	587/480	600/450
Superheater		302/273	313/308	na		575/523	595/587	na

(4) Combined with boiler

(6) Pilot Plant assembly consists of 2 heat exchangers (HEX); 10/1 turndown ratio for each heat exchanger; assembly turndown ratio = 20/1
Commercial Plant assembly consists of 5 heat exchangers (HEX); 4/1 turndown ratio for each heat exchanger; assembly turndown ratio = 20/1

(7) Pilot Plant assembly consists of 2 HEX; 5/1 turndown ratio for each HEX; assembly turndown ratio = 10/1
Commercial Plant assembly consists of 5 HEX; 2/1 turndown ratio for each HEX; assembly turndown ratio = 10/1

Table 2-1 (Page 6 of 7)

THERMAL STORAGE SUBSYSTEM DESIGN CHARACTERISTICS

		SI Units				English Units			
		Units	Pilot Plant	Commercial Plant	SRE	Units	Pilot Plant	Commercial Plant	SRE
21.	Storage Fluid Flowrates for Each Heat Exchanger (min/max)	10^3 kg/hr				10^3 lb/hr			
	Heater(6)		23/230	195/795	1.53/18.4		50.7/507	429/1,748	3.38/40.5
	Preheater(7)		49.1/254	473/917	0.173/3.47		108/560	1,040/2,018	0.382/7.65
	Boiler(7)		49.1/254	473/917	0.173/3.47		108/560	1,040/2,018	0.382/7.65
	Superheater(7)		8.2/42.5	161/312	na		18/93.6	354/686	na
22.	Storage Fluid Pressure Loss for Each Heat Exchanger (max)	KPa				psia			
	Heater		170	170	103		25.0	25.0	15
	Preheater		48	48	(4)		7.0	7.0	(4)
	Boiler		34	34	34		5.0	5.0	5
	Superheater		28	48	na		4.0	7.0	na
23.	Storage Fluid Velocity for Each Heat Exchanger (min/max)	m/sec				ft/sec			
	Heater(6)		0.27/2.7	0.68/2.7	1.5/4.6		0.89/8.9	2.2/8.9	5/15
	Preheater(7)		0.27/1.41	0.69/1.34	(4)		0.89/4.61	2.27/4.4	(4)
	Boiler(7)		0.18/0.93	0.37/0.72	0.9/1.8		0.59/3.06	1.21/2.36	3/6
	Superheater(7)		0.08/0.42	0.39/0.76	na		0.26/1.37	1.29/2.5	na
24.	Steam/Water Temperatures for Each Heat Exchanger Unit (inlet/outlet)	°C				°F			
	Heater		343/246	360/249	na		650/475	680/480	na
	Preheater		121/229	121/231	variable		250/444	250/447	variable
	Boiler		229/229	231/231	variable		444/444	447/447	variable
	Superheater		229/277	231/299	na		444/530	447/570	na
25.	Steam/Water Enthalpy for Each Heat Exchanger (in/out)	J/g				Btu/lb			
	Heater		2892/1067	2961/1080	variable		1243/458.6	1273/464.4	variable
	Preheater		507/984	507/991			218.5/424	218.5/427	
	Boiler		984/2796	991/2796			424/1205	427/1205	
	Superheater		2796/2935	2796/2993			1205/1265	1205/1290	
26.	Steam/Water Flowrates for Each Heat Exchanger (min/max)	10^3 kg/hr				10^3 lb/hr			
	Heater (downstream of DSH) (6)		2.96/29.6	24/97.8	na		6.53/65.3	52.75/215.2	na
	Preheater (7)		4.60/23.80	45.45/82.3	variable		10.1/52.4	100/181	variable
	Boiler (7)		4.60/23.80	45.45/82.3	variable		10.1/52.4	100/181	variable
	Superheater (7)		4.60/23.80	45.45/82.3	variable		10.1/52.4	100/181	variable

(4) Combined with boiler
(6) Pilot Plant assembly consists of 2 heat exchangers (HEX); 10/1 turndown ratio for each heat exchanger; assembly turndown ratio = 20/1
Commercial Plant assembly consists of 5 heat exchangers (HEX); 4/1 turndown ratio for each heat exchanger; assembly turndown ratio = 20/1
(7) Pilot Plant assembly consists of 2 HEX; 5/1 turndown ratio for each HEX; assembly turndown ratio = 10/1
Commercial Plant assembly consists of 5 HEX; 2/1 turndown ratio for each HEX; assembly turndown ratio = 10/1

Table 2-1 (Page 7 of 7)

THERMAL STORAGE SUBSYSTEM DESIGN CHARACTERISTICS

		SI Units				English Units			
		Units	Pilot Plant	Commercial Plant	SRE	Units	Pilot Plant	Commercial Plant	SRE
27.	Steam/Water Pressure Loss for Each Heat Exchanger (max)	KPa				psia			
	Heater		103	103	na		15.0	15.0	na
	Preheater		17.2	20.7	variable		2.5	3.0	variable
	Boiler		6.89	6.89	variable		1.0	1.0	variable
	Superheater		34.5	41.4	variable		5.0	6.0	variable
28.	Conditions of Charging Steam from Receiver								
	Rated	MPa	10.1	10.1	na	psia	1,465	1,465	na
		°C	510	510	na	°F	950	950	na
	Derated	MPa	10.1	10.1	na	psia	1,465	1,465	na
		°C	343	360	na	°F	650	680	na
29.	Conditions of Steam Discharging from Storage	MPa	2.76	2.72	variable	psia	400	395	variable
		°C	277	299	variable	°F	530	570	variable
30.	Feedwater temperature to thermal storage steam generator	°C				°F			
	Max Flowrate		121	121	variable		250	250	variable
	Min Flowrate		121	121	variable		250	250	variable
31.	Number of extraction stages used for feedwater heating	--	—	—	—	--	—	—	—
32.	Number of degrees of superheat in steam from thermal storage	°C	47.4	68.3	0	°F	85.3	123	0
33.	Flow of charging steam from receiver to thermal storage (upstream of DSH)	10^3 kg/hr				10^3 lb/hr			
	Rated: Min		2.38	19.7	na		5.23	43.3	na
	Max		47.27	401	na		104	883	na
	Derated: Min(8)		—	—	na		—	—	na
	Max		59.3	489	na		130.5	1,076	na
34.	Flow of discharging steam from thermal storage	10^3 kg/hr				10^3 lb/hr			
	Min		4.58	45.5	variable		10.1	100	variable
	Max		47.5	412			104.7	905.6	

(8) Not limiting case for design (i.e., above rated steam minimum)

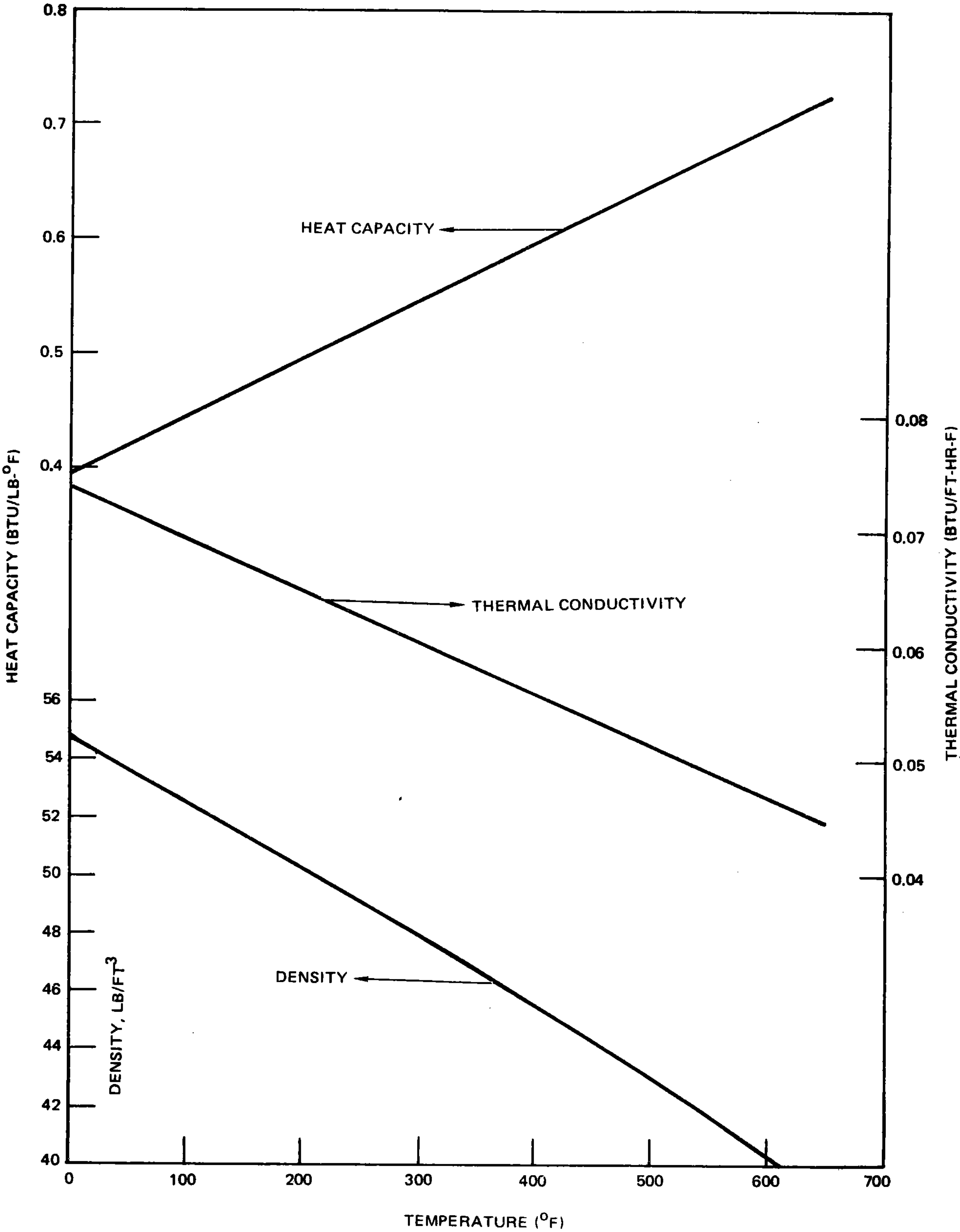

Figure 2-1. Properties of Exxon's Caloria HT43

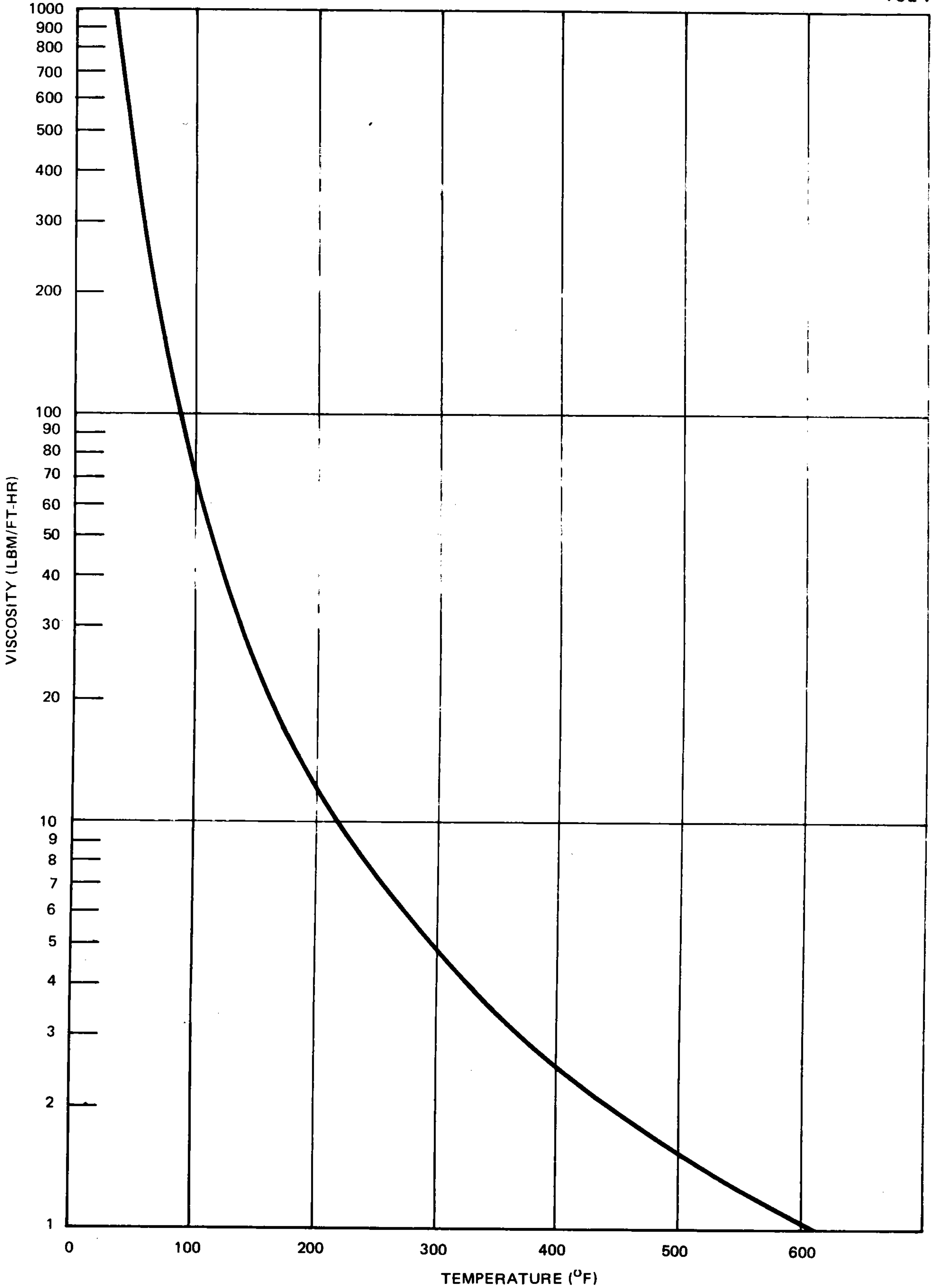

Figure 2-2. Viscosity of Exxon's Caloria HT43

2.2 DESIGN DISCUSSION

This section gives information on fourteen design discussion features, as listed in Reference 2-1. Given are brief summaries and references to the body of the report where details of the listed design item can be found.

(1) A description of the design, including the design rationale and functional role, cost/performance analyses/tradeoff studies, analytical tools used, applicable safety codes, allowable stresses and safety factors, safety precautions, reliability, manufacturing and maintenance for the following elements:

a. Fluid Maintenance Unit: this assembly is comprised of commercially available components, which are combined in a way to perform three functions related to maintaining the heat-transfer fluid (Caloria HT43) in good condition for operation throughout the life of the plants. The basic functions of the unit are: (1) removal of particulate solids in the fluid stream (using commercial filters), (2) removing polymerized organic material which may form gradually over long periods of time (using a commercial thin-film evaporator, operating intermittently on a side stream of the fluid), and (3) adding fresh makeup fluid to maintain a constant fluid inventory. Details about the unit can be found in the report sections listed below.

Topic	Commercial TSS	Pilot TSS
Design, rationale, analyses and trades, analytic tools, etc	3.2.4	4.3.3
Safety	4.5	4.5
Reliability	3.2.4, 5.2.5, 5.3.4	4.33, 5.2.5, 5.3.4
Manufacturing and maintenance	3.2.4, 3.3	4.3.3, 5.2, 5.3, 5.4

b. Tankage: the tanks(s) for the TSU(s) are welded, carbon-steel tanks fabricated by commercial fabricators using current standard techniques. Sizing and design involves some special analysis. The details and results are given in the report sections listed below.

Topic	Commercial TSS	Pilot TSS
Design, rationale, trades, etc.	3.2.2	4.3.1
Safety	4.5	4.5
Reliability, manufacturing and maintenance	3.2.2, 3.3	4.3.1, 5.2, 5.3, 5.4

c. Heat Exchangers: there are three types of heat-exchanger units in the subsystem: charging exchanger (or thermal storage heater), desuperheater, and steam generator. All are standard, commercial components which will be designed, manufactured, installed, tested, and maintained according to long-established industry practice.

Topic	Commercial TSS	Pilot TSS
Design, rationale, etc:		
for desuperheaters	3.2.5.1	4.3.4
for thermal storage heaters	3.2.5.2	4.3.5
for steam generators	3.2.6.1	4.3.7
Safety (all exchangers)	4.5 plus design sections	4.5 plus design sections
Reliability, manufacturing and maintenance	3.3 plus design sections	5.2, 5.3, 5.4 plus design sections

d. Control Systems: The control systems for both Commercial and Pilot TSS are identical in concept. Each TSS is required to respond automatically to commands from operating personnel and/or from the central controller. Each TSS is designed to change smoothly and rapidly between various operating modes and to respond to changes in thermal charging rate and extraction rate. Details can be found in the report sections listed below.

Topic	Commercial TSS	Pilot TSS
Overall subsystem control concept	3.2.1	Lead-in paragraphs to 4.3
Design, rationale, trades, etc	3.2.7	4.3.9

Further information on functional role and operation	3.4	4.4
Safety	4.5, 3.2.7	4.5, 4.3.9
Reliability, manufacturing and maintenance	3.3, 3.2.7	5.2, 5.3, 5.4

e. Ullage Maintenance Unit: this unit is an assembly of commercial components which has the functions of: (1) maintaining ullage pressure in the TSU tanks, (2) providing an inert (nitrogen) gas ullage, and (3) permitting tank venting as necessary. Details are given in the sections listed below.

Topic	Commercial TSS	Pilot TSS
Design, rationale, trades, etc	3.2.3	4.3.2
Safety	4.5, 3.2.3	4.5, 4.3.2
Reliability, manufacturing and maintenance	3.3, 3.2.3	5.2, 5.3, 5.4, 4.3.2

f. Fluid Flow Loops: there are two major fluid flow loops (charging and extraction loops) in each TSS. Each is assembled entirely of standard, commercial components (e.g., pipe, fittings, valves, and pumps). Details are given in the listed sections.

Topic	Commercial TSS	Pilot TSS
Overall flow loop diagram	3.2.1	Lead-in paragraphs to 4.3
Design, rationale, trades, etc	3.2.5.3(charging) 3.2.6.2 (extraction)	4.3.6(charging) 4.3.8 (extraction)
Functional role	3.4 plus design sections	4.4 plus design sections
Safety	4.5 plus design sections	4.5 plus design sections
Reliability, manufacturing and maintenance	3.3 plus design sections	5.2, 5.3, 5.4 plus design sections

(2) The selected thermal storage materials are Caloria HT43, rock and sand, combined in a dual-medium-type of storage unit. The rationale for their selection and related matters are covered in detail in the sections listed in the following paragraphs.

The storage material selection rationale is covered in Sections 4.3.1, 4.5, 6.2.2, and 6.2.3. Specifically, material properties and cost tradeoff studies are discussed in 4.3.1.2 under the heading "Selection of Dual Heat Storage Media." Safety is discussed in 4.5. Material stability and compatibility testing is discussed in 6.2.2. Test results, including heated surface fouling, weight loss, viscosity versus time, and annual fluid makeup requirements, are discussed in 6.2.3. Specific fluid makeup requirements for the Pilot Plant are found in 4.3.3, and for the Commercial Plant in 3.2.4. Analytical methods to determine storage material quantities are treated in 4.3.1.2 under the headings "Dual-Medium Heat Storage Analysis" and "Dual-Medium Thermal Design."

(3) Insulation covering the TSS will reduce thermal losses to the range of 2 to 3% on the Pilot Plant and below 2% in the Commercial Plant over a 24-hr period.

Insulation for the TSS has been selected to provide a maximum amount of energy recovery with a safe and cost effective approach. Since thermal energy loss from the storage system is a direct loss in electrical power generation, it is essential that thermal losses be kept to a minimum. Since over 99% of the heat is stored in the TSU, the type and thickness of the TSU insulation was investigated in depth. From the results of cost optimization studies (Reference 2-2), it was determined that insulation thickness is not a major component of TSS cost; therefore, it was practical to use a good grade of low-conductivity, high-temperature insulation for the sides and top of the TSU, i.e., 200 mm (8 in.) of Owens-Corning Intermediate Service Board (ISB) with a 0.5 mm (0.020 in.) aluminum weather covering. The TSU tank is all welded, which allows the use of open-pore insulation. All instrumentation and fluid line connections will be made to welded nozzles which protrude past the weather covering. No threaded or bolted connections will be made under the insulation.

The bottom of the TSU is another heat-loss path. Foundation construction alternatives considered included: (1) firm bolting to a strong insulating concrete slab, (2) unattached to a low-density insulating concrete slab, and (3) setting on dry soil. Consideration of earthquake effects, soil bearing loads, and heat transfer led to the selection of alternative number three. With respect to heat transfer, dry soil is equivalent to low-density concrete and provides an acceptable thermal barrier to meet overall heat-loss requirements. Soil adjacent to the TSU will be paved to retain thermal insulation during periods of rain. Dry soil foundations used with large petroleum storage tanks have successfully sustained major earthquakes without damage. Tanks bolted to a firm foundation frequently fail at the bolt connections or the bolts pull out of the concrete slab. Lateral movement during a quake is inhibited by sloping the soil into a low-angle cone. Fluid line connections will be long enough to provide ample flexibility for both thermal expansion and earthquake movement.

Pittsburgh-Corning Foamglas insulation has been selected for the lines, heat exchangers, valves, and ancillary equipment. This type of insulation is closed pore and is best suited for portions of the system with bolted or threaded connections. Experience with heat-transfer fluids at high temperature indicates that some seepage can be expected. The use of closed-pore insulation plus metal weather cover will reduce fire susceptability to an absolute minimum.

Additional details can be found in the report sections listed below.

Topic	Commercial TSS	Pilot TSS
TSU insulation	3.2.2.2	4.3.1.2
TSU description	3.2.2.3	4.3.1.3
Piping description	3.2.5.3	4.3.6.3, 4.3,8.3

4. Correlations for the heat-transfer characteristics of the dual-medium storage units appear in Section 4.3.1.2 under the heading "Dual-Medium Heat Storage Analysis." Pressure drops are discussed in Section 4.3.1.2 under the heading "Fluid Distribution and Bed Packing."

   Correlations for pressure drop, heat transfer, and fouling of the steam generator appear in Sections 3.2.6.1 (CP) and 4.3.7.2 (PP) Similar correlations were used in analyses of the charging exchangers; the results were combined with analyses by several commercial exchanger design/fabrication firms, each of which used their standard design techniques and correlations. These calculations are discussed in Sections 3.2.5.2 (CP) and 4.3.5 (PP).

5. The TSS is designed with considerable flexibility to accommodate variations in temperature, pressure, and flowrate from the receiver and to the EPGS.

Factor	Commercial TSS	Pilot TSS
Steam from receiver to TSS:		
Thermal rate, MWt	12.5 to 255	1.5 to 30
Flowrate, kg/hr	24 to 489,000	2,965 to 59,310
Temperature, °C	360/510	343/510
Pressure, MPa	10.1 ± 0.1	10.1 ± 0.1
Steam from TSS to EPGS:		
Thermal rate, MWt	31.4 to 285	3.1 to 32.1
Flowrate, kg/hr	45,450 to 412,000	4,590 to 47,590
Temperature, °C	299 ± 4	277 ± 4
Pressure, MPa	2.72 ± 0.069	2.76 ± 0.069

   The variations in conditions from the receiver are nominal values, based on expected requirements; greater variations could be accommodated if necessary. The variations in steam conditions to the EPGS were set rather broadly, since tighter requirements by the turbine are not necessary; but tighter tolerances could be imposed if necessary.

Additional details are given for the Commercial TSS in Tables 3-1, 3-2, and 3-6, and Figures 3-1, 3-2, and 3-11 (found in Section 3); and for the Pilot Plant TSS in Tables 4-1, 4-2, and Figure 4-2 (found in Section 4).

6. Discussions regarding the time required for the TSS to switch between various operating modes are given in Section 3.4.7 for the Commercial Plant and Section 4.4.7 for the Pilot Plant TSS. A brief summary is given below:

Estimates of the time required for switching from:

a. Operation directly from the receiver to operation both from the receiver and from thermal storage:

   If the TSS is in hot standby mode, then the estimated time is 5 minutes.

   If the TSS is not in hot standby mode, then the estimated time is 30 minutes.

b. Operation directly from the receiver to operation entirely from thermal storage:

   The times are estimated to be virtually the same as for those given in "a" above.

c. Operation entirely from thermal storage to operation directly and entirely from the receiver:

   The time required is limited by factors outside the TSS rather than those associated with the TSU.

d. Charging thermal storage to discharging thermal storage.

Charging and discharging are entirely independent operations which have only negligible influence on each other. The charging heat-exchanger thermal inertia drives the charging time delay. The same is true of discharge, except that on hot standby mode the discharge limiting time lag is much shorter, but is still limited by thermal inertia factors.

Charging and discharging simultaneously are entirely possible as long as the TSU charge condition is not very close to being either fully charged or fully discharged. Even so, if the TSU is near the fully charged condition, simultaneous charge and discharge can still take place freely as long as the charging rate does not exceed the discharging rate. Similarly, if the TSU is near the fully discharged condition, then the system can operate in the simultaneous charge and discharge mode as long as the discharging rate does not exceed the charging rate.

When the Commercial design is compared with the Pilot Plant, the control responses are similar, but the thermal inertias for the heat exchangers result in longer controlling time constants than with the Pilot Plant.

(7) Variations in temperature, pressure, and flowrate in the feedwater system from the EPGS are acceptable since the components involved (steam generator and desuperheater) contain control elements that adjust flowrate over a considerable range to achieve desired output conditions.

The steam generation system can tolerate reduction in pressure to the level where feedwater flow is lower than demand flow into the kettle boiler. At this point the control system will signal reduced capability and electrical power demand will be reduced. High feedwater pressure will have little or no impact. Low feedwater temperature can be tolerated to the point where the preheater cannot supply adequate heat before the water enters the kettle boiler. Again this will be signaled as a decrease in operating power level. High feedwater temperature will have little or no impact.

The preheater, boiler, and superheater have approximately 10 to 20% design margin which will be available for out-of-tolerance operation when new. However, this margin will gradually decrease with time. Feedwater flowrate into the steam generation system is established previously by the steam demand. If the feedwater flowrate cannot meet these demands, restricted power level operation will occur.

Feedwater into the desuperheater is used to detemperize the incoming steam when above rated values. Variations in water temperature have little effect on demand since the heat of vaporization of water is 1000 times the sensible heat per degree. Increases in pressure will be compensated for by the inlet throttle valve. Decreases in pressure will have no effect on the desuperheater until the water pressure is below that required to overcome the pressure drop through the throttle valve and spray nozzles into the desuperheater. Below this value of pressure a reduction of steam temperature or flow from the receiver will be as required to avoid overheating the heat-transfer fluid. Again, as in the steam generator, any restriction in flow below demand flow will be signaled as a reduction in energy receiving capability at rated steam conditions and will require appropriate action by the master controller.

It should be noted that many characteristics will be established during the detailed design portion of Phase II including: equipment limit values of temperature, pressure, and flow; compensation control circuit limits; alternate courses of action; override commands; and other characteristics of the operating parameters. Flexibility is included in the control design so that these characteristics can be adjusted and changed as desired during the checkout period.

Additional information is contained in Sections 3.2.5.1, 3.2.6.1, 3.2.7, and 3.4 (CP) and in Sections 4.3.4, 4.3.7, 4.3.9, and 4.4 (PP).

(8) The Master Control supplies a steam weight flowrate demand level and steam generator output reference pressure level, for set points in the extraction controller. It also supplies anticipatory information to the TSS operating personnel (as discussed under item 9). Additional details are given in Sections 3.7.7 (CP) and 4.3.9 (PP).

(9) The TSS supplies Master Control with the following information:

a. The existing TSU vertical temperature distribution is transmitted to Master Control so it can calculate the TSU charge level and TSU average temperature and so it can initiate a warning signal if the average temperature is below a set value (typically 177°C).

b. Quantity indications of fluid and ullage maintenance gas in the TSS, primarily: (1) TSU fluid level, (2) fluid makeup tank liquid level, (3) GN_2 bottle bank pressure, (4) GN_2 reserve bottle bank pressure, (5) ullage gas storage tank pressure. These parameter levels are for information only and are values normally supplied to the TSS control panel, and thus it is not an absolute requirement that they be sent to Master Control.

c. When indications of unusual operating conditions occur, for example, if the TSS should require one of the two extraction fluid flow pumps to shut down because of an abnormal condition, a signal is sent automatically to Master Control so that Master Control will not command an extraction steam weight flowrate which will exceed the TSS extraction capability when operating on only one pump. Similarly, if controlled parameters are out of the specified error band, enunciator signals sent to the TSS control panel will be sent also to Master Control for information purposes.

Additional information is given in Sections 3.2.7 and 3.5.7 (CP) plus 4.3.9 and 4.5.7 (CP).

(10) The TSS has three control loops: (1) thermal storage fluid temperature control at the outlet of the thermal charging heat exchangers, active only during TSS charging mode, (2) steam generator output mass flowrate and superheat temperature control, active only during thermal extraction, (a boiler liquid level control loop is provided also), and (3) receiver steam desuperheater control to provide constant-temperature dry steam to the input of the TSS charging heat exchangers.

Master Control provides a steam weight flowrate demand level (and steam pressure level) to the steam generator control system. It also supplies predictive information regarding impending TSS mode changes to the TSS control panel, permitting operating personnel to preset control set points as applicable. This function can be automated subsequently if required.

Master Control also furnishes operating personnel with an updated figure showing TSU thermal charge level and TSU average temperature. If the average temperature is below a set value (typically 177°C) then Master Control sends a warning signal to the TSS control panel that the liquid level is too low to permit heat extraction from the TSS. (A charging mode must be used first to store some heat.)

Additional information is given in Sections 3.2.7 and 3.5.7 (CP), plus 4.3.9 and 4.5.7 (PP).

(11) Subsystem energy recovery efficiency (energy out versus energy in) plus parasitic losses are tabulated below.

	Commercial TSS	Pilot TSS
Thermal Energy In, MWHt	1891	106.6
Thermal Energy Out, MWHt	1857	103.8
System Energy Recovery Efficiency	0.982	0.974
Charging Pump Energy, MWHe	7.1	0.4
Extraction Pump Energy, MWHe	5.9	0.3
UMU Storage Compressor, MWHe	0.08	0.007
FMU Vacuum and Return Pump	negligible	negligible
Total Parasitic Energy, MWHe	13.08	0.707
Total Parasitic Thermal Energy, MWHt	52.3	2.8
Total Energy In, MWHt	1943	109.4
Overall Subsystem Energy Efficiency	0.956	0.949

(12) Figure 2-3 provides a summary of the energy accounting of the thermal energy in and out of the TSS. It includes all energy in the nominal operating range for the Pilot and Commercial Plants. Thermal energy requirements for seal steam are satisfied by using

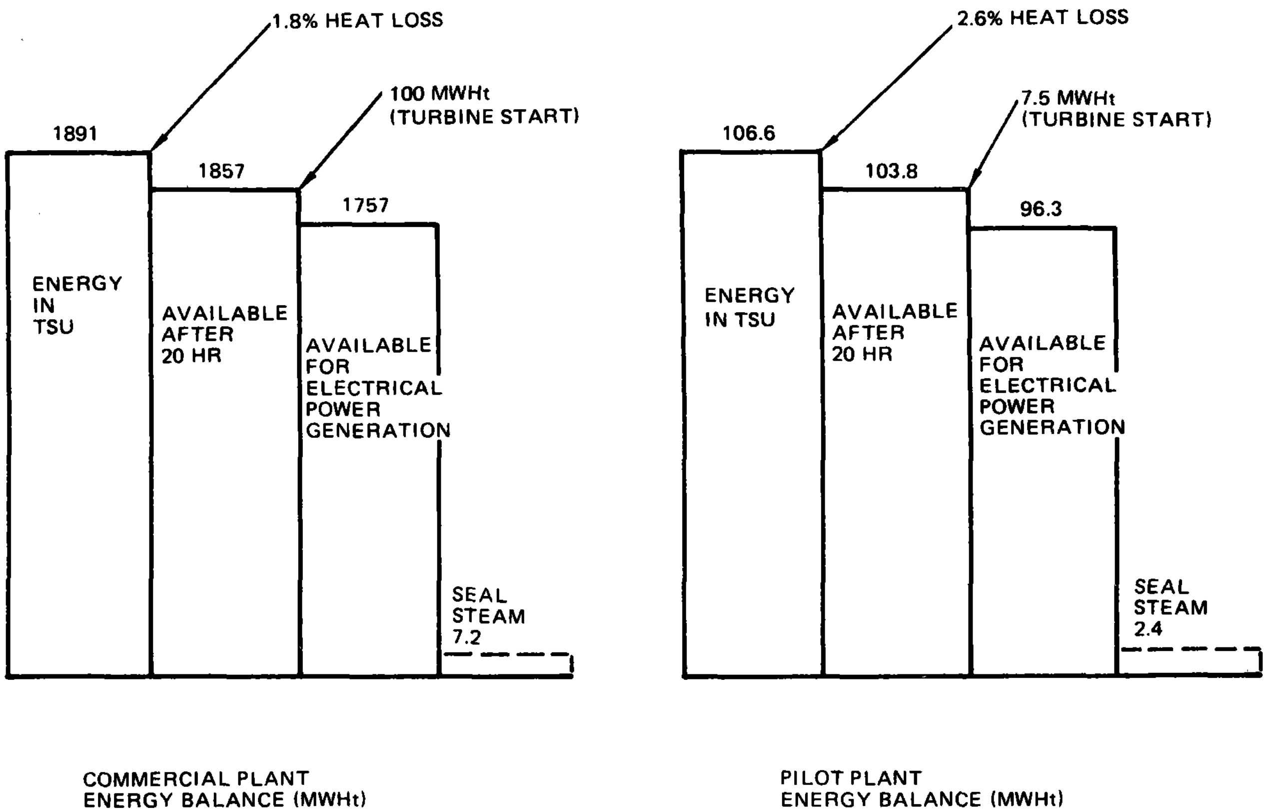

Figure 2-3. Summary of Accounting of Thermal Energy In and Out of Thermal Storage Subsystem

fluid below the nominal operating temperature range. Therefore, it is not part of the above accounting, but it is shown for reference. Turbine hot standby is assumed to be part of the EPGS energy.

(13) TSS schematics are given in several different levels of detail. For the Commercial TSS, a simplified schematic flow diagram, showing the major components and lines plus key process flow conditions, is included both as Figure 1-2 and as Figure 3-2; the detailed schematic diagram, Figure B-3, is enclosed in Appendix B of this volume as a fold-out print. For the Pilot Plant TSS, a simplified schematic flow diagram, showing the major components and lines plus key process flow conditions, is included both as Figure 1-3 and as Figure 4-2. Figure 4-2 is a more detailed schematic, showing controls and all components. Figure B-1, enclosed in Appendix B of this volume, is the most detailed schematic.

Detailed descriptions of all operating modes of thermal storage, including the initial charging process and startup and shutdown modes, are given in Sections 3.4 and 4.4 for Commercial and Pilot TSS's, respectively.

(14) Temperature-enthalpy diagrams for charging and discharging are given (with charging steam, fluid, and discharging steam shown on the same diagram) in Section 3.2.6.1 (CP) and Section 4.3.7.2 (PP).

2.3 REFERENCES

2-1 A.C. Skinrood. Central Receiver Solar Thermal Power System Pilot Plant Preliminary Design Report (PDR) Requirements. Letter dated 11 February 1977, Sandia Laboratories, Livermore, California.

2-2 Central Receiver Solar Thermal Power System, Pilot Plant Preliminary Design Baseline Report. MDC G6040, McDonnell Douglas, Huntington Beach, January 1976, Volume 1, Book 2.

Section 3
COMMERCIAL THERMAL STORAGE SUBSYSTEM DEFINITION

3.1 SUBSYSTEM REQUIREMENTS

The thermal storage subsystem (TSS) "buffers" the electrical power generating subsystem (EPGS) from excessive variations in insolation, and extends the plant's generating capacity into periods with low or no insolation. The general requirement for the TSS is to provide a means of converting a portion of the thermal output from the receiver subsystem into stored thermal energy and subsequently transferring this stored thermal energy to steam in a form suitable for generating electrical power with the conventional turbine-generator in the EPGS. More specific requirements for the Commercial plant TSS are given in the following subsections.

3.1.1 Functional Requirements

3.1.1.1 Fluid Conditions

There are five fluid streams crossing the boundaries of the TSS, made up of all the water or steam flows which enter or leave the subsystem. These five streams are shown in Figure 3-1. The required fluid conditions and flow rate ranges for each of these five streams are summarized in Table 3-1.

All other fluid conditions and flowrates are derived from the requirements for the five basic streams, given in Table 3-1. Although not a requirement, per se, it was established early in the design that the heat-transfer fluid in the extraction loop would cycle between 232°C (450°F) and 316°C (600°F).

3.1.1.2 Performance

The TSS is required to have an extractable storage capacity of at least 1,857 MWHt, which is comprised of 100 MWHt to provide a turbine hot start and 1,757 MWHt to permit the turbine-generator to supply 70 Mwe net

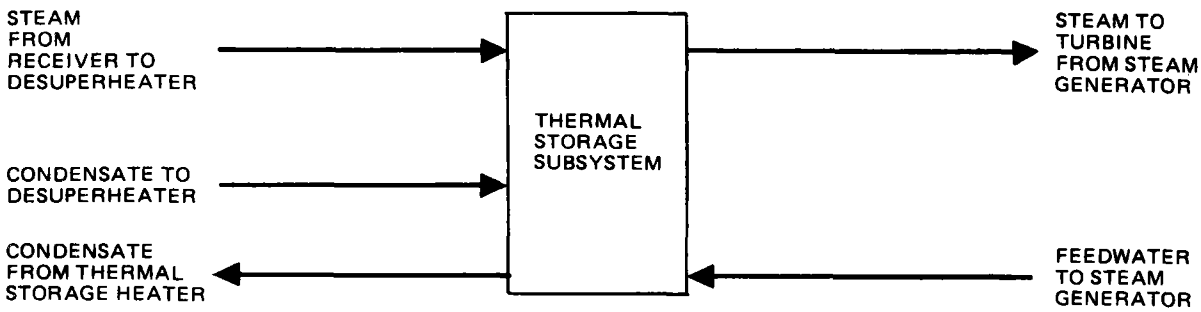

Figure 3-1. Process Streams Crossing TSS Boundaries

Table 3-1

PROCESS STREAM REQUIREMENTS FOR COMMERCIAL PLANT TSS

Process Stream	Case 1 - Charge with Rated Steam	Case 2 - Charge with Derated Steam	Case 3 - Extract Energy from Storage
Receiver stream to desuperheater inlet:			
Temperature, °C (°F)	510 (950)	360 (680)	No flow
Pressure, MPa (psia)	10.1 (1465)	10.1 (1465)	
Maximum flow, kg/hr (lb/hr)	330,450 (727,000)	489,000 (1.076×10^6)	
Minimum flow, kg/hr (lb/hr)	19,680 (43,300)	*	
Condensate to desuperheater:			
Temperature	218 (425)	No flow	No flow
Maximum flow,	72,000 (158,400)		
Minimum flow,	4290 (9,440)		
Condensate from thermal storage heater:			
Temperature,	249 (480)	249 (480)	No flow
Pressure,	9.7 (1400)	9.7 (1400)	
Maximum flow,	402,400 (885,250)	489,000 (1.076×10^6)	
Minimum flow,	23,980 (52,750)	*	
Feedwater to steam generator:			
Temperature,	No flow	No flow	121 (250)
Pressure,			2.90 (420)
Maximum flow,			411,600 (905,600)
Minimum flow,			45,450 (100,000)
Minimum flow,			45,450 (100,000)
Steam from steam generator to turbine:			
Temperature,	No flow	No flow	299 (570)
Pressure,			2.72 (395)
Maximum flow,			411,600 (905,600)
Minimum flow,			45,450 (100,000)

*Not limiting case for design

(76 Mwe gross) for 6 hr following turbine start-up. This extractable capacity is to be available following a full charge and a 20-hour hold period. The required charging rates are 12.5 MWt (rated steam operation) to 255 MWt (derated steam operation). Required discharging rates are 31.1 to 285 MWt. The maximum allowable heat loss is 2% of extractable capacity in 24 hr, starting in a fully charged condition. The subsystem is also required to provide nighttime seal steam at a temperature of at least 135°C (275°F) and at a rate of 0.42 MWt for approximately 12 hr.

3.1.1.3 Operational

The subsystem is required to operate stably and safety in all normal and emergency operating modes. The major operating modes are: startup (both cold and hot), charging, discharging, intermittent cloud, shutdown, standby, and emergency modes. These modes and operating characteristics are described in detail in Section 3.4.

3.1.2 Physical Requirements

3.1.2.1 Interface

The TSS has interfaces with the receiver subsystem and the EPGS. The requirements in terms of process flows and conditions are given in Section 3.1.1. Additional physical requirements are that the piping, connections, and mounting fixtures will match those of: (1) the receiver downcomer, (2) the EPGS at the entrance of the turbine automatic admission port header, and (3) the outlet of the EPGS condensate loop.

The TSS also has requirements based on its interface with master control. The subsystem controls must be responsive to standard control signals (per power industry practices) from master control. There are two pairs of major control commands: (1) start/stop energy storage and (2) start/stop steam generation from stored energy. In addition to these major commands from master control, the interface must provide for throttling controls imposed by master control on the basis of subsystem status measurements sent to master control and on variations in steam and water flow rates imposed on the subsystem by the interfacing subsystems. The subsystem control must respond to such flow rate modulations by metering heat-transfer-fluid flows and subsystem functions. The TSS is also required to accept override commands, including complete control, by master control.

3.1.2.2 Environmental

The environmental requirements on the TSS are identical to those imposed on the total plant; these requirements are given in Volume 2. The primary requirements which are of significance in design of the TSS are earthquake forces, ambient temperatures, and wind velocities (the latter two of primary importance in calculating heat losses and insulation requirements). Design requirements for horizontal earthquake loadings are 0.165g operational and 0.33g safe shutdown; vertical components are two-thirds of horizontal components. Temperature extremes for survivability are -30 to 60°C (-22 to 140°F). Conditions for heat-loss calculations are an ambient temperature of 28°C (82.6°F), 3.5 m/s (8 mph) wind speed at 10 m elevation, and a velocity profile with velocity in m/s equal to $3.5\,(H/10)^{0.15}$, with height H in meters.

For design purposes, safety margins will be used that are commensurate with availability and performance requirements to ensure operation during and/or after exposure to the environmental conditions, as appropriate, for the 30-yr life of the subsystem.

3.1.2.3 Structural

All critical components of the TSS must be designed and installed such that the environmental and site conditions described in Section 3.1.2.2 do not induce a dynamic environmental condition which exceeds the structural capability of the component. All components must be designed to withstand handling and hoisting inertial loads, as applicable, during fabrication, transportation, installation, and maintenance.

The thermal storage unit (TSU) is required to be designed in accordance with API Standard 650 as modified by the ASME Boiler and Pressure Vessel Code - Section VIII for elevated temperature operation. All heat exchangers are to be designed, fabricated, and inspected in accordance with the ASME Boiler and Pressure Vessel Code - Section VIII. Piping is to be designed, fabricated, and inspected in accordance with the American National Standard Institute Code for Pressure Piping, ANSI B31.3.

Materials of construction throughout are to be selected to ensure compatibility with the process fluids at the maximum operating conditions.

3.1.2.4 Safety

The TSS must be designed to minimize safety hazards to operating and service personnel, the public, and the equipment. Electrical components must be insulated and grounded. All parts or components operating at elevated temperatures must be insulated against contact with or exposure to personnel. Any moving elements must be shielded to avoid entanglement, and safety override controls and interlocks must be provided for servicing. Isolation valves are to be provided on all major assemblies and on all interface utility lines to permit isolation and shutdown of assemblies and segments of the subsystem. Concrete and/or earth berms and dikes are to be provided to contain the maximum quantity of heat transfer fluid which can be emptied from all above-grade sections of the subsystem. Safety showers and eyewashes must be provided. Ladders, handrails, and platforms must meet OSHA standards. Fire-protection equipment must be provided.

3.2 DESIGN CHARACTERISTICS

Design characteristics are given and discussed in this section for the entire Commercial Plant TSS, with a subsection devoted to each major assembly. Given for each major assembly are details of the requirements, design analyses, and trade studies made in developing the preliminary design, also presented is a description and discussion of the selected preliminary design.

3.2.1 Subsystem Summary Description

The 100-MWe Commercial Plant TSS employs sensible-heat storage using dual liquid and solid media for the heat storage in each of four tanks, with the thermocline principle applied to provide high-temperature, extractable energy independent of the total energy stored.

In the cyclical operation, heating of the bed (charging) is achieved by removing lower temperature fluid from the bottom of the bed, heating it in a heat exchanger with steam from the receiver, and returning the fluid to the top of the tank. The fluid flow is reversed for heat extraction.

Figure 3-2 is a simplified schematic flow diagram of the Commercial Plant showing the major components and lines. Key process flow conditions are shown at various points in the subsystem. Figure B-3, in Appendix B of this volume, is a detailed schematic.

As shown in Figure 3-2 the subsystem can be considered in three major parts: (1) the central TSU, (2) the thermal charging loop, and (3) the heat extraction loop. In the charging loop, energy is removed from the receiver steam and stored in the TSU tank. A commercial heat-transfer fluid is used to permit economical ambient pressure storage in the tank. The extraction loop uses the fluid to remove energy from the storage unit and produce steam for either power plant operation or heating the feedwater returned to the receiver for operation with low-solar insolation. Additional details are given in the following sections, each dealing with a major components or assembly of the subsystem.

3.2.2 <u>TSU Design</u>

The function of the TSU is to act as a reservoir for solar thermal energy charging the TSU during hours of high insolation. At other times, when insolation is partially or completely unavailable, thermal energy is extracted from the TSU to produce steam for the electrical generation subsystems. This section deals with the requirements, design analyses and optimizations, and the design description of the TSU for the 100-MWe Commercial Plant.

/

3.2.2.1 Requirements

The steady-state operating requirements to be met for energy charging and extraction are shown in Table 3-2. Design requirements for the TSU are listed in Table 3-3. The TSU design criteria for the 100-MWe Commercial Plant, developed to fulfill the requirements listed in these two tables,

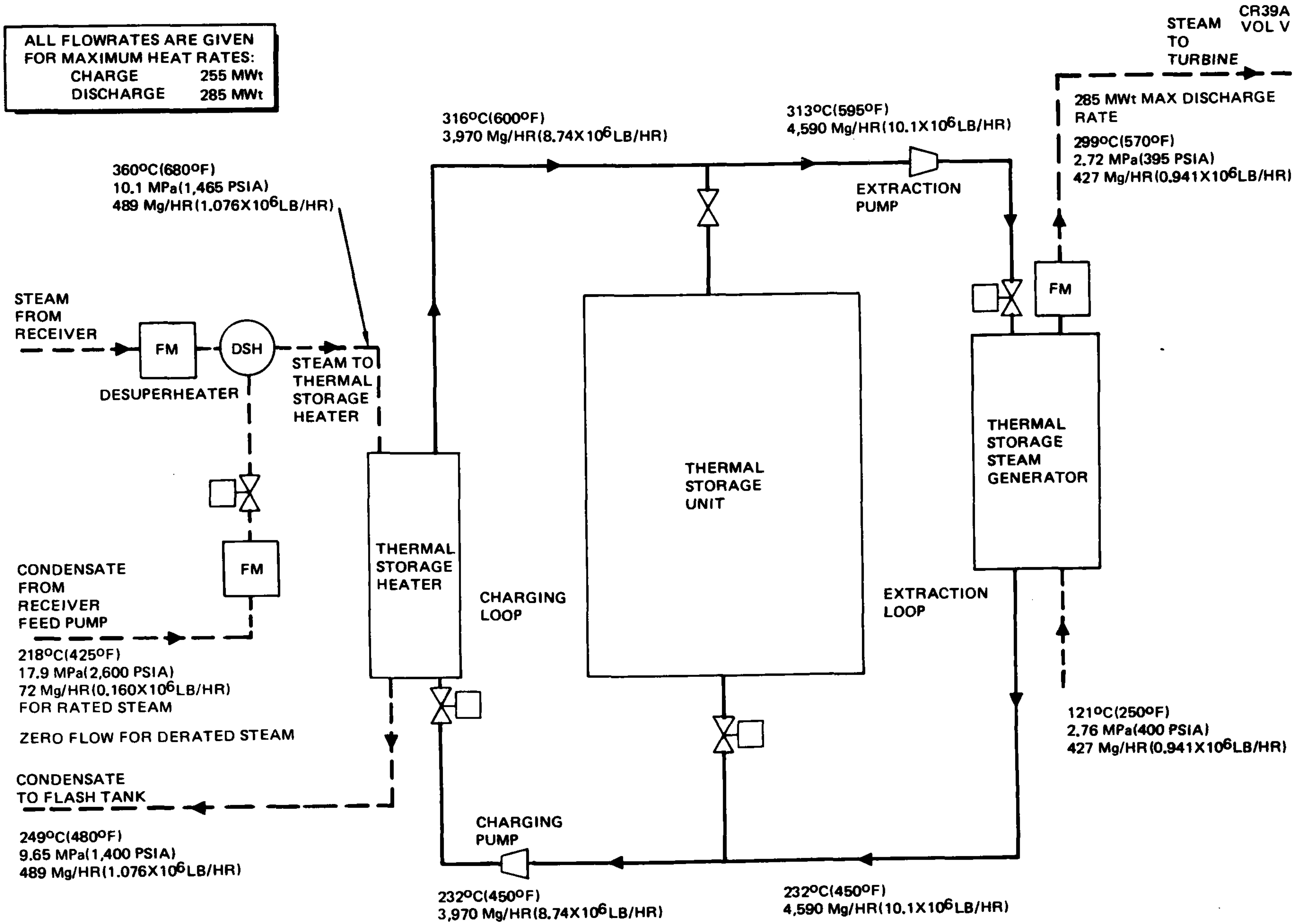

Figure 3-2. Schematic Flow Diagram of Thermal Storage Subsystem for 100-MWe Commercial Plant

Table 3-2

100-MWe COMMERCIAL PLANT THERMAL STORAGE REQUIREMENTS

Design storage temperature:

Max 318°C (600°F)
Min 232°C (450°F)

Extractable capacity after 20-hr hold time:

Net	1,757 MWHt ⟵	($6,000 \times 10^6$ Btu)
Turbine hot start	100 MWHt ⟵	(341×10^6 Btu)
Total	1,857 MWt ⟵	($6,341 \times 10^6$ Btu)

Allowable degradation of thermal storage fluid temperature during extraction = 8.3°C (15°F).

Losses during 24 hour hold: less than 3% of extractable capacity

		Charging	Extraction
Thermal rates:	Max	255 MWt (870×10^6 Btu/hr)	285 MWt (923×10^6 Btu/hr)
	Min	12.5 MWt (43×10^6 Btu/hr)	31.1 MWt (106×10^6 Btu/hr)
Time at max rate:			6 hr
Hot standby demand, fluid at 318-310°C (600-585°F)			0.1 MWt (0.37×10^6 Btu/hr)
Nighttime seal steam, fluid at 232-149°C (450-300°F)			0.4 MWt (1.44×10^6 Btu/hr)

Table 3-3

100-MWe COMMERCIAL PLANT DESIGN REQUIREMENTS

Design life (with routine maintenance)	30 yr
Conditions for heat loss calculations	
• Ambient temperature	28°C (82.6°F)
• Wind speed at 10 m elevation	3.5 m/s (8 mph)
• Velocity profile, m/s (H in m)	$V_H = 3.5\,(H/10)^{0.15}$
Barstow soil bearing capacity	
• 0.07 MPa (1500 psf) at 0.61 m (2 ft) depth	
• 0.24 MPa (5000 psf) at 1.53 m (5 ft) depth	
• 0.48 MPa (10,000 psf) at 3.05 m (10 ft) depth	
(More soils data in ERDA letter of 14 January 1977)	
Barstow annual rainfall: 4 in. (100 mm)	

are presented in Table 3-4. The maximum heat storage unit temperature for the 100-MWe Commerical Plant was chosen to be 316°C (600°F). Additional requirements in the field of safety are contained in Reference 3-1.

3.2.2.2 Design Analyses

Briefly presented here are the multidisciplinary considerations which, in combination, produced the proposed TSU design described in 3.2.2.3. These considerations are treated individually at some length in Section 4.3.1 on the Pilot Plant TSU. They are (in order): dual-medium heat storage analysis; selection of dual heat-transfer media; dual-medium thermal design; heat losses and insulation; fluid distribution and bed packing; and structural analysis.

The basic design concepts were those which appear in Reference 3-2 and which were used in the construction of the TSU tested as a component of the SRE Thermal Storage System. Most of these concepts were validated in tests conducted by Rocketdyne during 1976. As a result of the tests and because of the availability of additional information, certain design concepts

Table 3-4

DESIGN CRITERIA FOR THE 100-MWe COMMERCIAL TSU

Cost effectiveness
Use of existing technology
Compatibility with all specified operating modes
Selection of optimum storage media
Structural design according to recognized vessel code(s)
Consideration of thermal expansion of tank and media
Consideration of thermocline nonideality in packed bed
Low pressure drop during fluid flow through bed
Consideration of fluid flow nonidealities in TSU
Inert gas ullage to prevent high-temperature oxidation of fluid
Consideration of heat losses to environment
Optimum use of insulation on tank walls

were modified. The design analyses presented here, therefore, contain material previously presented in Reference 3-2 and new material. These analyses represent the currently considered design rationale for the dual-medium TSS.

In this section, the detailed presentation of the design procedures and numerical results of Section 4.3.1, in some cases including the Commercial Plant, will be paraphrased briefly, where considerations unique to the Commercial Plant are concerned, they will be given fully. Section 4.3.1 should be consulted when additional details are desired.

The dual-medium sensible-heat storage concept, using a packed bed of particles and a suitable heat-transfer fluid, was adopted, as illustrated in Figure 3-3. The fluid fills the voids in the bed and flows through the bed to deposit or withdraw energy. Heat storage occurs in both the liquid filling the voids between the solid bed particles and in the solid bed particles themselves. Optimizations for maximum energy storage per unit bed volume and for minimum cost per unit energy stored were made by analysis and experiment (the latter on fluid degradation at high temperatures), to arrive

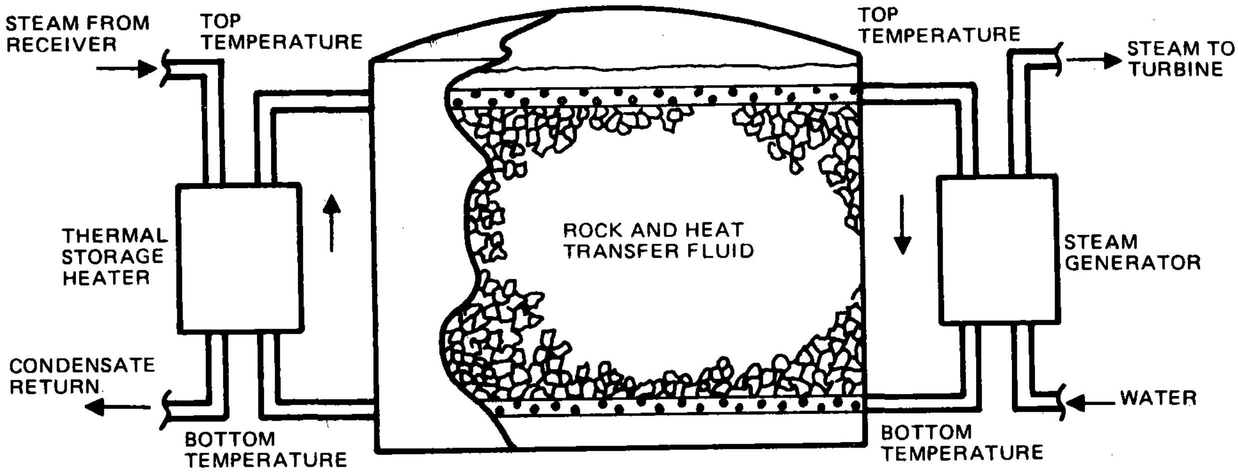

Figure 3-3. Dual-Medium Thermal Storage Concept

at a selection of fluid and solid bed materials. These were the high-temperature organic fluid Caloria HT43 and a rock mixture of 25 mm (1 in.) river gravel and 1.5 mm (1.16 in.) No. 6 silica sand.

The differential equations for energy storage in this system were modeled in 1974 by Rocketdyne in two computer programs: one a closed-form solution with constant properties and the other a more realistic variable mesh numerical integration method featuring variable properties and arbitrary boundary and initial conditions. The computer models were used to investigate the effect of such factors as variable bed dimensions and energy flow rates on performance. SRE tests of a well-instrumented TSU, built for these tests, validated the theory and developed realistic parameters for use in design of TSU's for larger capacities. In particular, during thermal charging or extraction, the thermoclines (horizontal interfaces within the bed separating the higher and lower temperature regions) were found to be of the sharpness, predicted by theory, required for efficient performance.

Structural integrity of the bed and its containing tank must be assured for the operating life of the power plant. The Barstow soil data show a maximum bearing capacity of 0.48 MPa (10,000 psf) at a depth of 3.05 m (10 ft). In the case of the Commercial Plant, it was decided to limit the bearing capacity to 0.38 MPa (8,000 psf) at a depth of 2.44 m (8 ft). For the bed with 0.25 void fraction, the effective density including fluid is 2.22 kg/liter (139 lb/ft^3), giving an allowable height of 17.5 m (57.4 ft). Considering the added weight of tank walls, roof, manifolds, etc, which are of the order of 1% of the bed weight, a bed height of 17.1 m (56 ft) was selected. The very large energy storage requirement for the Commercial Plant dictates multiple tanks to reduce wall thickness and for operating flexibility. Use of the methods in Section 4.3.1 and consideration of the thermal capacity requirements resulted in a four-tank design with a maximum wall thickness of 44.4 mm (1.75 in.) at the base of the tank. Each of the tanks has the same dimensions; the diameter is about 27.4 m (90 ft), and depends on the heat losses to be allowed.

The SRE TSU tests validated the design of the multiple-orifice manifolds through which fluid flowed into and out of the bed. Horizontal uniform flow and temperature profiles, operationally very desirable, were achieved with these manifolds. The tests also provided data on heat loss to the environment, an important factor in view of the 20-hr hold requirement at full charge prior to heat extraction at maximum rate. Proper insulation design satisfies this requirement. A TSU heat loss scaling curve derived from the SRE tests (Figure 4-16 in Section 4) indicates (at the heat storage capacity of 1,857 MWHt/4 = 464 MWHt for one tank) that the 24-hr heat loss is 1.8% of the extractable energy, within the 2% requirement. This 24-hr value may be taken for the 20-hr hold. With the previously designated bed height of 17.1m (56 ft) and using the thermal design method established from the SRE tests, each of the four tanks will be able to provide the required energy for extraction after a 20-hr full-charge hold with a bed diameter of 27.6 m (90.5 ft). During the 20-hr hold, the average drop in temperature of the contents of the bed is about 1°C (1.8°F), requiring the temperature of the charging liquid to be elevated by that amount over the desired fluid extraction temperature.

3.2.2.3 Design Description

The Commercial Plant TSU tankage has been designed to meet or exceed all structural and safety requirements of the Solar Central Receiver Power Plant, as well as the requirements set forth in Section 3.2.2.1. The dimensions and other characteristics of the selected tank design appear in Table 3-5 and Figure 3-4.

3.2.3 Ullage Maintenance Unit

3.2.3.1 Requirements

The purpose of these units is to provide a controlled-pressure, oxygen-free gas atmosphere above the fluid in each of the TSU's. There are four identical units, one for each TSU. It is necessary to have an oxygen-free gas above the heat-transfer-fluid surface to prevent fire hazards and long-term oxidation of the fluid. The ullage pressure must be controlled within a moderately narrow band to avoid underpressurizing or overpressuring the tank. When the fluid (Caloria HT43) is heated from 232 to 316°C (450 to 600°F), the volume expands approximately 9.5%. If the gas in the ullage space were not released, the pressure would rise above the allowable upper

Table 3-5 (Page 1 of 2)

DESCRIPTION OF DESIGN FOR 100-MWe COMMERCIAL PLANT TSU

TSU Configuration:

- Four cylindrical tanks, axis vertical, installed above ground, supported on dry soil of 0.383 MPa (8,000 psf) bearing strength by excavation to 2.44 m (8 ft) below grade

Tank Dimensions (heights measured at circumference):

• Inside diameter	27.6 m (90.5 ft)
• Overall height	18.3 m (60 ft)
• Packed bed height	17.1 m (56 ft)
• Free fluid surface height at 318°C (600°F)	17.7 m (58 ft)
• Effective height of top manifold	17.0 m (55.6 ft)
• Effective height of bottom manifold	0.305 m (1.0 ft)
• Tank crosssectional area	598 m^2 (6,432 ft^2)

Thermal Performance (all tanks):

• Extractable capacity after 20-hour hold time:		1,857 MWHt
• Design storage temperatures:	maximum	318°C (600°F)
	minimum extraction	310°C (585°F)
	minimum	232°C (450°F)

- Thermal rates: maximum charge 255 MWt, maximum extraction 285 MWt; minimum charge 12.5 MWt, minimum extraction 3.1. MWt
- Heat losses during 24-hour hold less than 2% of extractable capacity.

Solid Storage Medium:

- 25 mm (1 in.) river gravel and 1.5 mm (1/16 in.) No. 6 silica sand in 2:1 ratio, with 0.25 bed void fraction

• Superficial bed volume per tank	10,200 m^3 (360,225 ft^3)
• Weight of solids per tank	20,270 Mg (22,325 tons)
• Total weight of solids (all tanks)	81,100 Mg (89,300 tons)

Liquid Storage Medium:

- Caloria HT43 heat transfer fluid
- 2,208 m^3 (0.58x10^6 gallons) at 21.1°C (70°F) in one tank
- 8,830 m^3 (2.33x10^6 gallons) at 21.1°C (70°F) in all tanks
- Two manifolds, each with 19,300 holes of 3.1 mm (0.125 in.) diameter uniformly spaced over cross-section
- One seal steam manifold

Table 3-5 (Page 2 of 2)

DESCRIPTION OF DESIGN FOR 100-MWe COMMERCIAL PLANT TSU

Tank Structural Details: • Fabricated of A537 Class 2 Structural steel with field welded construction • Upward conical bottom plate 6.35 mm (0.25 in.) thick, 2% slope • Plate thickness for 1.83 m- (6 ft-) high shell courses varies from 44.5 mm (1.75 in.) at bottom to 6.35 mm (0.25 in.) at the top. • Roof is single skin with trusses, 1-in-12 pitch conical. • Roof and sides covered with 204 mm (8 in.) fiberglass blanket insulation with corrugated aluminum weather cover.
Interfaces (Flow penetrations): • Caloria HT43 piping per tank for primary thermal charging and extraction: top and bottom manifold, each 14-in pipe • Calòria HT43 for night-time seal steam, 2-in. pipe in sidewall of each tank at 1.2 m (4 ft) height. • Nitrogen gas for ullage gas blanket, 10-in. pipe into roof of each tank

limit. The gas released must be replaced, or stored and returned to the ullage space during the cooling cycle because an equal amount of gas is required to prevent the pressure from going below the allowable lower limit as the fluid cools and contracts.

These functions must be accomplished in as simple a manner as possible with minimum capital, maintenance, and operating costs. The design also must permit reliable operational control and afford a measure of redundancy.

The major specific requirements on the ullage maintenance units are as follows:

A. The gas in the ullage space must not react with the fluid. Therefore, oxygen and air must be excluded from the entire system and an inert gas such as nitrogen or carbon dioxide or a reducing gas must be substituted.

B. A control system with suitable backups must be provided to ensure that the pressure in the ullage space will remain within the gauge pressure range of zero (i.e., at the ambient absolute pressure)

SHELL COURSE SCHEDULE (ASTM A537 CLASS 2 STEEL)				
COURSE	HEIGHT, m (FT)		PLATE THICKNESS, mm (IN.)	
1 (BOTTOM)	1.83	(6)	44.4	(1.75)
2	1.83	(6)	39.6	(1.56)
3	1.83	(6)	35.0	(1.38)
4	1.83	(6)	30.2	(1.19)
5	1.83	(6)	25.4	(1.00)
6	1.83	(6)	20.6	(0.81)
7	1.83	(6)	16.0	(0.63)
8	1.83	(6)	11.2	(0.44)
9	1.83	(6)	6.35	(0.25)
10 (TOP)	1.83	(6)	6.35	(0.25)

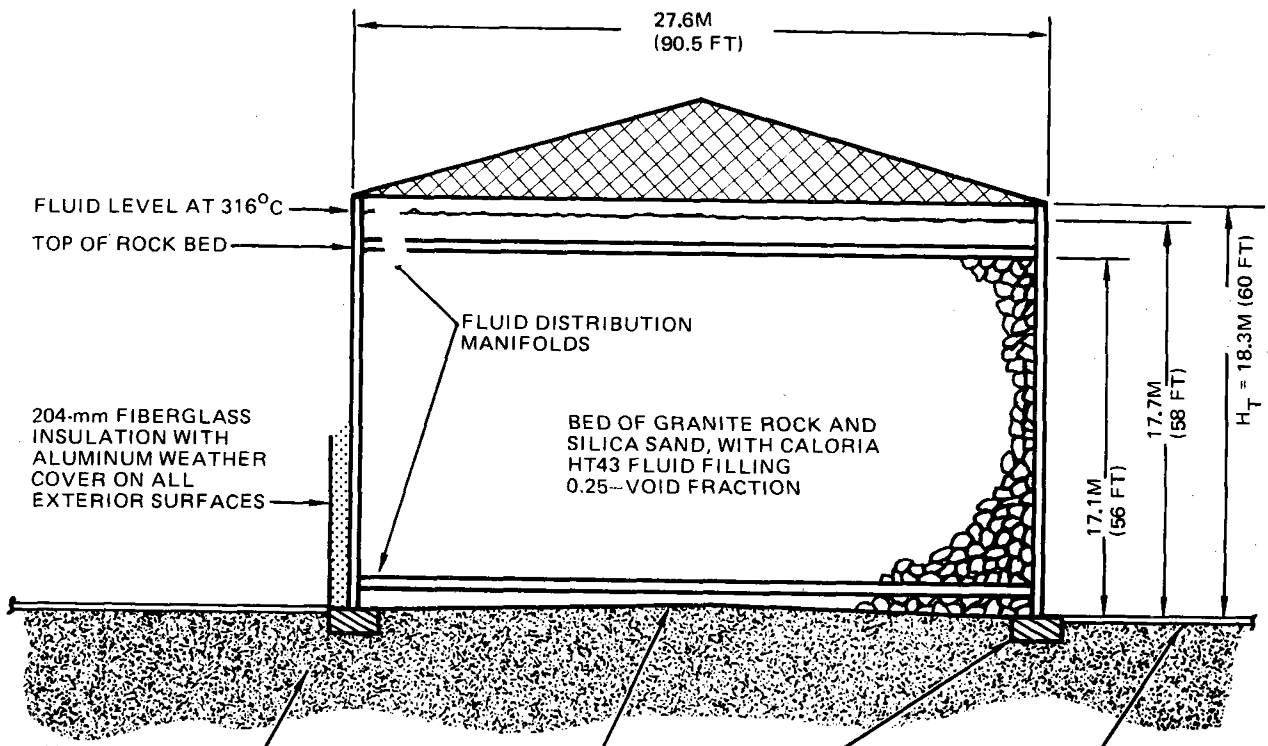

Figure 3-4. Design for 100-MWe Commercial Plant Thermal Storage Unit

to 2 KPa (0.3 psi). A pressure below ambient is not tolerable since this might permit air to enter the tank.

C. Since the emerging gases will be hot and will carry with them vapors given off by the fluid, they must be cooled and the condensate either returned to the tank or properly disposed of. Cooling the gas is required to accommodate storage of the gas in one form or another.

D. The products of thermal degradation of the fluid are mostly low-molecular-weight hydrocarbons (along with a small amount of high-molecular-weight substances). These volatile fractions will evaporate into the ullage space and must be disposed of without creating a pollution problem. These degradation products are produced at a rate dependent upon the time-temperature history of the fluid and must be removed as produced, either by stripping them from the ullage gases or by removal in the Fluid Maintenance Unit described in Section 3.2.4).

3.2.3.2 Analysis/Trades

There are many types of ullage maintenance units which can be used. The following alternatives were examined and evaluated:

A. Nonrecovered Gas Systems: (1) using GN_2, (2) using CO_2, and (3) using an inert gas generator.

B. Condensable Vapor Systems: (1) using water, (2) using a hydrocarbon such as heptane or hexane, and (3) using other compounds.

C. Recovered Gas Systems: (1) storage at TSU pressure (gas holder), and (2) storage of compressed gas.

Summaries of the description and evaluation of each of these units are given in Section 4.3.2.2.

The last type of unit listed (a recovered gas unit with compressed gas storage) was selected for the Commercial Plant design. It is shown conceptually in Figure 3-5 and is described in the following section (3.2.3.3).

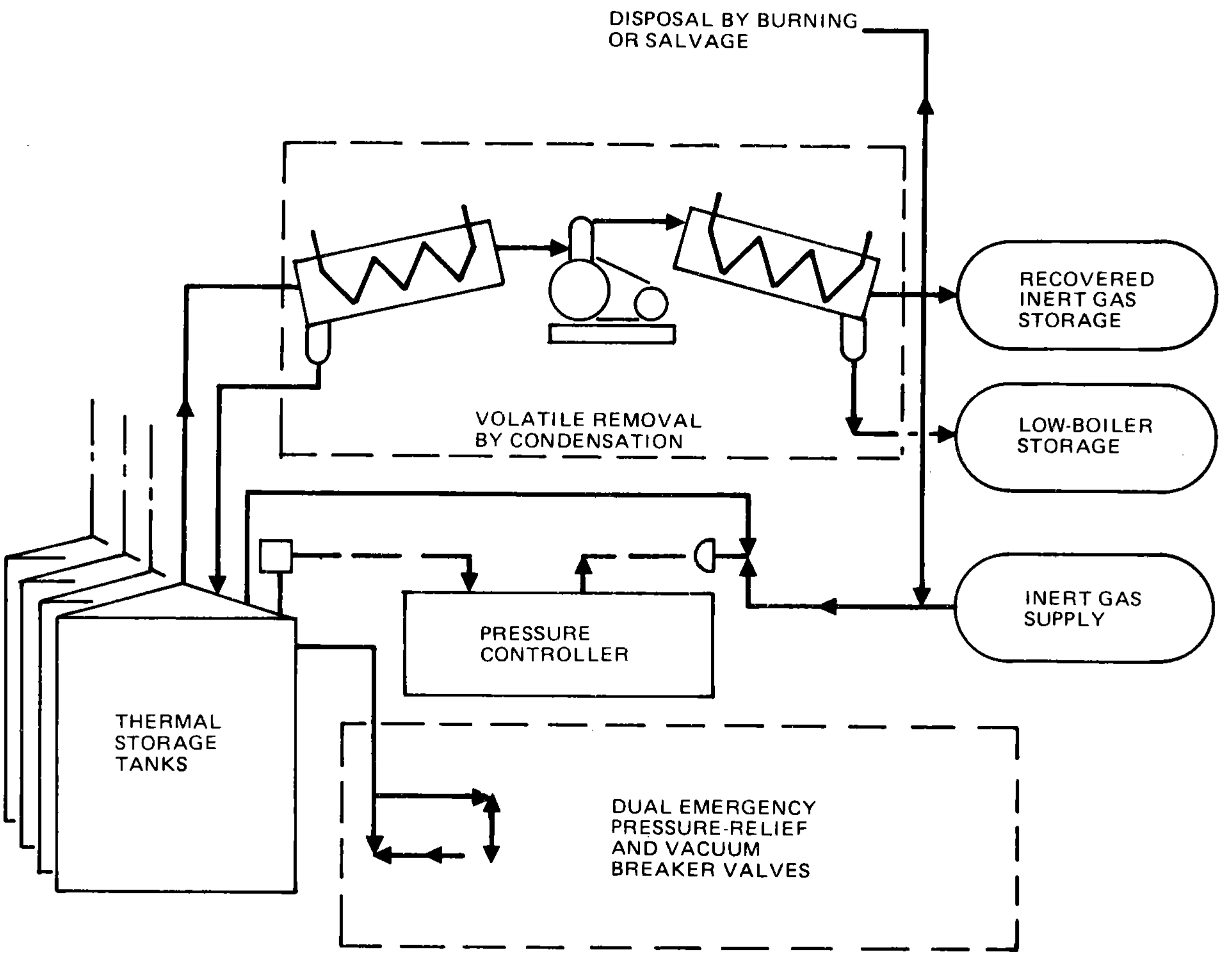

Figure 3-5. Conceptual Schematic Diagram of Ullage Maintenance Unit for 100-MWe Commerical Plant

3.2.3.3 Design Description

The ullage maintenance unit (UMU) for the 100-MWe Commercial Plant design is of a type which recovers the ullage gas, compresses it for storage, and reuses it for each daily storage cycle. A separate UMU will be provided for each of the four parallel TSU tanks. Each unit is identical to that designed for the Pilot Plant except for size. Each UMU was designed first to meet the requirements of one TSU tank and then further enlarged so that three UMU's can meet together, the full requirements for four TSU's. This permits one unit to be down for repairs without restricting full-capacity plant operation.

General Description

The description below is for one of the four identical units. The UMU is shown in conceptual form in Figure 3-5. The unit controls the pressure and vapor quality in the ullage space of the tank. Air is excluded since it will accelerate the heat-transfer-fluid degradation rate, and could, in extreme instances, create a hazardous condition. An inert ullage gas is added and vented as the fluid in the system expands and contracts during thermal cycling. Nitrogen was selected as the ullage gas for its economy and ready availability.

The ullage control unit features the following:

A. Ullage pressure automatically maintained at 10 cm of water above ambient atmospheric pressure.

B. A redundant system of emergency pressure and vacuum relief values to protect the tank in case the pressure-control system is unable to handle the necessary reduction or increase in gas flow into the ullage space.

C. Ullage gas storage at a working pressure of 1.2 MPa (175 psig) and with a gas volume of approximately 317 m^3 (11,600 ft^3) at 20°C and 1 atm, which is more than sufficient gas for one complete thermal charge of each TSU.

D. Measurement of inert gas flow into each TSU ullage space to allow monitoring and recording of system operation; TSU tank pressure and liquid level are also measured and recorded.

E. Performing a dual function: (1) of maintaining an adequate supply (and pressure) of ullage gas in the ullage space, and (2) of removing fluid degradation products from the ullage space. It is anticipated that the ullage gas compressor and subsequent storage tank will remove by condensation all of the volatile degradation products except, e.g., hydrogen, methane, ethane, etc, here defined as noncondensible (for purposes of this design). The anticipated rate of production of the noncondensibles will be great enough to require that they be vented either by flaring to the atmosphere or by disposal in some more appropriate energy saving manner. It is estimated that these noncondensible gases will be produced by the system at an average rate of 1,550 standard cubic meters per day (54,370 standard cubic feet per day).

Initially, the ullage space will be filled with gaseous nitrogen; however, as the ullage gas is reused many times, the concentration of nitrogen in the ullage gas will decrease steadily due to the fact that it is lost to the atmosphere along with the noncondensibles which are disposed of each day. This nitrogen gas is replaced by noncondensible vapors which are fed back into the ullage space during periods when the system is in a thermal discharging condition. Thus, eventually the ullage space in the TSU will be filled with noncondensible products of fluid degradation only and almost all of the nitrogen will have disappeared. An objection to using these gases in the ullage space is that they are flammable. However, adequate safeguards against their possible escape have been taken. Venting takes place only for burning in a makeup fluid heater or a remote flare tower in an emergency, and the entire system, except for the ullage gas storage tank, is kept at a low positive pressure of only a few centimeters of water column above the surrounding atmospheric pressure. Any leakage is, thus, bound to be outward and extremely slow if present at all. If rapid venting is required or if rapid addition of inert gas is required, these operations can be performed in a safe manner by the system, which has complete redundancy.

The oxygen content of the ullage gas is monitored continually and an alarm will be given should the oxygen level reach to within striking distance of the flammability limit.

It is very doubtful that oxygen diffusing into the system (provided such were possible against the internal pressure) will ever reach more than trace concentrations because the degradation products of the fluid will contain some molecules having unsaturated chemical bonds which will be reactive enough to slowly react with any trace amount of oxygen, thus immediately removing them and preventing their accumulation.

Operating Description (Normal Storage Cycle)

In the following description, only one of the four UMU's is described. It will be assumed that the TSS is in operation and has come to a standard cyclic condition. Beginning at the start of a heating cycle, the heat-transfer fluid expands and raises the pressure in the ullage space above the outside ambient pressure as indicated by the differential pressure measuring transducer and indicator DPI-18 (all components are referenced with numbers shown in Figure 3-6). The pressure controller uses this signal to turn on the compressor, C-80, provided this differential exceeds the first upper pressure limit. Gas and vapors are then pumped up through a simple air-cooled heat exchanger, HE-70, and down through a second simple air-cooled heat exchanger, HE-71 (which is also the compressor intake manifold) into the compressor inlet. The "exchangers" are just lengths of uninsulated pipe. The fluid vapors are quickly condensed in HE-70 and the condensate runs by gravity back into the TSU tank, TA-1. The remaining gas, consisting of the inert gas combined with low-boiler products of heat-transfer-fluid degradation, passes down through a vertical pipe labeled HE-71 which further cools the gas to a temperature close to ambient; any further condensation is collected at the bottom in a sump tank and is removed periodically by opening valve, HV-21. This is done when the liquid level reaches a given height as indicated by the sight-glass, SG-105, or signaled by the "high-level" liquid level probe (Table 3-6 gives a parts description).

The compressed gases are conducted into tank TA-2, which is tall and narrow, allowing the heat of compression to be lost easily to the surrounding air, and allowing those low boilers which will condense on the walls to collect at the bottom where they are periodically removed by opening solenoid valve HV-22. A sight-glass and liquid level probe indicate the liquid level. If

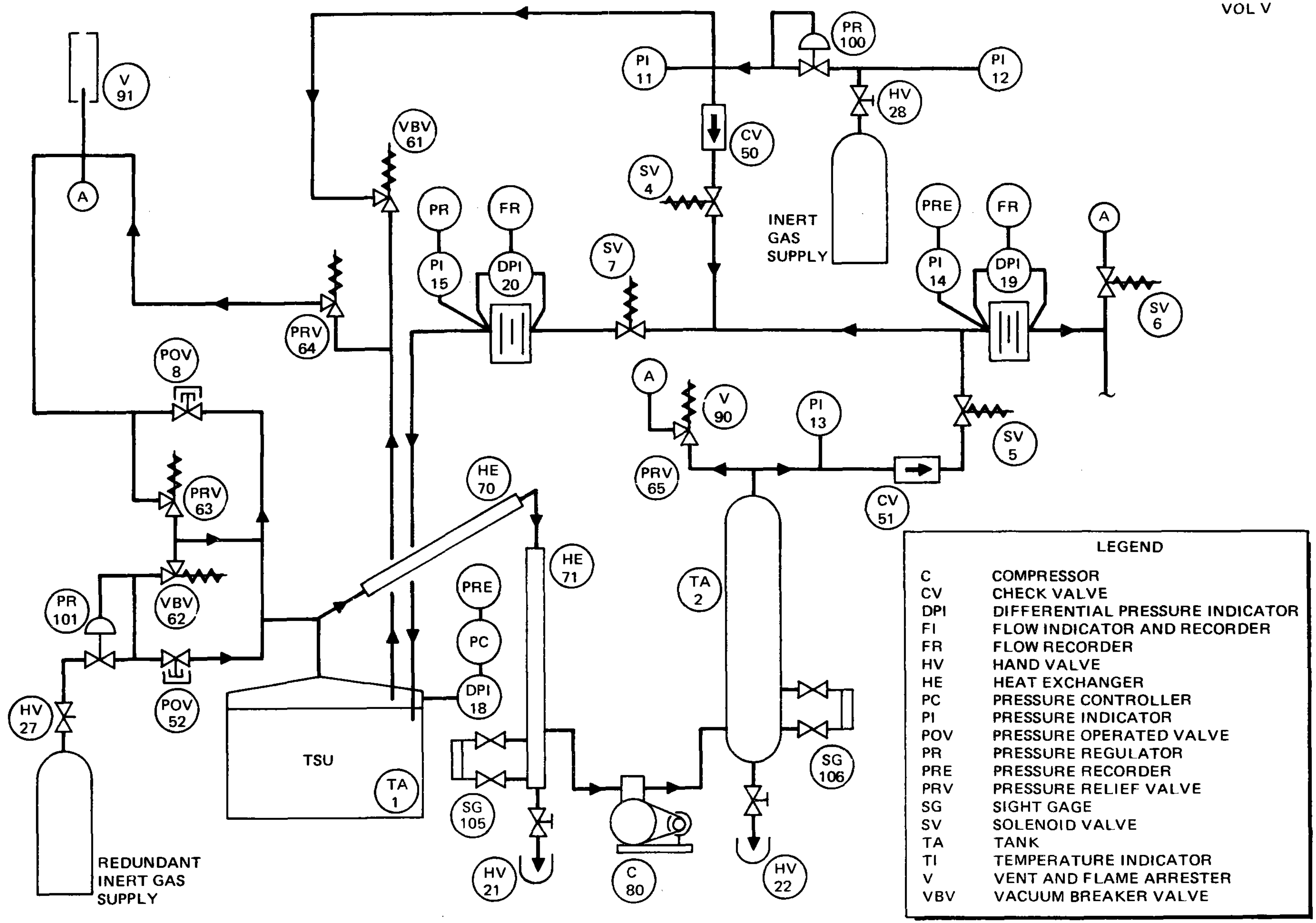

Figure 3-6. Ullage Maintenance Unit for 100-MWe Commercial Plant

Table 3-6 (Page 1 of 3)

PRELIMINARY COMPONENT LIST FOR COMMERCIAL PLANT UMU

Component Number	Designation (Figure 3-6)	Function	Range, Size, and Comments
1	TA1	Thermal storage	15.2m ID by 13.4m high cylindrical tank
2	TA2	Inert gas storage tank	15,100 liters (4,000 gal)
3	SV-4	Inert gas supply	3/4 in. NPT solenoid valve
4	SV-5	Inert gas storage	3/4 in. NPT solenoid valve
5	SV-6	Inert gas vent	3/4 in. NPT solenoid valve
6	SV-7	TSU-gas isolation valve	3/4 in. NPT solenoid valve
8	POV8	Automatic pressure relief valve	6-in. diameter with Bettis actuator and with 4-way solenoid valve
11	PI11	Pressure indicator, gas supply	0-1.38 MPa (0-200 psi) ordinary Bourdon pressure gage with long line
12	PI12	Pressure indicator, gas supply	0-34.5 MPa (0-5,000 psi) ordinary bourdon pressure gage with long line
13	PI13	Pressure indicator, gas storage	0-1.38 MPa (0-200 psi) ordinary Bourdon pressure gage with long line
14	PI14	Pressure indicator for flow measurement	Remote indicating and recording
15	PI15	Pressure indicator for flow measurement	Remote indicating and recording
18	DPI18	Differential pressure indicator	Recorder needed; the sensing line must be equipped with a very small inert gas bleed to provide a non-condensing temperature barrier

Table 3-6 (Page 2 of 3)

PRELIMINARY COMPONENT LIST FOR COMMERCIAL PLANT UMU

Component Number	Designation (Figure 3-6)	Function	Range, Size, and Comments
19	DPI19	Differential pressure indicator	With recorder
20	DPI20	Differential pressure indicator	With recorder
21	HV21	Condensate shutoff	0.86 MPa (125 psi) maximum working pressure
22	HV22	Condensate shutoff	1.38 MPa (200 psi) maximum working pressure
27	HV27	Inert gas supply shut-off	34.5 MPa (5,000 psi) maximum working pressure
28	HV28	Inert gas supply shut-off	34.5 MPa (5,000 psi) maximum working pressure
50	CV50	Check valve	Standard type
51	CV51	Check valve	Low release pressure
52	POV52	Automatic Pressurizing Valve	Identical to POV8
61	VBV61	Vacuum breaker valve	
62	VBV62	Vacuum breaker valve	
63	PRV63	Pressure relief valve	
64	PRV64	Pressure relief valve	
65	PRV65	Pressure relief valve	
70	HE70	Heat exchanger	To be constructed from 10-in. diameter, schedule 40 pipe

Table 3-6 (Page 3 of 3)

PRELIMINARY COMPONENT LIST FOR COMMERCIAL PLANT UMU

Component Number	Designation (Figure 3-6)	Function	Range, Size, and Comments
71	HE71	Heat exchanger	To be constructed from 10-in. diameter, schedule 40 pipe
80	C80	Compressor	11.2 Kw; 1.53 SCMM at 1.2 MPa (15 hp, 54 SCFM at 175 psi)
91	V91	Torch vent	To be constructed and wired to pilot ignite automatically
100	PR100	Pressure regulator	Standard unit used for pressure reduction on high-pressure gas bottles
101	PR101	Pressure regulator	Standard unit used for pressure reduction on high-pressure gas bottles
105	SG105	Sight glass	Standard sight glass 1 m (3 ft) high with isolation valves
106	SG106	Sight glass	Standard sight glass 1 m (3 ft) high with isolation valves

required, these valve operations can be automated easily; this decision will depend upon the relative rate of low-boiler production which will be determined from test data. The pressure in TA-2 will build up to 1.2 MPa (175 psig) by the end of the heating cycle; this is ensured by correctly sizing the tank. Should the pressure increase above this as indicated by pressure indicator PI-13, the excess will be released automatically (by the control system) through opening of valves SV-5 and SV-6. The quantity of gas thus released will be obtainable from the recording of the differential pressure indicator DPI-19. When the heating cycle ends, the pressure will drop below the first upper pressure limit of 11.4 cm (4.5 in.) of water column and the controller will stop the compressor.

When the cooling cycle begins, the contraction of the fluid will cause the pressure in the ullage space to drop below the first lower pressure limit which will cause the controller to automatically open valves SV-5 and SV-7, allowing gas in TA-2 to flow through the check valve, CV-51, and through the flow measuring orifice, DPI-20, into the TSU ullage space where it will bring the pressure back up to an acceptable value. The pressurized gas in tank TA-2 will thus be used up, reaching a level of about 0.14 MPa (20 psig) by the time the end of the cold cycle is reached. Should the pressure fall below this value at any time, new gas from the inert gas supply will be supplied automatically by opening SV-4 for as much time as is required.

However, if tank TA-2 is sufficiently large and if no gases are being vented, then new inert gas additions should not be necessary. A reserve of gas exists as liquefied gas at the bottom of the ullage storage tank in that as the pressure in the tank is reduced, this liquid will be revaporized. This liquefied gas is composed of some of the degradation products of the thermal storage fluid.

A new heating cycle can now begin and the above steps repeated.

Should any malfunction occur so that the ullage pressure might climb above the first upper pressure limit to reach the second upper pressure limit, then the controller would automatically open POV-8, causing the ullage gas to be vented through a vent stack equipped with a torch and ignition device. A torch is needed at the top of the vent since the cooling due to flow through the stack will reduce the temperature far enough to eliminate any chance of autoignition at the exit point. The flash point of Caloria HT43 is 204°C (400°F). Its autoignition temperature is 404°C (759°F). Should the pressure still increase slightly to a level above the third upper pressure limit, then the pressure relief valves, PRV-63 and PRV-64, will release and vent the ullage space through the same stack. These, and other pertinent ullage pressure levels, are summarized in Table 3-7.

The above redundancy, and automatic operational procedure, is duplicated for the first, second, and third lower pressure limits by the vacuum breaker valves, VBV-61 and VBV-62, and by the control system. A redundant inert

Table 3-7

SUMMARY OF TSU ULLAGE PRESSURE
LEVEL IMPLICATIONS AND SAFETY FEATURES

Meaning or Action Indication of Ullage Pressure Level	Pressure in TSU Ullage, cm (in.) of Water (Gage)
Tank Structure Design Yield Point	40 (16)
Tank Test Pressure	20 (8)
Higher High-Pressure Warning by Digital Data Logger	17.8 (7.0)
Pressure Relief Valve Setting	17.8 (7.0)
High-Pressure Switch Alarm	15.2 (6.0)
High-Pressure Warning by Digital Data Logger	14.0 (5.5)
Upper Limit of Nominal Operating Pressure	12.7 (5.0)
High-Pressure Relief Switch Closes	11.4 (4.5)
Nominal Pressure	10.2 (4.0)
Low-Pressure Supply Switch Closes	8.9 (3.5)
Lower Limit of Nominal Operating Pressure	7.6 (3.0)
Low-Pressure Warning by Digital Data Logger	6.4 (2.5)
Low-Pressure Switch Alarm	5.0 (2.0)
Vacuum Breaker Valve Relief Setting	2.5 (1.0)
Lower Low-Pressure Warning by Digital Data Logger	2.5 (1.0)

gas supply is provided for VBV-62 to ensure that enough inert gas is available to relieve any low-pressure condition by allowing the entrance of inert gas only.

There is an additional design consideration wherever the ullage maintenance system lines connect with the TSU. At these points vapors from the fluid will condense continually on the inside, colder wall of the pipe near the junction with the tank. These lines are thus constructed with a slight slope toward the tank to allow condensed fluid to return to the tank and to keep it from collecting at the other ends of these lines.

The compressor intake manifold sensors include an oxygen content monitor sensor and a moisture content monitor sensor. In addition, a liquid level switch is located just above the sight glass on the sump tank. A similar switch is placed just above the sight glass of the ullage gas storage tank, TA-2. (The above items are called out in Figure B-4 in Appendix B.)

Each of the four TSU ullage spaces are connected to a common point through a 0.254m (10-in.) manually operated valve and lengths of 0.254m (10-in.) pipe. The valve permits isolation should it be required. This valve is purposedly not remotely operable to prevent accidental closure. The common manifold provides extra assurance that no single ullage pressure at one tank can be very different from any other, and allows one compressor to be shut down with the inherent feature that the other compressors will take over the task of maintaining proper ullage conditions in all four TSU's.

3.2.4 Fluid Maintenance Unit

3.2.4.1 Requirements

The efficient, economic operation of the TSU requires that the large inventory of heat-transfer fluid be maintained in a satisfactory condition for continuous use. Pyrolysis, the natural result of exposing the fluids to elevated temperatures, produces low-molecular-weight species that are volatilized and removed in the UMU and high-molecular-weight polymeric material (high boilers) that accumulates in the fluid. For continuous, long-term use of the fluid, limitations must be placed on its high-boiler content to avoid fouling of heat-transfer surfaces. Investigations of fluid thermal stability, discussed in Section 6.2.2, resulted in the selection of Caloria HT43 for use in the Pilot Plant and the Commercial Plant, but have provided only very qualitative information on high-boiler production. The requirements for removal of polymerized materials, are thought to be conservative. Information gained from the Pilot Plant operation should provide assurances for the Commercial Plant. The fluid maintenance unit design will be required to:

A. Keep the level of polymerized material below 10%.

B. Filter suspended solids from the circulating fluid.

C. Add fresh fluid to make up for pyrolysis losses; maintain a constant fluid inventory.

D. Provide virtually automatic operation using a simple direct method requiring relatively inexpensive equipment.

E. Use existing state-of-the-art technology and, as far as practicable, use existing commercial components.

F. Use only a modest quantity of parasitic electrical power and thermal energy.

3.2.4.2 Design Analysis

The design analysis of a fluid maintenance unit must consider the best available information on the fraction of fluid lost per unit time as volatiles, the fraction that accumulates in the fluid as polymerized high-boiling material and residues, and the tolerable concentration of high boilers in a practical system.

Using a rate equation obtained for the weight loss of volatiles from Caloria HT43, Equation 4-1, and a Commercial Plant TSU temperature-time operating cycle given in Figure 6-13 in Section 6 (top temperature of 316°C (600°F) and a bottom temperature of 232°C (450°F)), the steady-state fluid weight loss for one year (330 cycles) was 12.9%.

The rate of high-boiler formation in the Commercial Plant has been estimated to be about 5% or less of the total weight loss by volatilization calculated from Equation 4-1. For a 9.08×10^6 l (2.4×10^6 gal) fluid inventory in the Commercial Plant, then, the fluid loss by volatilization would be 3,550 l/day (938 gal/day) and the polymer materials formation rate would be 177.6 l/day (46.9 gal/day). Assuming the waste stream from the fluid processing unit to be 50% polymeric residue, approximately 355 l/day (93.8 gal/day) of this concentrate must be removed to maintain a steady-state value of high boiler in the fluid.

The tolerable steady-state concentration of the polymer materials, or high-boilers, in the fluid has been arbitrarily set at a level of 10%. Thus a 177.6 l/day (46.9 gal/day) polymer materials production rate from the Commercial Plant would require that 1,776 l/day (469 gal/day) of fluid be processed. Since the Commercial Plant fluid maintenance unit will operate 24 hr/day, 74 l/h4 (19.5 gal/hr) of fluid must be processed to maintain a steady-state condition. Information gained from the additional fluid tests (Section 6.2.4) and the operation of the Pilot Plant should permit a more realistic upper limit to be placed on the allowable level of high boilers.

Although the Commercial Plant fluid maintenance unit must process about thirty times as much fluid as the Pilot Plant unit to control the high-boiler concentration, the method used in the Pilot Plant, i.e., vacuum distillation, can be scaled up. Alternative processes such as steam distillation, solvent extraction, and catalytic hydrocracking (to reclaim the high boilers), briefly discussed in Section 4.3.3.2, all have drawbacks.

3.2.4.3 Design Description

The fluid maintenance unit for the Commercial Plant is shown schematically in Figure 3-7. It performs three functions to keep the fluid in good operating condition: filtration to remove suspended solids, distillation of a side stream to remove high-boiling polymeric compounds, and the addition of fresh makeup fluid to the system. The unit is designed to use existing commercial components.

Filters

As in the Pilot Plant, 80 mesh filters that will remove solids above 177 μ (0.0177 cm), are located at the entrance to the thermal storage main flow line pumps. As discussed in Section 4.3.3.3, the TSU should behave as a filter bed and remove particles on the order of 16 μ (0.0016 cm) or larger from the fluid. Low-density particles formed from organic materials will be removed in the waste stream from the vacuum distillation unit along with the 50% concentrated high boiler.

Heat Transfer Fluid Processing

The Commercial Plant fluid-processing unit will be a scaled-up version of the mechanically aided thin-film vacuum evaporator and condenser that is proposed for the Pilot Plant. The unit will operate at the same pressure as the Pilot Plant unit, that is, about 133 Pa (0.0193 psia) to 67 Pa (0.0096 psia) and will function 24 hr/day. A brief description of the thin-film

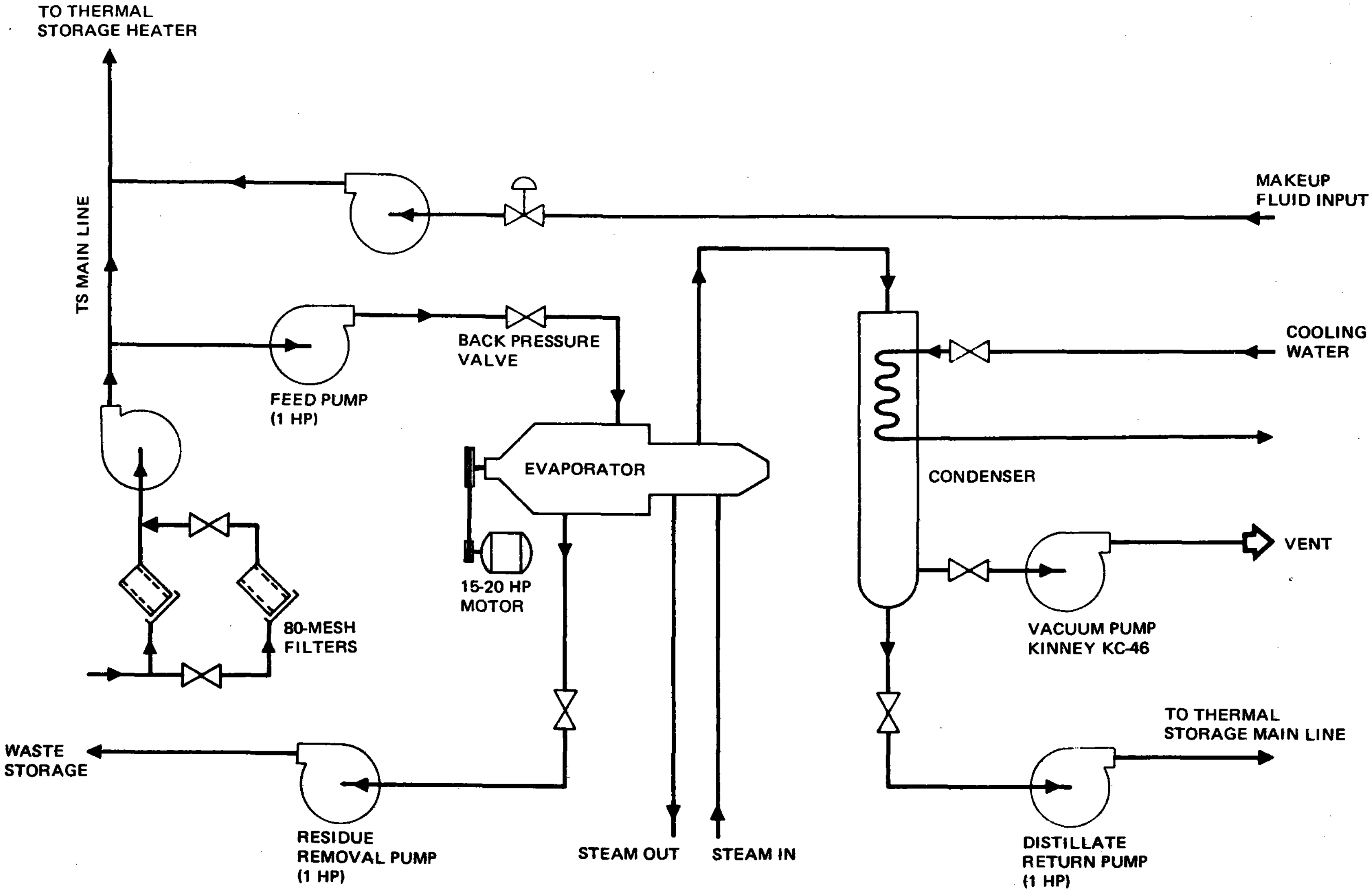

Figure 3-7. Fluid Maintenance Unit Design for the Commercial Plant

vacuum evaporator and condenser is given in Section 4.3.3.3. The scaled-up version of this unit will have a 2.79 m^2 (30 ft^2) evaporator, driven by a 14.9 to 11.2 KWe (20 to 15 hp) motor, and a 9.29 m^2 (100 ft^2) condenser.

Any heat required by the evapoarator can be supplied by recirculating steam or hot fluid through a heating jacket surrounding the evaporator. The total energy load for the evaporator for evaporation and heat losses is estimated to be 64.5 KW (220,000 Btu/hr). The specifications for the unit presently anticipated for the Commercial Plant are given in Table 3-8.

Table 3-8

CHARACTERISTICS OF FLUID MAINTENANCE UNIT MECHANICALLY AIDED THIN-FILM VACUUM EVAPORATOR SYSTEM FOR THE COMMERCIAL PLANT

Fluid Feedrate	74 l/hr (19.5 gal/hr)
Evaporator Area	2.79m^3 (30 ft^2)
Evaporator Motor	11.2-14.9 KWe (15-20 hp)
Condenser	9.29m^2 (100 ft^2)
Fluid Pumping Power (3 pumps)	3.72 KWe (5 hp)
Vacuum Pump	2.24 KWe (3 hp) 1m^3/min (35.3 ft^3/min)
Cooling Water	75.7-94.6 l/min (20-25 gal/min)
Total Power Requirement for Evaporative Load and External Heat Losses	64.5 KWe (220,000 Btu/hr)

Fluid Makeup

The fluid makeup rate required for the Commercial Plant will consist of the sum of (1) the fluid loss due to volatilization of very-low-molecular-weight species produced by thermal cracking, calculated from Equation 4-1 to be 3,550 l/day (938 gal/day), and (2) the residues removed from the vacuum evaporator, which are estimated to be 10% of the volatiles lost (5% high boiler at 50% concentration), or 355 l/day (93.8 gal/day). Thus, the estimated fluid replacement rate is about 3,900 l/day (1,030 gal/day). Assuming a thirty-day supply of fresh Caloria HT43 is kept in storage, the required tank size is 117,000 l (30,900 gal). The fluid loss will be replaced periodically (once a day or every several days) by pumping fresh fluid from the 117,000 l tank into the TSU main line to the heat exchangers.

3.2.5 Charging Loop

3.2.5.1 Desuperheater (DSH)

The function of the DSH is to reduce the temperature of incoming rated steam to a value acceptable to the thermal storage heater (TSH), 360°C (680°F). This temperature at the TSH is limited by the maximum working temperature of the fluid, 316°C (600°F). The DSH cools the incoming steam when being supplied to the TSS at rated conditions, 510°C (950°F). When the receiver is generating derated steam at 360°C (680°F), the DSH is inactive. Requirements are listed in Table 3-9. The DSH mixes water from the tower feedwater supply, 231°C (448°F), with the receiver steam to produce derated conditions.

Table 3-9
DESUPERHEATER REQUIREMENTS

Superheated Steam (inlet)	Pressure	10.1 MPa (1465 psia)
	Temperature	510°C (950°F)
	Flow rate	91.8 kg/sec (727×10^3 lb/hr)
Water (inlet)	Pressure	17.9 MPa (2600 psia)
	Temperature	218°C (425°F)
	Flowrate	20.0 kg/sec (158×10^3 lb/hr)
Steam (outlet)	Pressure	10.1 MPa (1465 psia)
	Temperature	360°C (680°F)
	Flowrate	111.8 kg/sec (885×10^3 lb/hr)

Turndown ratio: 17

Maintenance: minimum; must be accessible for cleaning and replacing spray nozzles

Life: 30 yr

The DSH is a commercial product made of standard pipe sections and represents state-of-the-art technology. Dimensions and operating characteristics are shown in Figure 3-8.

3.2.5.2 Thermal Storage Heater (TSH)

The function of the TSH is to transfer energy from receiver steam (available as a result of either high or low insolation) to the thermal storage fluid. The TSH must be responsive to constantly changing input steam conditions while maintaining constant fluid outlet temperature.

Several TSHs will be used in parallel for the Commercial Plant, determined by operating flexibility, reliability, and manufacturing constraints. Preliminary design estimates place the number at a minimum of five exchangers. Specific design requirements of the Commercial Plant for five TSH's are listed in Table 3-10. A range of from five to ten was investigated with regard to annual cost. Beyond the cost analysis of the TSH, evaluation of the structural life and reliability under operating conditions were performed. Effects of fluid velocity, temperature difference, and interstream leakage were investigated.

Preliminary design choices of the TSH were narrowed down to a U-tube, shell baffled heat exchanger. Free expansion of the tubes and shell with the minimum of stress was the primary reason for this choice. Reduction of any thermal induced stresses is an important factor when considering the life of a heat exchanger. In addition, removable tube bundles allow easy access to the tubes for periodic maintenance and inspection.

Many preliminary design and tradeoff studies were made for the Commercial Plant TSH's with earlier, slightly different requirements than those shown in Table 3-10. The same general trends were realized for heat-exchanger characteristics (i.e., length/diameter ratio, tube OD, pitch, etc) as for the Pilot Plant. The results of this optimization are listed in Table 3-11. In addition, services were obtained from an estimated heat-transfer consultant firm (Reference 3-3) and design specifications from heat-exchanger manufacturers were obtained.

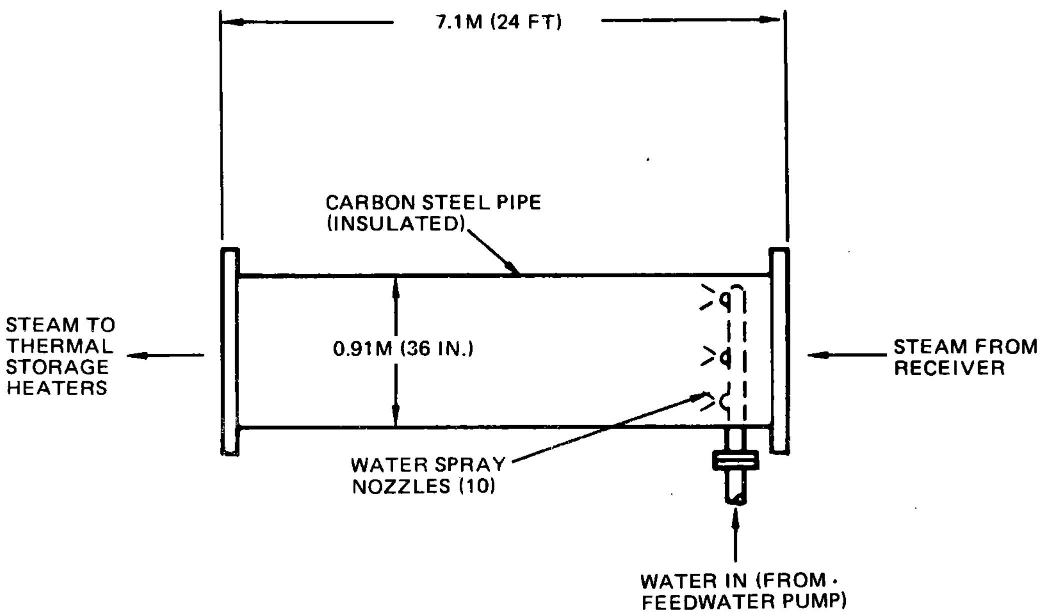

Figure 3-8. Desuperheater for 100-MWe Commercial Plant Thermal Storage Subsystem

Table 3-10

100-MWe COMMERCIAL PLANT TSH DESIGN REQUIREMENTS

Number of Exchangers Required	5
Following Information for Each of Five Parallel Exchangers;	
Duty	51 MWt
Tube Side Conditions (Steam/Water):	
Inlet	360°C (680°F), 10.1 MPa (1465 psia)
Outlet	249°C (480°F), 9.6 MPa (1400 psia)
Minimum Flow	23,980 kg/hr (52,745 lb/hr)
Maximum Flow	97,800 kg/hr (215,200 lb/hr)
Fouling Factor	0.000174 $\frac{°C\text{-}m}{W}$ $\left(0.0003\ \frac{HR\text{-}FT^2\text{-}°F}{BTU}\right)$
Shell Side Conditions (Caloria HT43):	
Inlet	232°C (450°F), 414 KPa (60 psia)
Outlet	316°C (600°F), 241 KPa (35 psia)
Minimum Flow	194,900 kg/hr (428,700 lb/hr)
Maximum Flow	794,400 kg/hr (1,747,000 lb/hr)
Fouling Factor	0.00058 $\frac{°C\text{-}m}{W}$ $\left(0.0010\ \frac{HR\text{-}FT^2\text{-}°F}{BTU}\right)$
Service:	The system operates on a daily cycle starting in the morning and ending in the evening. The duration of the daily operation varies between 8 and 10 hr/day.
Life Expectancy:	30 yr
Maintenance:	Minimum maintenance or refurbishment
Other:	Must meet highway transportation limits.

The first item considered was the possibility of interstream leakage. Any heat-exchanger leaks will be normally from the high-pressure steam to the fluid side. Due to the cyclic activity of the heat exchanger, however, there will be many instances where there will be a vacuum on the steam side as a result of condensation. The portion most susceptible to leakage is where

Table 3-11

RESULTS OF HEAT-EXCHANGER OPTIMIZATION STUDIES*

Number of Exchangers	5
Length/Diameter Ratio	10
Tube OD	1.9 cm (0.75 in.)
Tube Pitch (square)	3.18 cm (1.25 in.)
Baffle Spacing	76 cm (30 in.)
Overall Coefficient	1,135w/m^2-°C (200 Btu/hr-ft-°F)
Heat Transfer Area (Based on Tube Mean Area)	1,450m^2 (15,500 ft^2)

*Guidelines for evaluation purposes.

there is a junction between the tubes and the tube sheet. It has been suggested that a double-tube-sheet design be employed. The use of double-tube-sheet designs are common in many shipboard condensors and coolers. The extra cost of the double-tube sheet is in the range of 5 to 15% depending on the size and number of tubes. If a leak occurs on either the steam- or fluid-tube sheet the fluid enters into a special ringed compartment around the space between the two tube sheets. The leak may be easily detected and the unit can be taken off in time. In addition, an Inconel overlay on the tube sheet can provide extra protection from both corrosion and erosion. The shell of the heat exchanger should be equipped with high-pressure relief valves in case of tube failure.

In most cases, U-tube heat exchangers will normally have baffles attached to the tube bundle in an effort to enhance the heat-transfer process by increasing fluid velocity through mixed cross and parallel flow paths. Close baffle spacing increases both the heat-transfer fluid film coefficient and the pressure drop, while it decreases the heat-transfer area. The velocity must be limited, however, due to vibration of the tubes occurring at high velocities. Vibration causes an unusually high stress on and increases the erosion of the tubes. To increase the life of the heat exchanger, moderate velocities are generally recommended (i.e., less than 10 fps).

Carbon steel is a very suitable material of construction which is compatible with both steam and Caloria HT43. The use of seamless tubes is normally prescribed at these conditions. The 30-year service life may be somewhat stretching the capability of the material, but it is sufficiently inexpensive to made replacement bundles a viable economic choice. There is a process now whereby fins can be rolled readily and economically onto the exterior of a tube. This increases the smooth tube area by a factor of three for the same tube length. The use of finned tubes has been recommended by some manufacturers, but the net effects on heat-exchanger performance and size are not yet known.

The pitch of the tubes is another variable which is considered. Although triangular pitch is the most compact, reduces the shell diameter, and provides better heat transfer, it permits no mechanical cleaning except for the exterior tubes. A square pitch arrangement allows free access to all the tubes but requires a somewhat larger shell. No coking of the HT-43 fluid is anticipated based on SRE test results but if it should occur, mechanical cleaning would be a necessity. If, however, Pilot Plant operation confirms that no coking occurs, then the triangular pitch arrangement can be used, resulting in about a 12% reduction in shell diameter.

Drainage of both the water and fluid from the heat exchanger was considered. The tubes will be inclined for drainage when in a horizontal tube orientation. This really has no effect on heat-exchanger performance as any accumulated water will be blown out by the steam. The only requirement for draining the fluid occurs when maintenance is required. Most heat exchangers have provisions for removal of the fluid as well.

Three different configurations of a two-shell pass U-tube heat exchanger were investigated with regard to advantages and shortcomings for each unit. The discussion and associated figures for this comparison appear in Section 4.3.5 and are not repeated here.

Investigation of TSH multiplicity was also conducted in an effort to reduce parasitic power consumption during reduced flow conditions. By taking exchangers out of the system when not required, a savings in power cost can be achieved. This savings can only be realized, however, if there are equal numbers of pumps and TSH's. If this relationship does not exist, then either the number of TSH's are irrelevant or the pumps must be throttled. Analysis in a subsequent section (Section 3.2.5.3) has shown that no significant reduction in power cost occurs beyond the addition of three to four pumps. An increase of five to ten TSH's, therefore, has no real advantage. Five was closer, primarily on the basis of size and availability. It should be noted that because of the multiplicity of heat-exchanger pump units (i.e., five units on the charging side) and the unusual duty cycle of the solar thermal power plant, major refurbishment of single units will result in little or no loss of overall electrical energy production. The charging side of the plant operates only at peak capacity for a relatively small proportion of the year. Clean out or repairs of major components (i.e., heat exchangers, pumps) can be scheduled for periods of the year when maximum capability is not required and 80% of maximum capability is adequate to store all of the available energy.

Table 3-12 lists designs by three representative heat-exchanger manufacturers who responded to RFQ's sent out by Rocketdyne to twenty different companies. It also lists key parameters necessary to make an evaluation of their designs. The preliminary design on which the bids were sought was a result of the analysis and optimization performed at Rocketdyne and discussed in the foregoing section. The maximum overall size of the heat exchanger; length, 12.8m (42 ft); shell OD, 2.3m (7.5 ft); the tube OD, 1.9 cm (0.75 in.); the pressure drop, 172 KPa (25 psi); and a TEMA classification of "R" were specified by Rocketdyne for a 51-MWt heat rate. The vendors were free to determine the remaining information to incorporate into their designs.

It should be noted that the overall heat-transfer coefficient, U, for Case 1, reflects an adjustment to account for finned tubes (the other designs use smooth tubes). In addition, Case 1 requires two heat ecshangers to perform the same duty as the others do with one. The value of U for Case 1, based on the equivalent smooth tube surface area, gives a value (in parentheses) which agrees with Case 3. The U value for Case 3 may be higher due to the greater fluid-pressure drop while that of Case 2 still deviates somewhat

Table 3-12

SUMMARY OF TSH DESIGNS FROM EXCHANGER MANUFACTURERS

Case (each a different manufacturer)	1***	2	3
Shell Diameter, in.	42	83	47
Exchanger Length, ft	42	42	40
Heat Transfer Area, ft^2	29,400 (12,800)**	35,000	18,000
Overall Heat-Transfer Coefficient, Btu/hr-ft^2-°F	68.5 (157)**	108	164
Weight, lb	75,000	--	100,000
Pressure Drop (fluid side), psi	20	--	25
Relative cost	1.72	1.00	1.05

*Values are given for comparative purposes; therefore, are not translated to SI units.

**Basic design has finned tubes; numbers in parentheses are equivalent values based on smooth tube areas.

***Two heat exchangers in parallel are required; information is for one, except the cost which is for both.

from the other two. This accounts for the area being larger in Case 2 than in the others. It should be noted that heat-exchanger companies usually provide extra tubes so that a major overhaul would not be required in the event of tube failure. The areas listed here may include provisions for extra tubes. Also, some companies include the U-bend portion of the tube when computing area while others do not. These differences in area, however, are to large to be attributable to these causes and are a result of the variance in the value of U computed by each.

The overall external dimensions of the heat exchangers reflected the heat transfer areas for each exchanger. For instance the diameter of Case 2 is almost double that of Case 3 with the same relationship existing for the areas. The lengths for all three heat exchangers were about the same.

Cost is another important criterion. As expected, there is a wide range. Surprisingly, the heat exchanger with the largest area (Case 2) has the lowest cost. The first exchanger (Case 1) which is actually two in parallel is probably more expensive due to having two shells instead of one and because of the finned tubes. The cost of Case 3 is just slightly higher than that of Case 2, yet Case 3 has the least area, but this cost seems reasonable from the standpoint of weight when compared to Case 1.

The design specifications submitted by manufacturers covered a considerable range of sizes, costs, weights, etc; this is to be expected when bids are sought from a competing marketplace. As previously mentioned, each manufacturer has his own technique for determining heat-exchanger designs and cost, and they will vary. Considerable analyses was made, both before and after receiving the manufacturer's design, to evaluate the validity of these designs and costs.

The final heat-exchanger design was developed from consideration of: (1) Rocketdyne designs and design calculations made for this application during the past two years, (2) the designs submitted by exchanger manufacturers, and (3) calculations and design inputs of the heat-transfer consultant firm retained by Rocketdyne.

The design for the Commercial Plant TSH is summarized in Table 3-13. Five exchangers in parallel will be required for the Commercial Plant. Each is a TEMA type "DFU" removable bundle, two-shell pass, U-tube heat exchanger. The thermal storage fluid, Exxon's Caloria HT43, is on the shell side, steam/water is in the tubes.

The U-tubes are in the horizontal plane and there is a longitudinal baffle which is permanently attached to the shell. In addition, there is a double tube sheet and an Inconel overlay on the tube sheet to prevent tube sheet erosion in the event that any leakage should occur. There are six (unequal) tube passes, with a liquid water level maintained in the subcooling region. A rotated square pitch will also be used to facilitate any mechanical cleaning which may be required. Conventional carbon steel will be used for all materials of construction. The heat exchangers will be thoroughly insulated.

Table 3-13

100-MWt COMMERCIAL PLANT THERMAL STORAGE HEATER DESIGN DESCRIPTION

Flow Configuration	Caloria HT-43, Shell Side: Steam, Tube Side
Number of Units	5
Duty per Unit	51 MWt
HTF Inlet/Exit Temp	232/316°C (450/600°F)
HTF Flowrate/Unit	220 kg/s (486 lb/s)
HTF Pressure Loss	0.17 MPa (25 psi)
Water Inlet/Exit Temp	360/249°C (680/480°F)
Water Flowrate/Unit	27.3 kg/s (60.0 lb/s)
Number of Tubes/Unit	1200
Tube OD	1.57 cm (0.62 in.)
Tube OD	19 mm (0.75 in.)
Tube Length (Average)	21.8 m (72 ft)
Heat-Transfer Area Per Unit	1670 m^2 (18,000 ft^2)
Shell Diameter	1.19 m (47 in.)
Exchanger Length	12.2 m (40 ft)
Pitch Type	Rotated Square Pitch
Pitch/Diameter Ratio	1.25
Weight Per Unit	45,000 kg (100,000 lb)
Design Pressure	
• Shell	0.35 MPa (50 psi)
• Tubes	10.3 MPa (1,500 psi)

3.2.5.3 Fluid Pumps and Piping

The purpose of the fluid pumps and piping for the Commercial Plant TSS charging loop is to provide a means to transfer heat from the TSH's into the TSU's. Conceptual design of this charging loop emphasized the following points:

- Capable of operating over a wide range of heat input
- Minimization of parastic power consumption
- Least possible capital cost
- Highest reliability

Not all of these conditions can be satisfied, but an optimum does exist between these points. Cyclic flow conditions will prevail, thus, pumping requirements will be changing constantly, which also dictates the amount of parasitic power consumption.

A typical daily duty cycle for the Commercial Plant charging loop, has the same overall shape as for the Pilot Plant (Figure 4-34 in Section 4), although much greater in magnitude. Table 3-14 summarizes the annual duty cycle for the 100-MWe Commercial Plant TSS charging loop.

Piping loop and cost data (details following those given in Section 4.3.6) were combined in a cost optimization computer program with the Commercial Plant charging loop conditions. The pipe and piping component requirements were estimated on the basis of preliminary design layouts. The impact on the annual operating cost of the fluid velocity, the number of pumps, and the electricity cost were investigated. Each were investigated independently by holding all but one of the variables constant. The annual cost was computed on the basis of amortization of the initial capital cost and an annual power cost. An amortization rate of 18% was used. The program responded to the annual duty cycle by choosing the appropriate number of pumps for a particular flow rate. A range of from one to ten pumps was investigated. However, the primary range of interest is between 5 and 10 pumps, to provide high operating flexibility and reliability; and to keep individual pump size within current commercial sizes. In addition, the effect of fluid velocity and the impact of projected electricity cost increases were investigated.

Table 3-14

100-MWe COMMERCIAL PLANT ANNUAL CHARGING DUTY CYCLE

Fraction of Maximum Flow	Hr Per Yr
0.050	19.8
0.068	111.0
0.100	43.5
0.137	126.8
0.150	23.7
0.200	27.7
0.206	103.1
0.250	14.0
0.275	198.3
0.300	29.8
0.344	206.2
0.350	19.8
0.400	33.8
0.413	290.2
0.450	7.9
0.482	399.7
0.500	31.6
0.550	27.7
0.551	339.1
0.600	19.8
0.620	351.6
0.650	31.6
0.689	566.1
0.700	59.6
0.750	14.0
0.758	289.3
0.800	13.7
0.850	23.7
0.900	7.9
0.950	7.3
1.000	0.6

Parametric results are shown in Figures 3-9 and 3-10. Effects of velocity on annual cost are shown for different numbers of pumps. The optimum velocity occurs at about 3.7 m/s (12 ft/s) for an electrical power value of \$0.02/KWHe, and decreases as the electrical power value increases.

Beyond the addition of the first few pumps, little savings in power cost are realized, as shown by Figure 3-10, which shows the effects on annual cost of the number of pumps for different fluid velocities. The nature of the duty cycle determines the shape of this curve and, as shown in Table 3-14, the charging loop flowrates are distributed somewhat evenly throughout the year with the exception of the extremities. This accounts for the little impact that the number of pumps has on the power cost, since little throttling will be required beyond the first few pumps. Based on these results, the following design was developed. A velocity of 3.9 m/s (13 ft/s) was chosen, resulting in a pipe size of 26-in., schedule 40 for main lines and 14-in., schedule 40 pipe for all segmented flow spools. It was decided to use five identical pumps in parallel, with a maximum capacity of $0.379 m^3/s$ (6000 gpm) each and $1.895 m^3/s$ (30,000 gpm) combined total. The pump drivers are single-speed electrical motors. Other design characteristics are given in Table 3-15.

3.2.6 Extraction Loop

3.2.6.1 Steam Generator

The purpose of the thermal storage steam generator equipment is to provide the turbine with a steady-state source of steam in order to maintain continuous turbine operation during low solar power, intermittent cloud, or night-time periods. As a result, it must be designed to produce steam which is compatible with the turbine admission steam requirements while being responsive enough to pick up load during periods of decaying receiver steam produced by cloudy or low-sun-angle conditions. The steam generator provides a natural source of system sealing steam which is required during nightly shutdown periods, and also serves as a source of morning turbine startup steam prior to the availability of receiver steam.

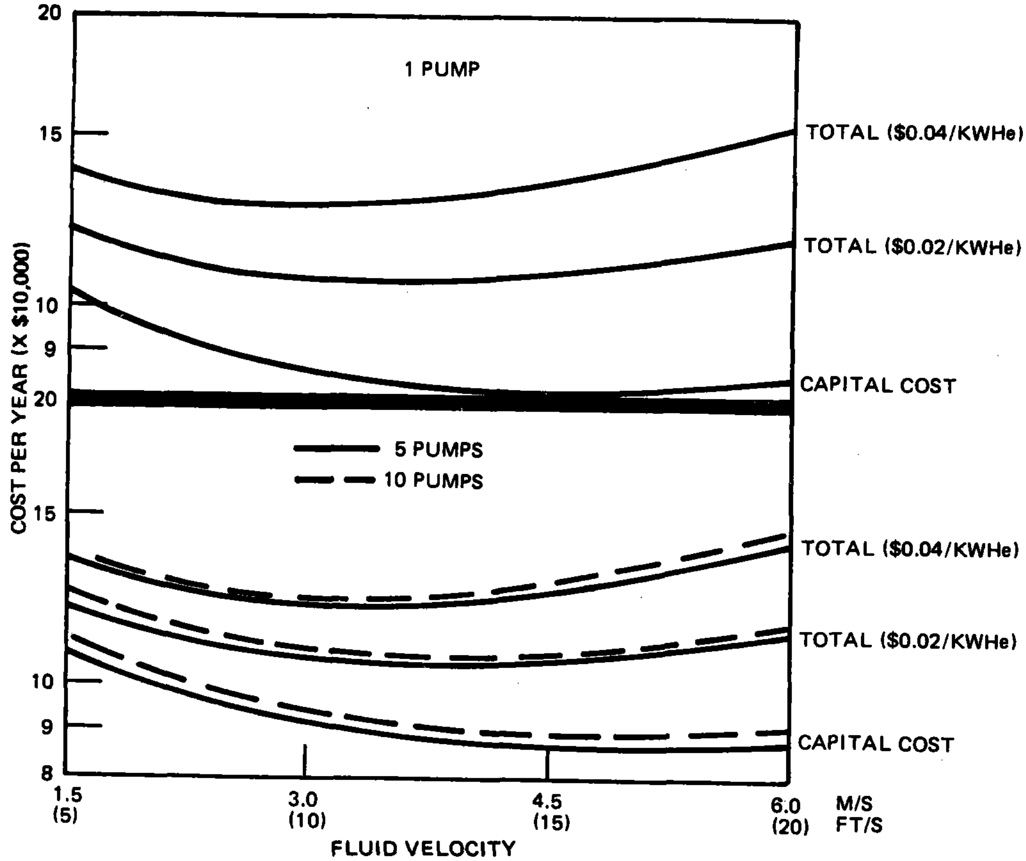

Figure 3-9. Annual Cost Optimization with Fluid Velocity, Number of Pumps, and Electricity Cost

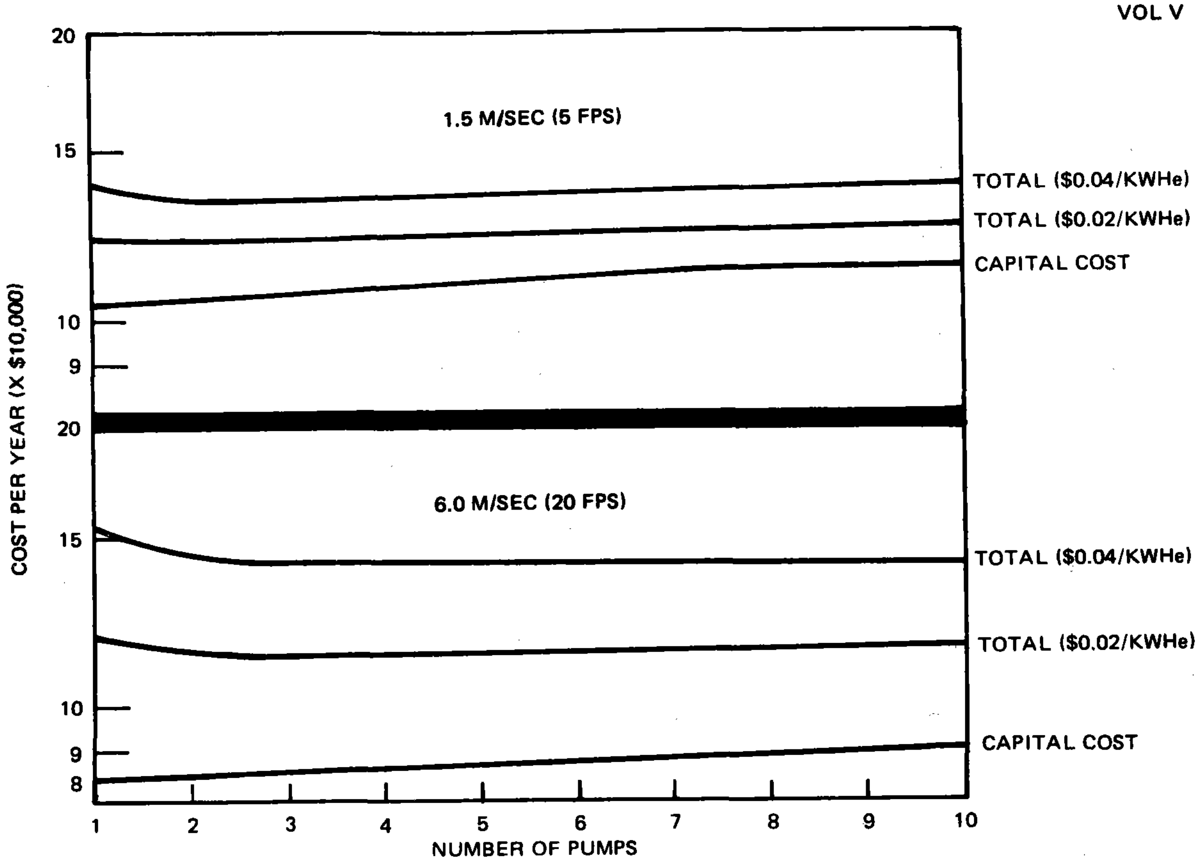

Figure 3-10. Annual Cost Optimization with Number of Pumps, Fluid Velocity, and Electricity Cost

Table 3-15

COMMERCIAL PLANT CHARGING LOOP DESIGN

Item	Quantity	Description
Pumps	5	Dean Brothers Model Number R484-8 x 10 x 15-1/2 in.
Motors	5	Single Speed, Induction Motor, 260*** Hp, 4,160 Volt, 3-Phase
Pipe	61m (200 ft) of 26-in., Schedule 40**	
	122m (400 ft) of 14-in., Schedule 40*	

*14-in. Schedule 40 pipe used for all segmented flow
**USA Standard pipe sizes given (no SI equivalent pipe sizes)
***Required motor input power; not motor capacity

The specific requirements to be satisfied by the steam generator equipment are shown in Table 3-16. The discharge rates and corresponding flow rates listed range from a maximum value, which is required by the turbine to produce 76.1-MWe gross power (70-MWe net) to a minimum value, which represents a reasonable lower limit to maintain controlled steam conditions for turbine startup or two-port turbine operation. The inlet feedwater and outlet steam conditions are defined in such a way as to satisfy admission steam requirements at the turbine while being compatible with pinch-point constraints associated with the TSS (Figure 3-11). The horizontal distance between the diagonal dashed line and the discharge flow line represents the idealized temperature potential for heat transfer on a local basis. It is seen that the two principal constraints (i.e., pinch points) occur at the onset of the boiling process, near the bottom of the figure, and at the superheated steam exit condition, at the top of the figure. It can also be seen that a more severe pinch-point condition would occur if the feedwater temperature or steam pressure were increased. Also, the 299°C (570°F) superheated steam condition represents the highest practical exit steam temperature from the superheat section. The hot standby demand requirement shown in Table 3-16 represents a recognition of the need for maximum thermal responsiveness during initial demand periods while the conditions for nighttime

Table 3-16

REQUIREMENTS FOR STEAM GENERATOR

Discharge Rate	
Max	285 MWt (0.923 x 10^9 Btu/Hr)
Min	31.1 MWt (106 x 10^6 Btu/Hr)
Feedwater/Steam Flow Rate	
Max	114.3 Kg/Sec (906 x 10^3 Lb/Hr)
Min	12.6 Kg/Sec (100 x 10^3 Lb/Hr)
Feedwater Inlet Conditions	
Temperature	121°C (250°F)
Pressure	2.90 MPz (420 psia)
Outlet Steam Conditions	
Temperature	299°C (570°F)
Pressure	2.72 MPa (395 psia)
Hot Standby Demand	
(Maintain Preheated Equipment)	0.1095 MWt (0.374 x 10^6 Btu/Hr)
Nighttime Seal Steam	0.423 MWt (1.44 x 10^6 Btu/Hr) For 12 hr with steam at 135°C (275°F)

sealing steam are consistent with requirements of the turbine and the feedwater heaters.

In selecting the generic type of steam generator equipment as the baseline for the commercial system, three general approaches were considered; single-pass-to-superheat, recirculating drum type, and multiple element kettle boiler concepts. The single-pass-to-superheat concept offers some economic advantage over either of the other two concepts. High-level estimates indicate that the single-pass steam generating hardware could be as much as 25 to 30% less expensive than equivalent duty drum or kettle-type concepts. For commercial-sized equipment, this could translate into a differential cost of $300,000 to $400,000.

The principal drawback to the single-pass approach for the steam generator is the severe operating envelope imposed by the rest of the solar electric system. The extremely wide range of steam flow rates required, the almost

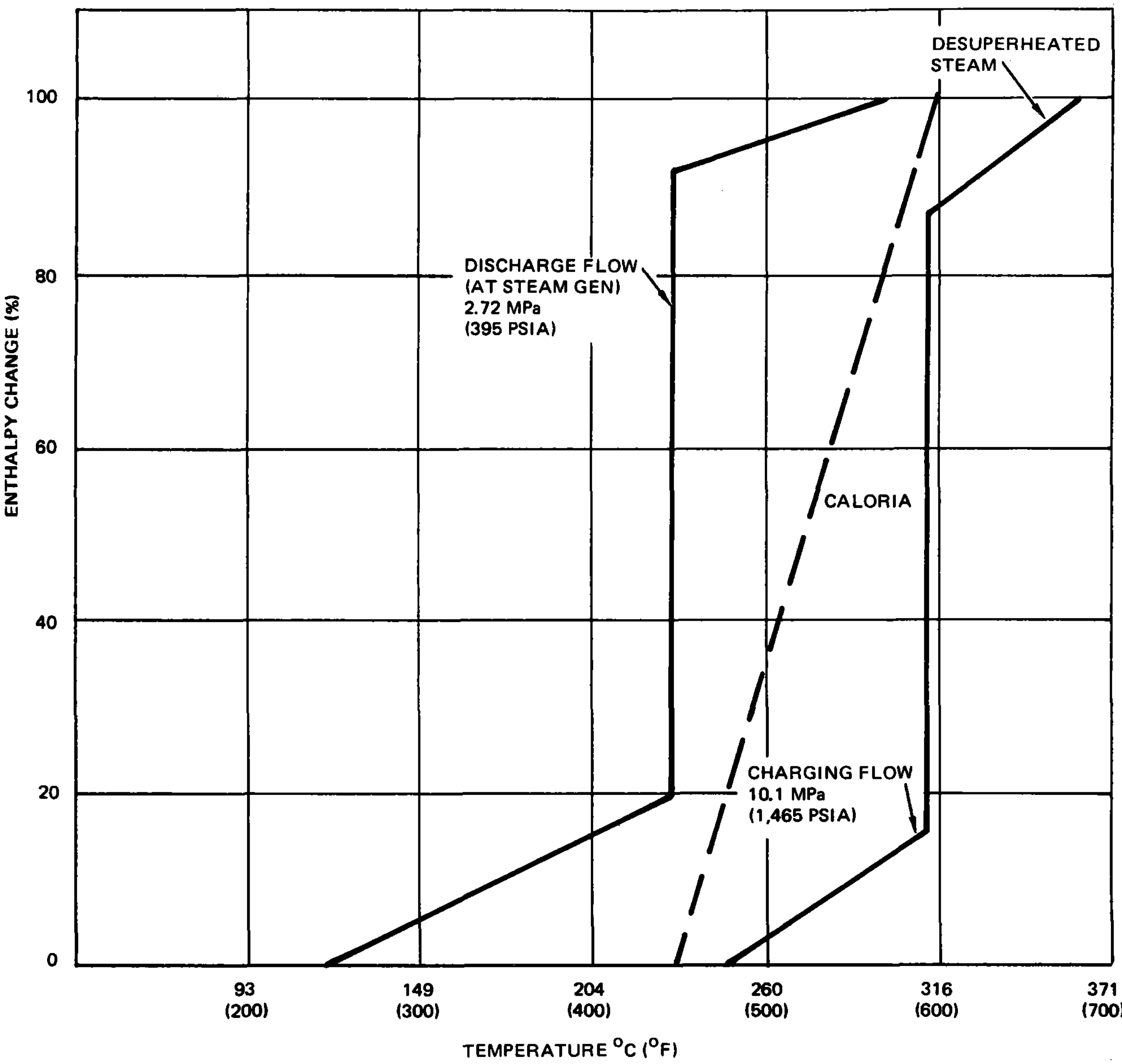

Figure 3-11. Thermal Storage Charging and Discharging Characteristics (Commercial System)

continuous hot stand-by requirement, the relatively low driving temperature of the heat-transfer fluid, and the lack of a water accumulating capability complicates the control problem for a single-pass steam generator and increases the possibility of sometimes producing wet steam. In order to minimize this possibility, a more costly and complex control system would probably be required along with the continued availability of a flash tank for flow diversion purposes.

At this point in the evaluation of steam generator approaches for the commercial system, some influence from the preferred Pilot Plant configuration was introduced. Since the purpose of the Pilot Plant is to demonstrate the overall operational capability of a complete solar electric system, it would be desirable to select a low-risk, well-established approach to thermal storage steam generation since it is not one of the mainline issues to be investigated. As a result, for the Pilot Plant, the more conservative drum or kettle-boiler approach to thermal storage steam generation would be attractive because the cost differential between them and the single-pass approach would only be approximately \$30,000 to \$40,000, excluding controls. This would allow the primary attention to be focused on the overall issue of receiver, thermal storage, and turbine operational interaction and control without spending a great deal of concern with a secondary level component such as a steam generator. As a result, for the current commercial system definition activity, the approaches to steam generation were narrowed to either a kettle- or drum-type configuration. As commercial solar thermal electric systems become viable, renewed consideration should be given to single-pass steam generation concepts due to their obvious cost advantages.

Although this selection of either a drum- or kettle-type configuration for the thermal storage steam generators is different from the selection of a single-pass concept for the receiver steam generator, it should be abundantly clear that the requirements and operating environments are also completely different. Large size, high thermal mass and slow response are acceptable and even desirable in the thermal storage steam generators, but would be totally unacceptable in the highly transient and seismically sensitive environment of the receiver.

Drum-Type Versus Kettle-Boiler Concepts

The key issues involved in making this comparison are related to cost, availability from manufacturers, and complexity. From a cost standpoint, the kettle-boiler approach is clearly superior at low pressures, less than 1.38 MPa (200 psia), with the superiority falling off as pressure increases. This is because the kettle boiler has the high-pressure water/steam on the shell side of the heat-exchanging surfaces. Thus, as pressure and capacity increase, shell thicknesses also increase. The drum-type boiler has the advantage that the high-pressure water/steam is contained in the tubes and the shell can therefore be maintained at the minimum allowable gage. The drum-type boiler, however, requires the use of an elevated high-pressure drum for the separation of steam from water. Thus, a pressure/cost relationship is also incurred with the drum configuration. In addition, the tubes must be located in a vertical or near vertical orientation to insure proper circulation, particularly if a natural circulation approach were used. If forced-circulation drum concepts were used, an additional recirculating pump would be required. For either recirculating concept a return line or downcomer would be required in the design to complete the recirculation pass, thus further complicating the configuration For the Pilot Plant sizes some comparative cost data was available, and a kettle boiler with separate preheater and superheater units was found to be about 25% less expensive. This trend is partly due to the much more competitive nature of the kettle-boiler market which supports many competing manufacturers each of which has a family of designs. The vertical-tube drum-type boiler, using oil as a heating fluid, is sufficiently specialized to require a special-order and special-design situation with a corresponding higher cost. Thus, the kettle-boiler concept was selected for the baseline design. It should be noted that separate preheater and superheater devices are required to supplement the kettle-boiler operation since the kettle boiler relies on high-boiling heat-transfer coefficients in a "stagnant" pool of water. This approach would be very inefficient in either preheating the feedwater to the saturation temperature or superheating the resulting steam. Therefore, a complete steam generator unit or "train" consists of a preheater, a boiler, and a superheater joined in a series configuration.

Single Versus Multiple Heat-Exchanger Units

With the selection of the kettle-boiler approach for the steam generator, the next issue to address is multiplicity of units. Since the majority of the cost and heat-transfer duty (surface area) is contained in the kettle boiler, it is possible to evaluate multiple-unit approaches by considering size and cost impacts on just the kettle boiler while ignoring the preheater and superheater. The results of a comparative cost study carried out for kettle boilers as a function of surface area are shown in Figure 3-12. This study treated different-sized boilers and adjusted the costs by the number required to produce 285-MWt power. It is seen that the relative cost decreases with increasing surface area to a point at which factory construction becomes prohibitive due to transportation and handling considerations. At that point, significant cost increases occur because of field erection requirements. The curve also shows a flattening out trend which occurs because of the shell costs which become significant at large sizes. If piping and valve costs were also included, which would be required to tie together many modules, the trend shown in the figure would be accentuated since the purchase and installation costs of many smaller piping elements required for large numbers of units outweighs the piping cost incurred to tie fewer, larger units together over the piping size ranges involved. Based on the transportation limit and the relative cost data shown in the figure, five parallel units or trains were selected as the baseline for the commercial system. Because the data presented in Figure 3-12 was based on a series of point designs, additional steam generator optimization is required before the final design is selected.

Sizing and Design of Steam Generator Train

The specific design of the steam generator train, which is sized to accommodate one fifth of the total steam generator duty (five parallel trains required), was arrived at as a result of a series of studies which focused primarily on initial cost, pumping power required on the oil side, and steam-side pressure drop. Implicit in these factors are the principal heat-exchanger design parameters which include tube length and diameter, number of tubes, number of passes, tube arrangement, and shell size. Wherever possible, standard heat-exchanger design information was included in the analysis to ensure that final design would be consistent with standard practices in the

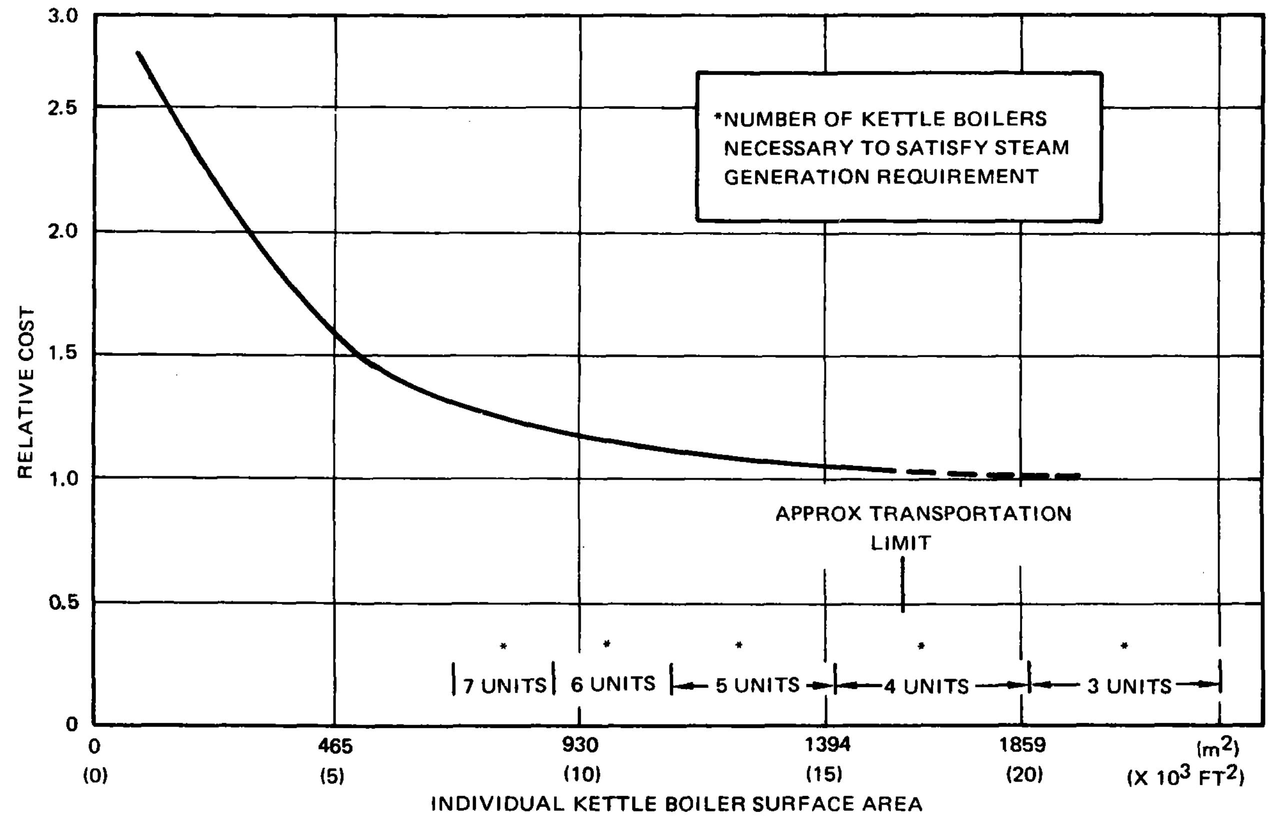

Figure 3-12. Nominal Size Impact on Kettle Boiler Cost

heat-exchanger manufacturing industry. These factors include the use of standard tube sizes and tube arrangement patterns and spacings.

The heat-exchanger designs which resulted from the optimization analysis are summarized in Tables 3-17 through 3-19 for the superheater, kettle boiler, and preheater respectively. For all design analyses a water/steam side fouling factor of 0.00029°C-m/W (0.0005 F-hr-ft^2/Btu) was assumed while a fouling factor of 0.00087 °C-m/W (0.0015 °F-hr-ft^2/Btu) was used for the Caloria side. These factors are consistent with heat-exchanger handbook values for general water and hydrocarbon heat-exchanger designs. It is felt that these are very conservative for the current design activity due to

Table 3-17
SUPERHEATER DESIGN SUMMARY

Number of Units	5
Configuration	Horizontal U-Tube Crossflow
Tube Side Fluid	Caloria HT-43
Shell Side Fluid	Steam
Number of Tubes	720
Tube Length	15.5m (51 ft)
Tube OD	1.9 cm (0.75 in.)
Tube Wall Thickness	0.21 cm (0.083 in.) (BWG-14)
Pitch (Staggered)	2.38 cm (15/16 in.)
Shell Length	8.23m (27 ft)
Shell Diameter	1.14m (3.75 ft)
Shell Wall Thickness	1.9 cm (0.75 in.)
Number of Passes	4
Mean Heat Transfer Area	594m^2 (6,389 ft^2)

Table 3-18
KETTLE-BOILER DESIGN SUMMARY

Number of Units	5
Configuration	Horizontal U-Tube
Tube Side Fluid	Caloria HT-43
Shell Side Fluid	Water/Steam
Number of Tubes	1,232
Tube Length	18.4m (60.5 ft)
Tube OD	1.9 cm (0.75 in.)
Tube Wall Thickness	0.21 cm (0.083 in.)
Pitch (inline)	2.38 cm (15/16 in.)
Shell Length	9.15m (30 ft)
Shell Diameter	1.49m/2.29m (4.9 ft/7.5 ft)
Shell Wall Thickness	3.8 cm (1.5 in.)
Mean Heat-Transfer Area	1,204m^2 (12,948 ft^2)

the high water quality which must be maintained because of the single-pass-to-superheat receiver (20-50 ppb dissolved solids) and the lack of deposit buildup observed in Caloria fouling tests. In all cases, carbon steel was assumed to be the tube and shell material.

In reviewing the design summary data, it is noted for all heat-exchanger elements that the higher pressure fluid is maintained on the shell side. For the kettle-boiler element which accounts for approximately 75% of the total heat-exchanger train cost, the boiling fluid (water) must be contained in the shell side to permit the pool boiling to occur. For both the preheater and superheater, it would be possible to reverse the flow paths by passing the low-pressure Caloria on the shell side. This design approach was rejected for two reasons. First, it is reasonable to assume that the relative magnitude of the water and oil side fouling factors noted above is also indicative of

Table 3-19

PREHEATER DESIGN SUMMARY

Number of Units	5
Configuration	Straight Tube Floating Head, Counterflow
Shell Side Fluid	Water
Tube Side Fluid	Caloria HT-43
Number of Tubes	800
Tube Length	7.47m (24.5 ft)
Tube OD	2.54 cm (1 in.)
Tube Wall Thickness	0.21 cm (0.083 in.) (BWG-14)
Pitch (Staggered)	3.18 cm (1-1/4 in.)
Shell Length	7.93m (26 ft)
Shell Diameter	1.14m (3.75 ft)
Shell Wall Thickness	1.9 cm (0.75 in.)
Number of Passes	2
Mean Heat Transfer Area	435m^2 (4,684 ft^2)

the relative fouling tendencies. Therefore, minimum downtime and maintenance costs and maximum benefit would result during cleaning operations if the Caloria were contained in the tubes.

The second and more important reason as far as the superheater is concerned is a characteristic which is really most sensitive in the Pilot Plant. For a heat exchanger which must operate over a wide flow range, sufficient flow rates must be maintained to ensure high Reynolds numbers and high heat transfer over the entire operational range. For the case of the tube side fluid, this means that the minimum flow rate must be sufficient to produce a turbulent flow and heat-transfer condition. When full-flow conditions are established, significant increases in pressure drop can occur. From a system standpoint, a high full-flow steam pressure drop through the superheater is unacceptable because it can compromise turbine performance when operating from admission steam. Thus, it is undesirable to pass the steam through the tubes of the superheater. If the steam is kept on the shell side, as in the case of the current design, a minimum Reynolds number constraint is not applicable in order to provide good heat transfer because the staggered tube arrangement over which the steam passes continually breaks up the formation of laminar boundary layers. The cost implication of containing the high-pressure steam in the shell side represents no more than a few percent of the total heat-exchanger cost due to the small shell diameter and minimum gage limitation imposed by the code for only slightly pressurized heat-exchanger shells.

A schematic of the equipment comprising one of the five parallel heat-exchanger trains required for the commercial system is shown in Figure 3-13, along with pertinent performance data at the design flow condition. The purpose of the Caloria bypass leg around the superheater is to provide a trim control on the outlet superheated steam temperature. At the design point, 25% of the inlet Caloria flow would be routed through the superheater with the remaining 75% bypassing the element. As the steam demand decreases, the improved heat-transfer performance of the heat-exchanger train at reduced flow results in a lower percentage flow being diverted to the superheater. On the other hand, if the superheater inlet Caloria temperature were to drop to 304°C (580°F), which could occur when the final increment of stored energy is being extracted from the storage tank, the total Caloria flow rate to each of the trains would have to be increased by 289 Kg/sec (2.29×10^6 lb/hr). To maintain the 299°C (570°F) exit steam condition, 72% of the inlet Caloria flow would be diverted through the superheater. At the same time, the Caloria outlet temperature leaving the preheater would increase slightly above the design value to 233°C (452°F). Although extensive analysis of such off-design operation was not made for other cases, intuitively, it can be seen that as the inlet Caloria temperature is further reduced below the 304°C (580°F) level considered here, the Caloria flow diverting valve would be asymtotically increasing the percentage of the total flow being diverted to the superheater. Once all of the flow was diverted, a further reduction in inlet Caloria would result in a reduction of the maximum steam temperature leaving the superheater.

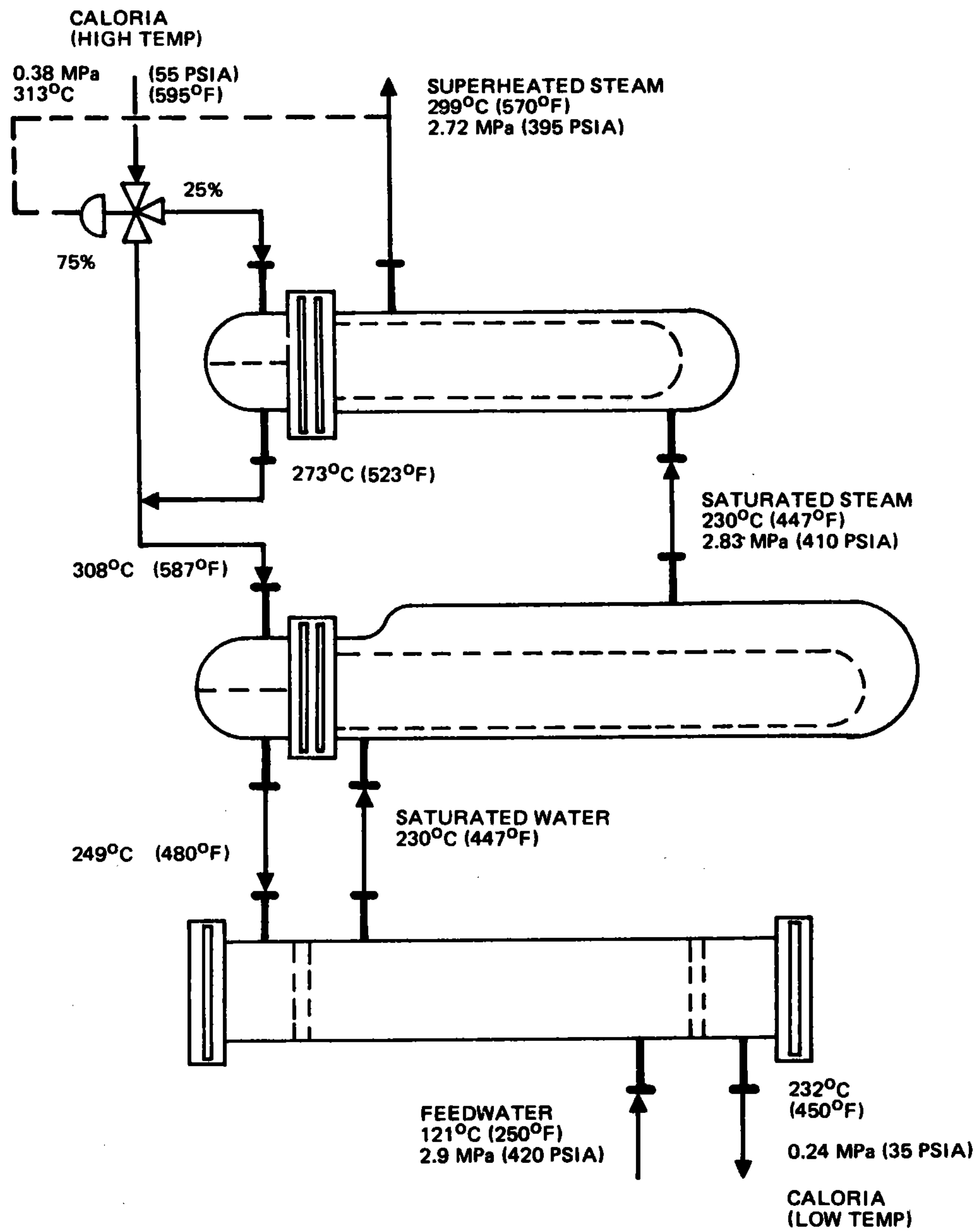

Figure 3-13. Commercial System Thermal Storage Steam Generator Train (1 of 5 Parallel Units)

3.2.6.2 Fluid Pumps and Piping

The function of the Commercial Plant TSS extraction loop pumps and piping is to provide a means to remove heat from the TSUs to produce steam. Unlike the charging loop, which has varying flowrates throughout the day, the extraction loop operates at or near maximum capacity most of the time. The major exception occurs during days of intermittent cloud cover, when the flows will fluctuate.

A typical daily cycle is shown in Figure 4-46 in Section 4, which is for the Pilot Plant, but both the Pilot and Commercial Plants have the same shapes, with the orders of magnitudes differing.

The analysis of the Commercial Plant TSS extraction loop proceeded in the same manner as that of the charging loop (Section 3.2.5.3). The quantities of the various items assumed to be used were estimated and these data were combined in the cost optimization computer program.

As with the charging loop, addition of more than five pumps resulted in no capital or parasitic power cost reduction, so these curves are now shown. The number of pumps has even less an impact on the extraction loop due to the nature of the duty cycle. The velocity, however, still affects both capital and power cost as shown by Figure 3-14.

It can be seen that the shape of the curve is quite similar to that for the charging loop, with the optimums occurring at approximately the same velocities. The extraction loop, however, is more sensitive to operating at nonoptimum conditions, as evidenced by the better defined minimum. This is because the extraction loop operates primarily at maximum conditions and it has a higher maximum than the charging loop. An optimum design was developed, with a fluid line velocity of 3.9m/s (13 ft/s), which resulted in a pipe size of 26-inch, Schedule 40 for main lines, and 14-inch, Schedule 40 pipe for segmented flow spools. As for the charging loop, it was decided to use five identical pumps in parallel with a combined maximum capacity of 2.0m^3/s (32,000 gpm). The pump drivers are single-speed induction motors. Other design characteristics are shown in Table 3-20.

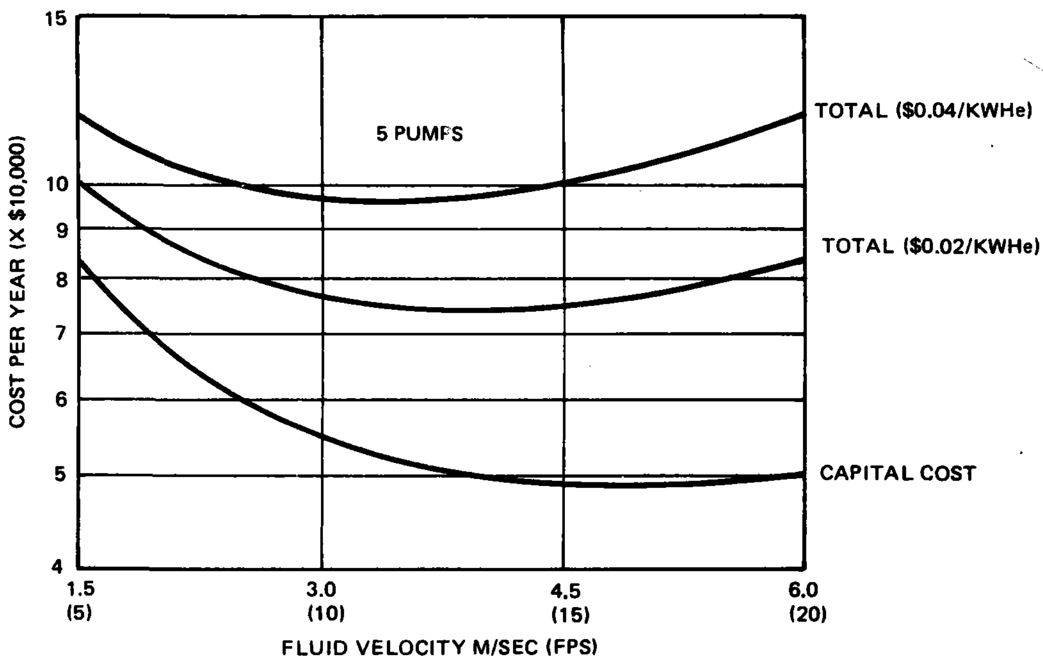

Figure 3-14. Annual Cost Optimization with Fluid Velocity and Electricity Cost

Table 3-20

COMMERCIAL PLANT EXTRACTION LOOP DESIGN

Item	Quantity	Description
Pumps	5	Dean Brothers Model No. R484-8x10x15-1/2 in.
Motors	5	Single-Speed Induction Motor, 250 Hp, 4,160 Volt, 3-Phase
Pipe	61m (200 ft) of 26-in., Schedule 40** 91m (300 ft) of 14-in., Schedule 40*	

**USA Standard pipe sizes given (no SI equivalent pipe sizes)
*14-in., Schedule 40 pipe used for all segmented flow

3.2.7 Controls and Instrumentation

3.2.7.1 Requirements

The control system will be required to respond automatically to commands from operating personnel and/or from the central controller. It will be required also to change smoothly and rapidly from one operating mode to another and, subsequently, to respond smoothly and rapidly to changes in thermal charging rate and/or extraction rate.

The basic requirement during the TSS charging mode will be to maintain the fluid output temperature from the charging heat exchangers at the command set point, nominally 316°C (600°F). This temperature will be maintained within $\pm$ 1°C of the nominal setting as a steady-state error band. Transient overshoot can be allowed to be about $\pm$4°C from the nominal final value.

In general, the penalty for operating above the nominal maximum steady-state temperature is that the degradation rate of the fluid increases exponentially with temperature. Operation below the nominal reduces the total heat stored and reduces turbine efficiency. Transient errors in temperature can be allowed to be relatively large.

In the extraction mode, the TSS must deliver steam at the nominal maximum temperature and pressure of 299 $\pm$4°C (570 $\pm$ 7.2°F) and 2.72 $\pm$ 0.069 MPa (395 $\pm$ 10 psia) respectively, with a rather broad tolerance on the temperature and pressure, since tighter requirements by the turbine are not necessary.

The desuperheater control loop will be required to hold the desuperheated steam output temperature at 360 ± 1.1°C (680°F ± 2°F). Transient response can allow ± 5.6°C (10°F) overshoot from the final value.

In general, response times are not required to be rapid since the large thermal inertias involved are expected to provide the longest controlling lag time in all these control loops. In addition, TSS damage will not result from control errors or overshoot, since the subsystem is inherently a passive one. Only reduced efficiency will result except for the following possibilities:

A. Thermal stress fatigue in the heat exchangers
B. Increased fluid degradation rate during excessively high-temperature excursions.

Instrumentation capabilities are required for the following:

A. Direct monitoring of temperatures, pressures, flow rates, and liquid levels with display and recording capabilities at a remote control
B. The above analog signals are to be used by data loggers and/or computers to store data and calculate several parameters such as:
    1. State of charge of the TSU's, i.e., KWHt available heat
    2. Quantity of fluid in the TSS based on the average temperature distribution vertically in each TSU tank.
    3. Average fluid temperature in each TSU and a warning signal when this average falls below 169°C (350°F)
    4. Warning signals to operating personnel regarding violation of limits on critical parameters (such as TSU tank pressure)
C. Provide recording, indicating, and enunciating displays on the TSS section of the master control panel for use by operating personnel

A combination of instrumentation, control, and logic functions will be required in the charging loop to aid personnel in automatically switching from one combination of pumps and heat exchangers to another as the demands on the system are changed.

3.2.7.2 Design Analyses

Instrumentation

The charge condition of the TSU will be determined by measuring the temperature distribution vertically through the bed. An array of fourteen thermocouples spaced evenly 1.2m (4 ft) apart along each tank centerline between the upper and lower manifold will be used as the basic temperature measuring system for determining the thermal storage charge condition.

A second set of thermocouples will be installed in two groups, one just below the top manifold and the other group above the bottom manifold. Each group consists of five extra thermocouples spaced at 0.2m (1 ft) intervals vertically between the thermocouples already provided in the previous set. These measurements will furnish temperature profiles near the top and bottom manifolds to determine more accurately the steepness of the thermocline as it approaches either a fully charged condition or a fully discharged condition. These measurements will aid greatly the control personnel in predicting when to stop a charging or discharging operation to remain within control limits on the TSS.

Another set of ten thermocouples will be arranged in various places in each TSU, as required based on Pilot Plant experience. There will be other temperature measurements throughout the TSS.

The thermocouples used for control input will have a special redundant system. Two thermocouples are used for each measurement with both measurements brought to indicating instruments on the control panel. Control personnel can choose between the two thermocouples by throwing a switch; the switch will connect either thermocouple into the associated control loop, while the remaining thermocouple will give a temperature indication on the control panel only. The operator can compare these two readings with other readings in the system and thereby evaluate very quickly which one is the more desirable to use for control purposes, and the operator can evaluate whether a measurement element is faulty or not.

System Dynamics

The general equations describing the system dynamics of the TSS were included as part of the entire power plant dynamic study which is reported in Volume II, Section 4.7.

TSS Remote Control

The TSS Panel provides for operator control of thermal storage operations. All remote-control valves, on-off and modulating, can be controlled with manual inputs. The panel will include display equipment to the extent required for operator supervision and manual control and will include recording equipment to the extent required for power-plant record keeping and for operational analysis.

In addition to the provisions for operator manual control, the control panel has provisions for semiautomatic control with operator supervsion . The TSS control loops are basically independent, electronically, of one another. However, a control logic system will be included having high flexibility so that some automatic interaction can be provided for, after experience is gained with the Pilot Plant system characteristics.

The control panel includes the circuits for closed-loop analog control of the various heat-transfer-fluid flowrates. Additionally, there are logic circuits for fluid pump starting, stopping, and valve resets, as flowrate demands vary. Additional details are shown in Section 4.3.9.2.

Control Logic

The TSS contains a number of areas in which decisions can be made regarding alternative operating modes, i.e., whether one or two or more heat exchangers should be used to respond to a given demand. The guiding strategy will be to base the design on Pilot Plant experience. It is estimated that the time constants associated with the TSS will be long compared to operator response time. Response times with the rapidity afforded by automatic logic circuits are not required. Repetitive decisions causing operator boredom can profitably be accomplished with logic circuits. Operational experience with the Pilot Plant can best provide input for evaluating such requirements.

Manual override and control commands by the Master Control will be possible because it will have predictive ability regarding future turndown ratios and can provide switchover point decision supervision and override with the objective of minimizing pumping power requirements and maximizing heat exchanger life by reducing thermal cycling.

This control logic design also builds in stability through an inherent "toggle" action, in which the decision process at switchover points allows the system to proceed beyond the nominal switching point before a command to switch is executed. This prevents the system from cycling back and forth when demand requires continuous operation at or very close to a nominal switching turndown ratio. This applies to switching from one to two heat exchangers or from one to more than one pump. The final control logic design will generalize and simplify to limit the number of possible operating choices, thereby relieving the operator of having to make too many decisions.

3.2.7.3 Design Description

The description of controls is organized under five subheadings: Charging Loop Controls, Extraction Loop Controls, Desuperheater Controls, Ullage Maintenance and Fluid Maintenance Unit Controls, and Control Equipment. Additional details are given in Section 4.3.9.3, which describes the almost-identical Pilot Plant TSS controls.

Charging Loop Controls

Figure 3-15 is a schematic diagram of the piping system and control elements that are required for thermal-storage-heater fluid-flow control. Heat-transfer fluid from the lower manifold of the TSU is pumped through the five TSH's (in parallel) in which the fluid temperature is increased by heat transfer from a supply of steam from the desuperheater.

Five centrifugal thermal storage charging pump units, TCP-1 through TCP-5, are connected in parallel with check valves in the pump-discharge lines to prevent backflow of fluid if a pump is shut down while other pumps are in operation. Flowmeters THFFR-1 through THFFR-5 measure the flowrates to each heat exchanger. The valves connecting the pump outputs are normally closed.

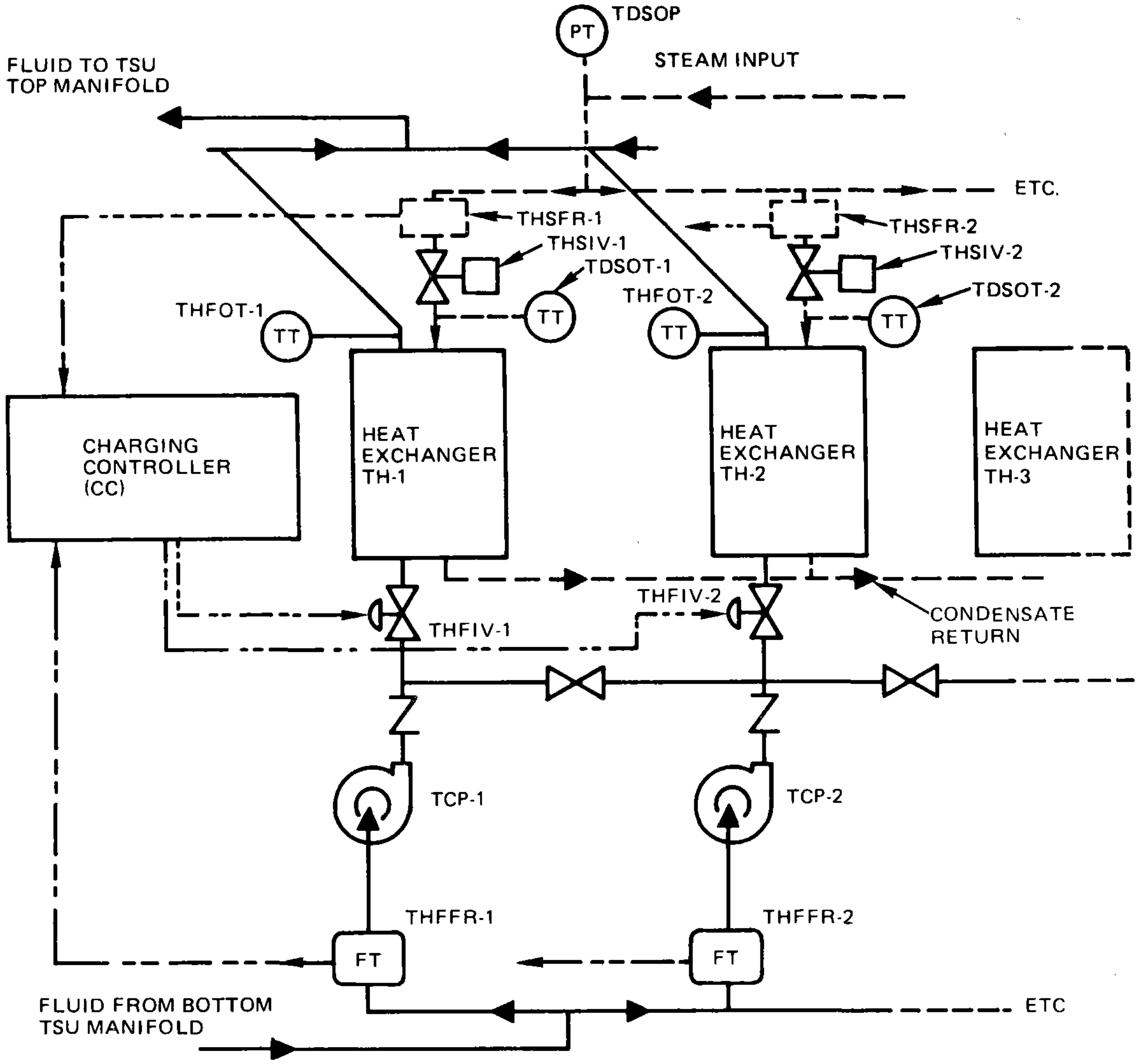

Figure 3-15. Charging Heat Exchanger Control Loop Components

Charging controller CC, which physically is a portion of a larger assembly identified as the TSU Controller, moduletes thermal-storage-heater fluid inlet valves THFIV-1 through THFIV-5 for continuous adjustment of each heat-exchanger flowrate to maintain constant heater fluid outlet temperatures THFOT-1 through THFOT-5, as heating steam flowrates vary. Electronic circuits in the charging controller process the signals from each set of two redundant temperature sensors, in each heat-exchanger fluid-exit line. The pumps have nominally-identical pressure-flowrate characteristics. When charging-loop total flowrate increases to the extent that the pumping capacity is approached, logic circuitry in the charging controller then switches on another pump and associated heat exchanger. These pumps have nominally-identical pressure-flowrate characteristics, and the control valves continue to operate with nearly-identical pressure drops, maintaining equally-good throttling control characteristics in the parallel flow paths.

The five pump outputs are connected (by means of isolation valves), permitting a number of combinations of different heat exchangers with different combinations of pumps if this is desired. However, each heat exchanger has its flow modulated individually by means of its own input flow control valve whose position is governed to maintain the associated heat-exchanger fluid-output-temperature constant. There are thus five temperature control loops. Each has its own feed forward signal consisting of the output of the steam flowmeter as well as the output of the fluid flowmeter measuring the fluid passing through the associated heat exchanger.

Modulating butterfly valves were selected because of their simplicity and adequate throttling control characteristics with low pressure loss under maximum flowrate conditions. A type with metal-to-metal seating contact between the valving element and the housing will provide adequate shutoff capability for this application, with no need for a flexible seat seal with limited capability. They are pneumatically actuated. Closed loop control of valve displacement from its closed position is provided by a positioner. A pneumatic input signal to the positioner is related to a mechanical feedback measurement of actuator and valve position, through a servovalve mechanism. An electropneumatic transducer converts an electric current signal input into a pneumatic output signal to the valve positioner. Control valve displacement is thereby controlled by electrical command signals from the charging controller portion of the TSU Controller. Solenoid-pilot-operated shutoff valves provide for on-off control of steam flow through each heater. The fluid charging loop control block diagram and description are identical to that for the Pilot Plant in Section 4-3.9.3, under Charging Loop Controls. This section should be consulted for details.

When all system controlled variables are at their nominal regulated values, the weight flowrate of heat-transfer fluid through the heaters will be throttled to be proportional to the heat energy to be extracted from the steam, i.e., heat-transfer fluid-weight flowrate will be controlled to be proportional to steam-weight flowrate. Close accuracy in generating fluid-weight flowrate command signals is not required. With the selected temperature-control concept, the commanded heat-transfer-fluid flowrate signal is primarily a scheduled function of the system variable input steam flowrate with trimming corrections as required to eliminate steady-state error in the controlled output (fluid temperature).

Transient response time lags in measurements of flowrates typically are orders of magnitude less than the dominant time lags in this control loop. Changes in incoming steam flowrate will result in fast-response changes in fluid flowrate, with fluid flowrate scheduled to be approximately correct in maintaining the reference temperature set-point value. Changes in fluid flowrate, resulting from integration of temperature error, will occur at a relatively slow rate.

However, certain predictions can be made regarding the TSS dynamic response based on practical experience and observations made during the SRE experiment and from future Pilot Plant operation. For Pilot Plant equipment, it is concluded that the controlling time lag or time response of sensing elements and signal conditioning equipment is on the order of a few seconds or shorter, the time response of valves are on the order of tens of seconds while the time response of the heat exchangers and associated fluid flow lines are on the order of minutes. The limits on time response of the entire system to changes in demand are, therefore, limited only by the thermal inertia of the passive elements such as heat exchangers and not by the sensing and active control elements such as valves. For the Commercial Plant the larger equipment (valves, etc) will have longer time response.

Adequate control system stability is predicted to be achieved, based on this simplified time constant analysis. However, the system will display transport lag which may be adequately dealt with at the nominal design maximum throughput, but which may change drastically as the turndown ratio is increased, resulting in large increases in transport lags.

The feed-forward-type automatic controls used are designed to deal with the problem as described above in Section 3.2.7.3 under the subheading of the control loop under consideration, e.g., Charging Loop Controls, etc.

The time lags, t_1', t_2', t_3', can best be determined once Pilot Plant data are available, at which time scaling laws can be applied. Preliminary estimates will be available once the Pilot Plant detailed design (phase 2) has been completed. Time constants, t_n', for the Commercial Plant will have the same designation as for the Pilot Plant except that a prime will be added.

The dominant time constants in the temperature control subsystem are related to thermal response time lags in the heat-exchanger tubes and in the temperature sensors. Typically, in a heat-exchanger continuous-temperature-control process, in which the control loop is only closed on controlled temperature measurement and a reference signal, output/input phase lag under transient operating conditions results in an inherent tendency toward oscillatory operation. A low-gain temperature-control loop, as required for dynamic stability, generally results in excessive temperature control bandwidth and in excessive transient error in response to input excursions. The addition of an integrator to minimize bandwidth and error in a loop which is closed on temperature signals only, generally is not feasible, because the additional phase lag that an integrator introduces will usually result in a dynamically unstable control loop.

With the selected control concept, fluid flowrate is scheduled to track excursions in steam inflow, whether or not the outlet temperature is in error, e.g. corrective action is initiated by a change in input before an output error occurs. Integration of temperature error is introduced for trimming purposes only, in a manner that does not contribute phase lag to the basic scheduled control loop.

The TSH flow control concepts which have been described in this discussion are designed for "load following" capability. However, as is often the case in power plant systems, efficiency and capability may be improved under transient operating conditions if the power plant master controls are designed to provide "feed forward" signals to the storage subsystem to result in anticipatory response to significant impending changes in operating conditions.

Extraction Loop Controls

Figure 3-16 is a schematic diagram of the piping system and control elements that are required for operation of the TSU extraction loop.

In the extraction mode of operation, heat-transfer fluid is pumped from the upper hot end of the TSU, through parallel steam generators, to the lower cooler end of the TSU. Centrifugal pumps, powered by alternating current electric motors, operate in parallel. A set of shut-off valves, labeled TSFBV in Figure 3-16, are provided to interconnect the outputs of the pumps should this be required; these are kept shut during normal operation. Check valves prevent backflow of fluid through a pump that is shutdown while the other pump is in operation.

If a pump should fail or inadvertently shut down, the other pumps can supply fluid to all steam generators, within the capacity limitations of the remaining pumps. Under such conditions, the extraction controller provides a signal to the power plant master controller to limit the steam flowrate demand. The use of five pumps operating in parallel enhances system reliability, and also minimizes pumping motive power demand on the electrical power supply during operation with low-flowrate demands, which can be accommodated with one to four pumps as required.

In general, upon receipt of a command from the power plant operator, or master control, to provide a specific steam output from thermal storage to

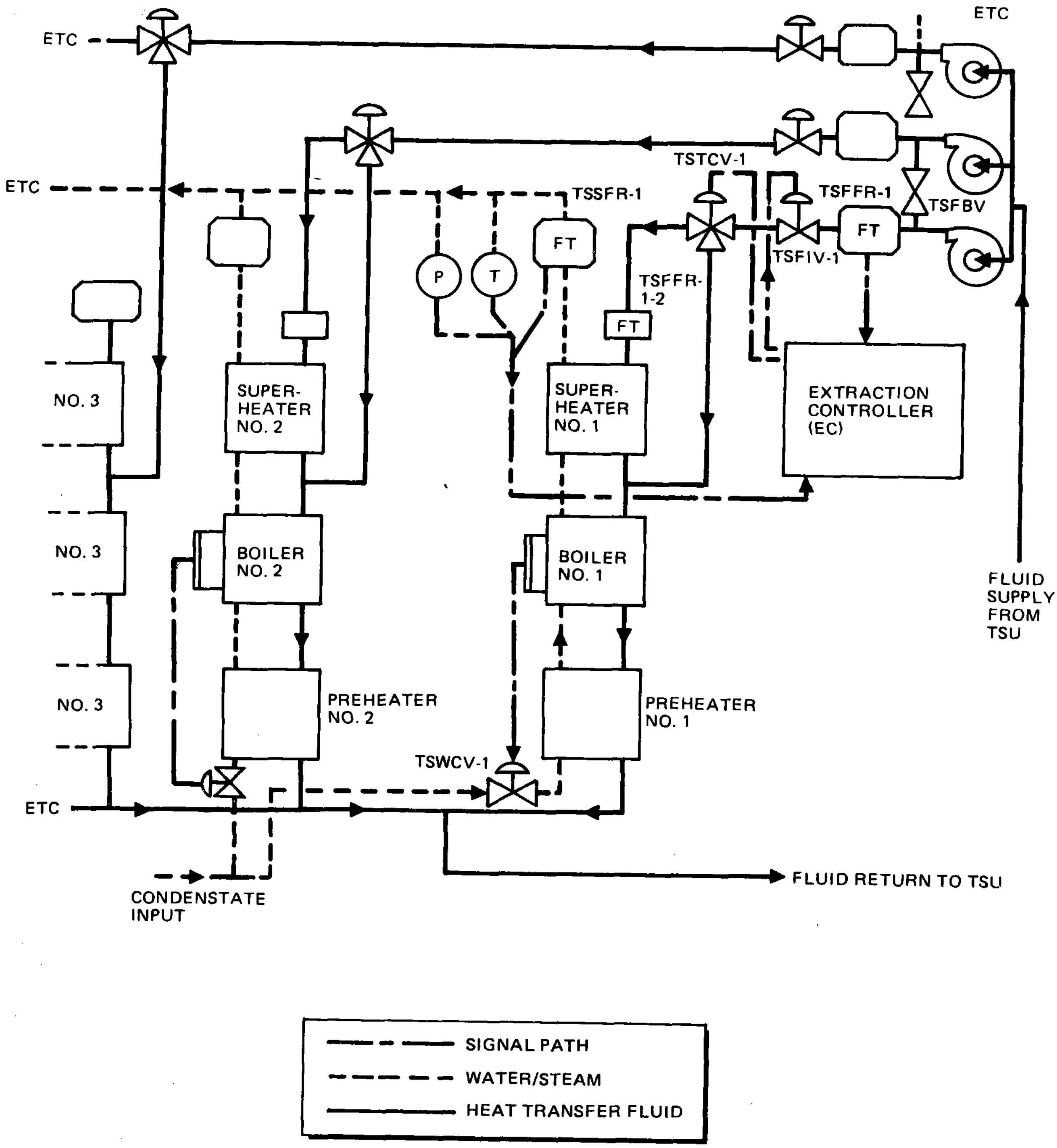

Figure 3-16. Block Diagram of Extraction Loop Controller

to the turbine, the superheater-fluid inlet valve (TSFIV) is opened to the appropriate position to provide the necessary hot fluid flow to the steam generator. As steam flow increases the turbine-admission steam valve will open to maintain system turbine inlet pressure. If all system-controlled variables are at their nominal regulated values, the weight flowrate of heat-transfer fluid through the steam generator will be approximately proportional to the steam-weight flowrate. Therefore, as indicated in Figure 4-55 in Section 4, a fluid flowrate command signal, is generated in proportion to the measured steam flowrate.

The superheater temperature control 3-way valve, TSTCV is positioned as required to maintain superheater steam outlet temperature. Steam temperature at the superheater outlet is measured by redundant temperature sensors. As indicated in Figure 4-55 in Section 4, a first-order time lag in temperature measurement is presumed. The measured average temperature is compared with a reference temperature signal and the difference, or error signal, is used to reset the 3-way valve (TSTCV) position in such a direction as to bring the temperature back to the reference value. The flow not going to the superheater is bypassed to enter the boiler along with the exit flow from the superheater.

The steam generator water control valve, TSWCV, functions to maintain proper water level in the boiler under all demand conditions upon command from the boiler water level sensor. A liquid level sensor measures the boiler water level which in turn is compared with a water level command value. The resulting difference or error signal is amplified and sent to reposition the water inlet control valve to the preheater in such a direction as to bring the liquid level back to the command level.

Even though steam generators are operating in parallel, each is controlled independently, with a control circuit as described in the analogous Section 4.3.9.3. This section should be consulted for details.

Analog flowrate control is supplemented by extraction controller logic and sequence-of-events control for operation with either one or two of the steam generators.

Fluid flowmeter TSFFR-1-2 is used for monitoring purposes only (not control), permitting the by-pass flow to be obtained by subtraction from the total flow indicator TSFFR-1-1.

Desuperheater Controls

Figure 3-17 is a schematic diagram of the piping system and control elements that are required for control of the TSU inlet steam desuperheater. The operation is identical to that for the Pilot Plant, as discussed in Section 4-3.9.3. However, a single desuperheater supplies five heat exchangers.

Under conditions for TSU charging loop operation, desuperheater inflowing steam is mixed with water to regulate the desuperheater outlet steam temperature at approximately 343 C (650° F). Desuperheater coolant water is supplied from the solar energy receiver panel feedwater pump at approximately 14.5 MPa (2,100 psia).

With the selected temperature control concept, the commanded coolant water flowrate is primarily a scheduled function of the computed incoming heat energy; with trimming corrections applied as required in eliminating steady-state error in desuperheater effluent temperature. Changes in influent enthalpy result in fast-response changes in coolant water flowrate in maintaining an approximately correct effluent temperature, followed by slower responding changes that integrate control error to a nominal zero.

Ullage Maintenance and Fluid Maintenance Control

The fluid maintenance unit and UMU are both ancillary to the primary system. There is negligible physical interaction between these systems and the TSU system provided that these ancillary systems perform their functions within their design specifications. Consequently, discussion of these control systems are given in respective sections dealing with these functions, i.e., Ullage Maintenance, Section 3.2.3 and Fluid Maintenance, Section 3.2.4.

However, if these units do not maintain their functions within specified control limits, then they do interact with the TSU system. For example,

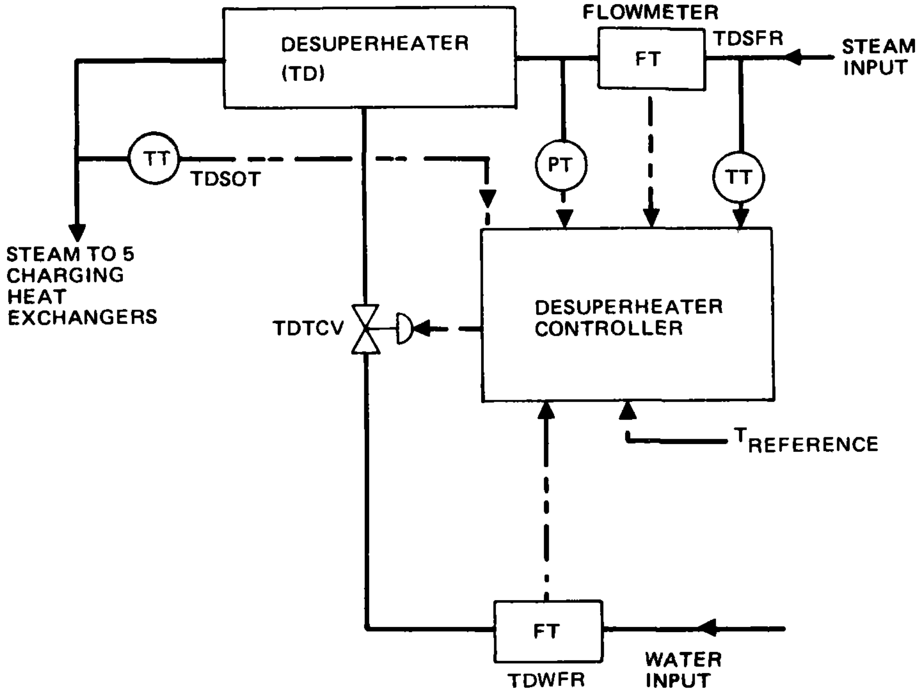

Figure 3-17. Desuperheater Control Loop Components

if the fluid filters which are part of the fluid maintenance system should become clogged, then these units will adversely affect the control functions of the TSU.

Monitoring of the filter condition and periodic checks to maintain them properly are discussed under Section 5.4 entitled Maintenance.

Control Equipment

The TSU piping system includes manually-operated shutoff valves that are used when part of the system is nonoperational and has been isolated for maintenance.

The TSU piping system also includes numerous solenoid-pilot-operated shutoff valves that are used for remote control of sequences of events, e.g., termination of TSU charging operations, initiation of TSU extraction operations. Additionally, heat-transfer-fluid flowrate through each TSH and through each steam generator is under closed loop control with a modulating butterfly for each unit. A modulating three-way valve controls heat-transfer-fluid flow division through each steam superheater and its bypass. A modulating poppet valve controls water inflow to the desuperheater. Each of the modulating valves is pneumatically actuated in response to electrical position-command signals.

Heat-transfer-fluid pumps in the TSU piping system are electric-motor powered in response to remotely controlled on-off signals. All parameter sensors, e.g. pressure transducers, thermocouples, fluid flow meters, liquid level sensors, etc, deliver electrical signals.

The use of electrical signals for parameter sensing and for control permits use of versatile electrical/electronic instrumentation and control equipment.

3.3 SUBSYSTEM FABRICATION/INSTALLATION

This section presents a brief discussion of the fabrication and installation of the Commercial Plant TSS. Because the Commercial and Pilot Plant subsystems are very similar, differing only in scale, the more detailed plans for production and installation of the Pilot Plant subsystem, given in Section 5, are also largely applicable to the Commercial Plant subsystem.

Generally, off-site or shop fabrication minimizes costs and installation time and will be encouraged by the subsystem design whenever possible. For this subsystem, off-site fabrication will include the heat exchangers, and prefabricated portions of the TSU, the fluid maintenance unit and ullage maintenance unit. The remainder of the off-site fabrication will be limited to catalog components, modified for the intended service, such as control devices, pumps, instrumentation, filters, and bulk stock.

Field construction will be performed by contractors who are licensed and experienced in the tasks they will be asked to bid. These tasks include site preparation (drain sump and interconnecting drain lines for all equipment and interconnecting lines and containment dikes, equipment foundations, and supports), the on-site fabrication/installation of the TSU's, and a mechanical contractor who will perform the remaining construction and subcontract specialty tasks. Each of the selected contractors will be responsible for his contractors and supplies.

3.3.1 TSUs Fabrication

The TSU design will permit maximum off-site or shop prefabrication of the tank plate sections, tank appurtenances, and the inlet and outlet manifolds. Shipping clearances to the site will determine the size of the largest shop fabricated subassembly. To reduce the total schedule time and to eliminate contractor mark-up, the four TSUs will be advance procured. The purchase will be made from a fabricator specializing in on-site construction of API storage vessels with special consideration given to those firms having complete off-site prefabrication facilities.

The TSUs will be field erected at the construction site with inspection and acceptance based upon the completed units. Prefabricated work, that has had source inspection of weld deposits and other critical characteristics, will be delivered as needed. Special site storage will not be needed. Field construction will be performed by the successful bidder as a continuation of the prefabrication phase.

3.3.2 UMUs Fabrication/Installation

The UMUs will be field fabricated from commercial components as part of the subsystem fabrication. No off-site work (other than vendor manufacture of standard components) is anticipated or required. Test and checkout of the UMUs can be performed independent of the systems.

3.3.3 Fluid Maintenance Units Fabrication/Installation

The fluid maintenance units includes filters to be installed in the TSU main flowlines, a storage tank for makeup fluid, and a thin-film vacuum distillation system.

The filters will be procured as equipment items and installed in the TSU main flow lines as part of the system piping by the mechanical contractor in accordance with the system plans.

The makeup fluid storage vessel will be purchased from a firm specializing in the shop-fabrication of API storage vessels. After approval by source inspection, the vessel will be transported to the construction site. Installation at a prepared site will be performed by the mechanical contractor.

The vacuum distillation systems will be purchased from a vendor as a turn-key operation and can be considered as a catalog equipment item. The subsystems will be installed at a prepared site by the mechanical contractor.

3.3.4 Pump and Heat-Exchanger Equipment Fabrication/Installation

The heat exchangers (including the TSH steam generator, and desuperheater) will be fabricated by firms specializing in the production of commercial shell and tube heat exchangers with special consideration given to those firms having designs and standard components that most closely approximate the requirements for the Commercial Plant. Prior to shipment, source inspection will examine characteristics that become concealed during the course of assembly, such as heat-exchanger tube to sheet attachment and heat-exchanger interconnection. Pumps will be procured from firms with designs and standard components that meet the required operating characteristics.

When delivered, the pumps and heat exchangers will be off-loaded and installed by the mechanical contractor on foundations previously prepared by the site preparation contractor. Their installation will permit continuation of the piping, controls, electrical wiring, insulation, and testing work.

3.3.5 Controls Fabrication and Integration

Control devices and instrumentation are off-site fabrication items. The mechanical contractor will subcontract minor control functions and install major shop-fabricated subassemblies and other specialty tasks. The controls will be installed after the large equipment items are in place.

3.3.6 Subsystem Final Installation

Fabrication/installation of piping, valves, electrical wiring, controls, insulation, painting, and installation of minor equipment items will begin immediately following installation of all major items. These tasks, items, and their installation will be handled by the mechanical contractor.

When the mechanical contractor has completed all installation assignments, the subsystem assembly will be checked. The functioning of instrumentation, all automatic electrical and pneumatic controls and backups, will be verified; all valves and pumps will be checked and activated. After completion of this phase, the subsystem will be filled with heat-transfer fluid; the pumps will be activated and fluid will be recirculated through the subsystem piping and heat exchangers. Flowrates and pressure drop through the equipment will be checked and several simulation TSU charge and extraction cycles will be performed. The fluid system will be inspected for possible minor leaks at all pipe connections and pumps. After several hours of operation, the fluid filters will be examined, cleaned, and reset.

3.3.7 Summary Commercial TSS Schedule

The master schedule summarizing the design, fabrication and test of the commercial TSS is presented in Figure 3-18.

3.4 OPERATIONAL CHARACTERISTICS

This section discusses the alternative modes of operation of the Commercial Plant TSS, with subsections devoted to each of the major operational modes. The operation is almost identical to operation of the Pilot Plant TSS; therefore, many details are given in Section 4.4 for both plant TSSs and are not repeated here.

3.4.1 Startup Mode

There are three types of startup operation which may occur with the TSS: (1) initial startup of a plant subsystem after construction, (2) cold startup after a very lengthy shutdown period, and (3) "hot" startup following previous operation within a few days, which is the startup mode encountered most of the time.

3.4.1.1 Initial Startup

It is possible to isolate each of the four TSU tanks from the others, and it is therefore possible to fill only one tank with fluid and follow the procedures developed for initial startup of the Pilot Plant (Section 4.4.1.1). However, it is recommended that all the tanks be filled with fluid before any one of them is heated. This is to safeguard against hot fluid inadvertantly being admitted into a tank containing fresh rock/sand with the normal water content. If the fluid at maximum temperature is allowed to contact damp materials the evolution of steam might create a problem.

Once all the tanks have been filled with fluid they should each be slowly brought up to near the boiling point of water. When all four have been heated and conditioned to a temperature level of about 120° C, then each tank can be conditioned up to 316° C individually, if desired.

The procedure is essentially the same as for the Pilot Plant and procedures similar to those given in Section 4.4.1.1 will be followed.

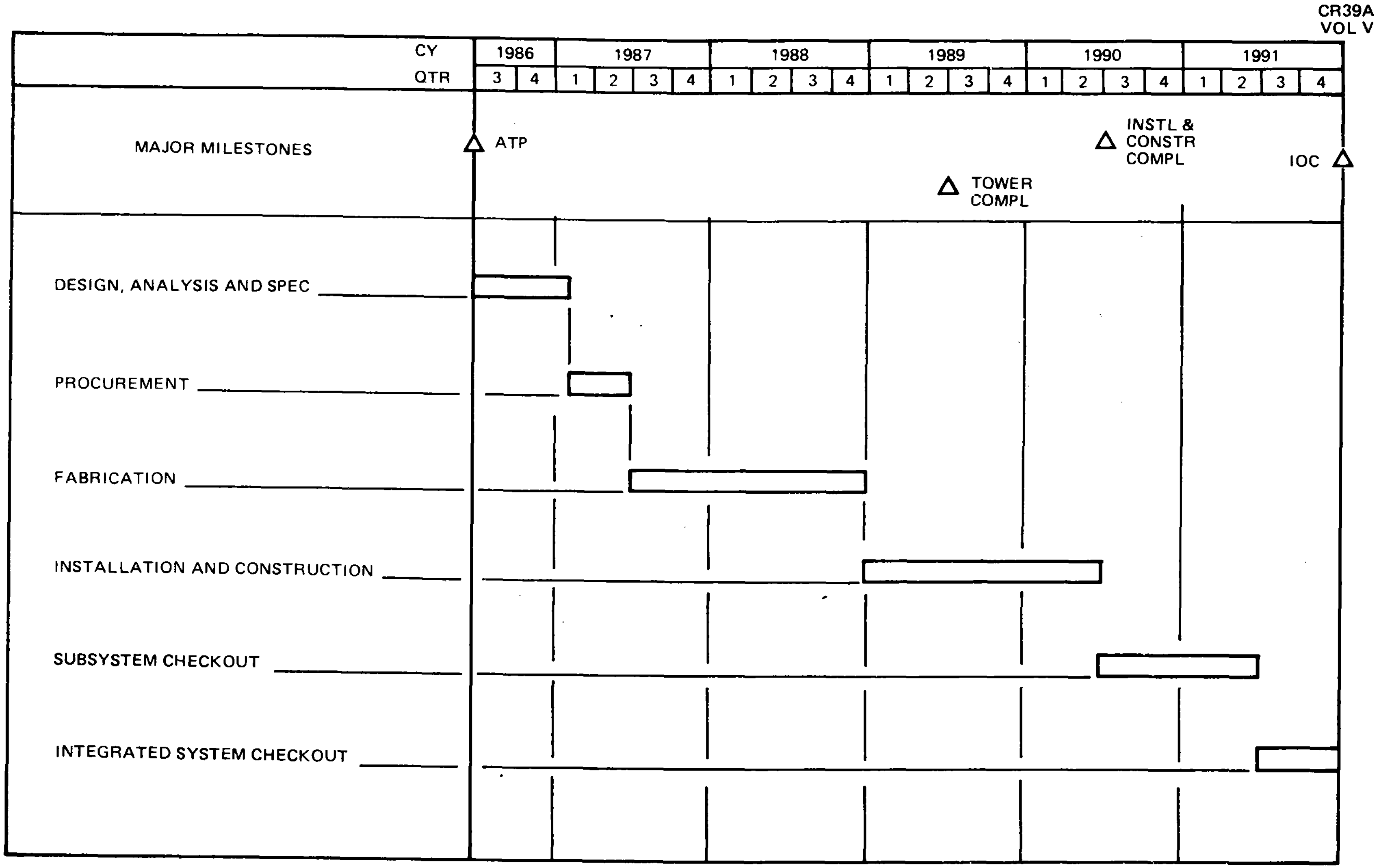

Figure 3-18. Thermal Storage Subsystem – First Commercial Plant

3.4.1.2 Cold Startup

It is predicted that, except for the initial startup following construction, it is highly unlikely that a "true" cold startup will occur. This is because the heat leakage rate from the Commercial Plant TSU is on the order of 1 to 2% per day and thus requires months to cool down to temperatures near ambient.

The piping, heat exchangers, and pumps will cool down much more rapidly than the TSU, and are expected to require a cold startup procedure after a few days of inoperation. The exact length of time after which a cold start procedure will be required can best be estimated once experience has been gained with the Pilot Plant; however, it is estimated to be longer than 24 hr.

One of the operational lessons gained from running the SRE is that gaseous degradation products tended to collect in the upper parts of the pump volute when the system was stopped for a number of days. This could cause cavitation of the pumps when they are first restarted. The design solution is to bleed the gases from these points of entrapment using small lines equipped with vent valves. This bleed system is equipped with small solenoid valves which can be operated from the control room, so that venting these spaces can be a routine procedure before starting the pumps. The vented gases are conducted to the ullage maintenance system compressor intake manifold, where they are treated as if they had come from the ullage space in the TSU.

The large pumps involved in the Commercial Plant may require start-up precautions especially if the fluid is cold and viscous. Pump manufacturer's recommendations should be consulted.

The procedures are otherwise essentially the same as for the Pilot Plant (see Section 4.4.1.2).

3.4.1.3 Hot Startup

The highest efficiency will be obtained if the thermal charge condition of the TSU is determined before startup as well as the anticipated future operating mode and heat load. Reference is then made to a chart to determine how many heat exchangers should be used.

A checklist is then used if startup is to be done manually. Automatic startup will be provided for, but not attempted until manual control techniques have been perfected; the automatic starting sequence and timing can then be properly programmed into the control circuitry.

The flow modulating valves are positioned to their minimum flow position to reduce the starting horsepower when the pumps are activated. Once fluid circulates, steam is allowed to enter the charging heat exchangers.

The system can soon be switched to automatic mode. However, the only active pieces of machinery in the TSS are the fluid pumps. As long as precautions are taken not to damage them, no catastrophic failures will occur in whatever sequence the system is started. The only serious precautions to be taken are in regard to the pressure level in the ullage space above the fluid in the TSU. However, this system must operate continually, even when the TSU is in a stopped or standby condition. Thus, it should not be necessary to change anything as regards the automatic controls of the UMU during startup or shutdown or at any time, except that the UMU should be monitored, maintained, and operated properly at all times regardless of operating modes or changes in operating modes.

3.4.2 Charging Mode

A schematic of the plant under normal charging operating conditions is shown in Figure 3-19. Steam from the receiver feeds the turbine and charges the TSS. The bypass flow charging the TSS is first desuperheated to avoid overheating the thermal storage medium. It then passes through the TSH and is eventually returned to the receiver. The thermal storage fluid is extracted from the bottom of the thermal storage tank and after being heated, is introduced through a manifold into the top of the TSU tank.

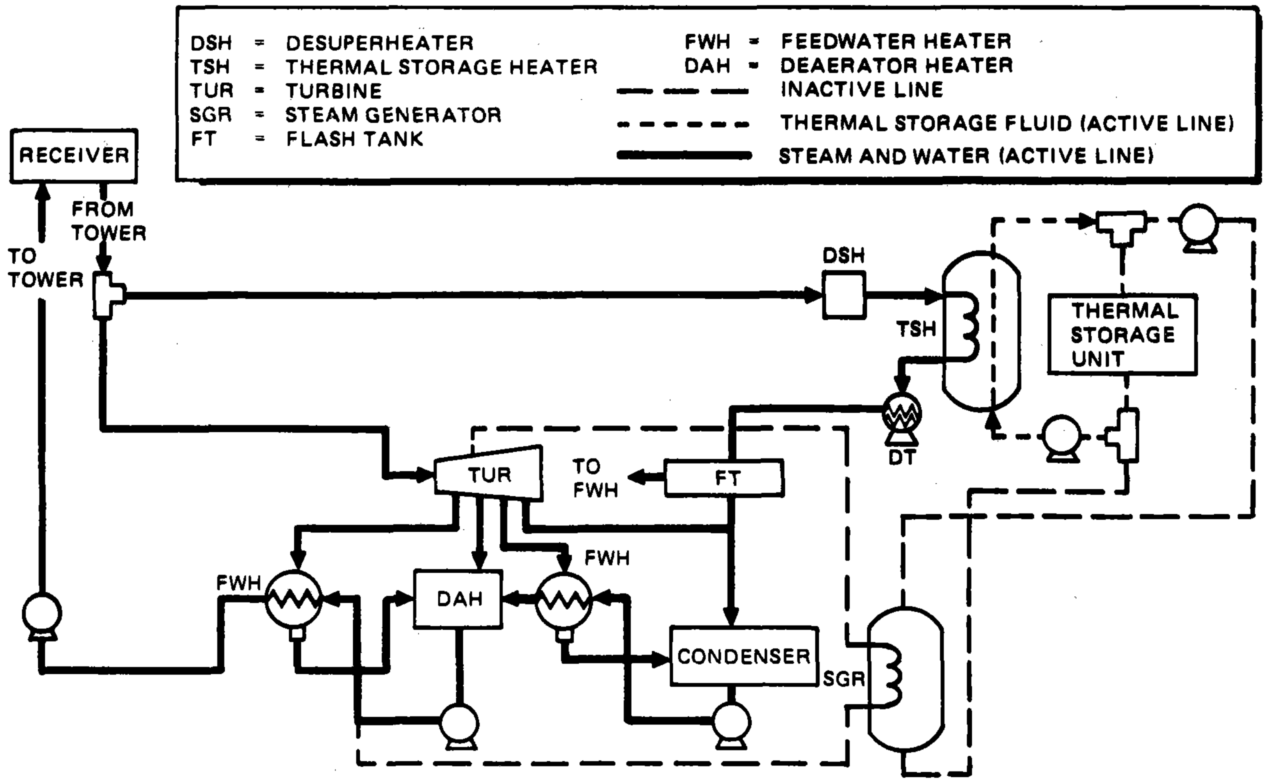

Figure 3-19. Plant Schematic for Normal Charging Operation

Operation with reduced solar power is indicated in Figure 3-20. When solar power available from the tower is reduced below the nominal value, the bypass flow that charges the TSS is reduced to maintain constant turbine inlet conditions.

The TSS is designed to absorb a maximum of 255 MWt of energy down to the 12.5-MWt rate (a turn-down ratio of 20.4 to 1). This energy enters the TSS in the form of superheated steam and represents the independent variable while in the charging mode. The TSS is designed so that it can accept any steam flow rate input within the rated range. The rate of change of steam flow input is limited to the ability of the desuperheater controller to follow this rate of change and to ensure that the temperature downstream of the desuperheater does not exceed or fall short of the design temperature of 360°C (680°F). The control system is designed to maintain a flow of heat-transfer fluid into the top of the TSU at a constant temperature independent of the thermal flow variations of steam at the input. Further details are given in Section 4.4.2.

3.4.3 Discharging Mode

Figure 3-21 shows the TSS operating loops in this mode. Operating personnel will prepare in advance to commence a discharge operation. The time required to build up a useful head of steam after the thermal discharge pumps have been started will be considered to estimate the exact time when the discharge is to commence, so that the steam will be ready at the proper pressure and temperature when required by the turbine. Upon receiving a go-ahead signal from the central controller, the operator will start the thermal storage extraction pumps, which will pull fluid from the top manifold in the thermal storage unit and through the thermal storage superheater fluid inlet valve which controls the fluid flowrate. The fluid flow is then split into two flows by a three-way valve. Part of the flow is conducted to the exit of the thermal storage superheater where the two flows recombine to enter the thermal storage boiler. Downstream of the boiler, the fluid flows through the thermal storage preheater and back to the bottom of the TSU.

At the same time, water from the high pressure condensate pumps is led through the thermal storage preheater water control valve into the preheater, where its temperature is brought up near to that of the temperature of the

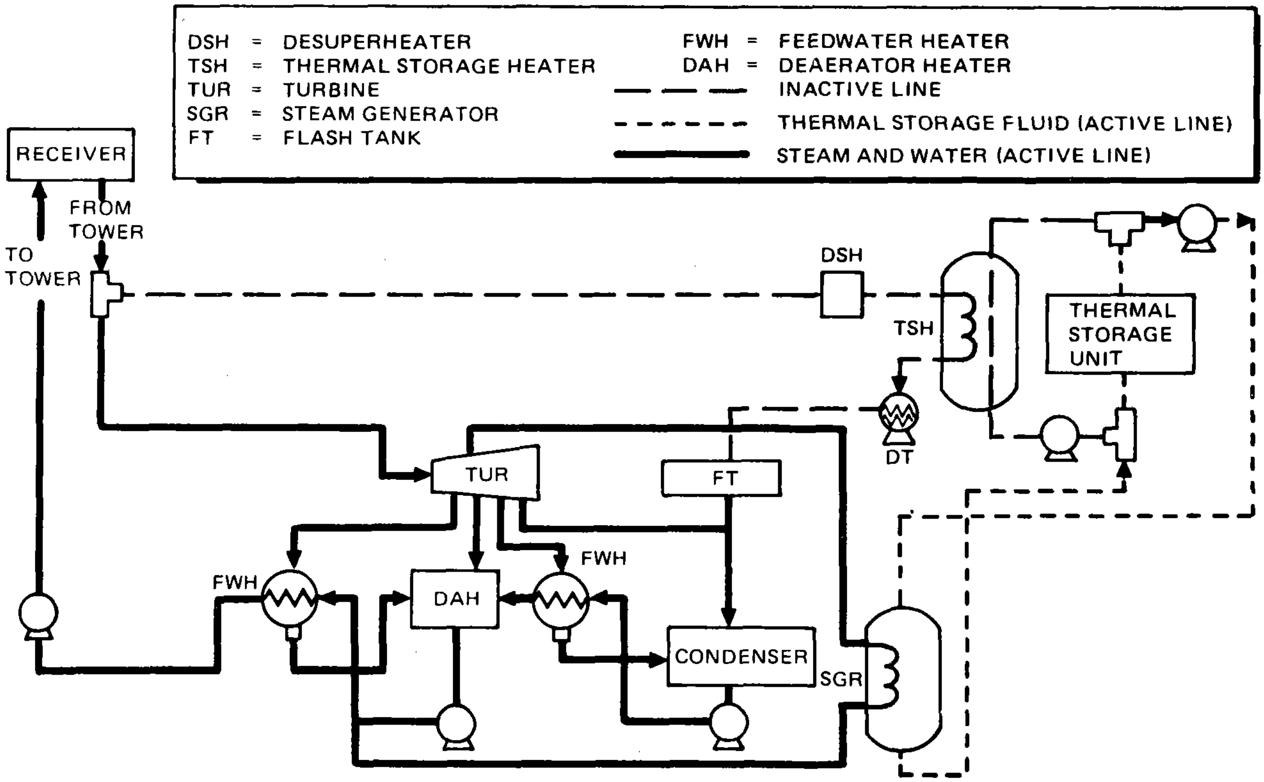

Figure 3-20. Plant Schematic for Low-Solar-Power Operation

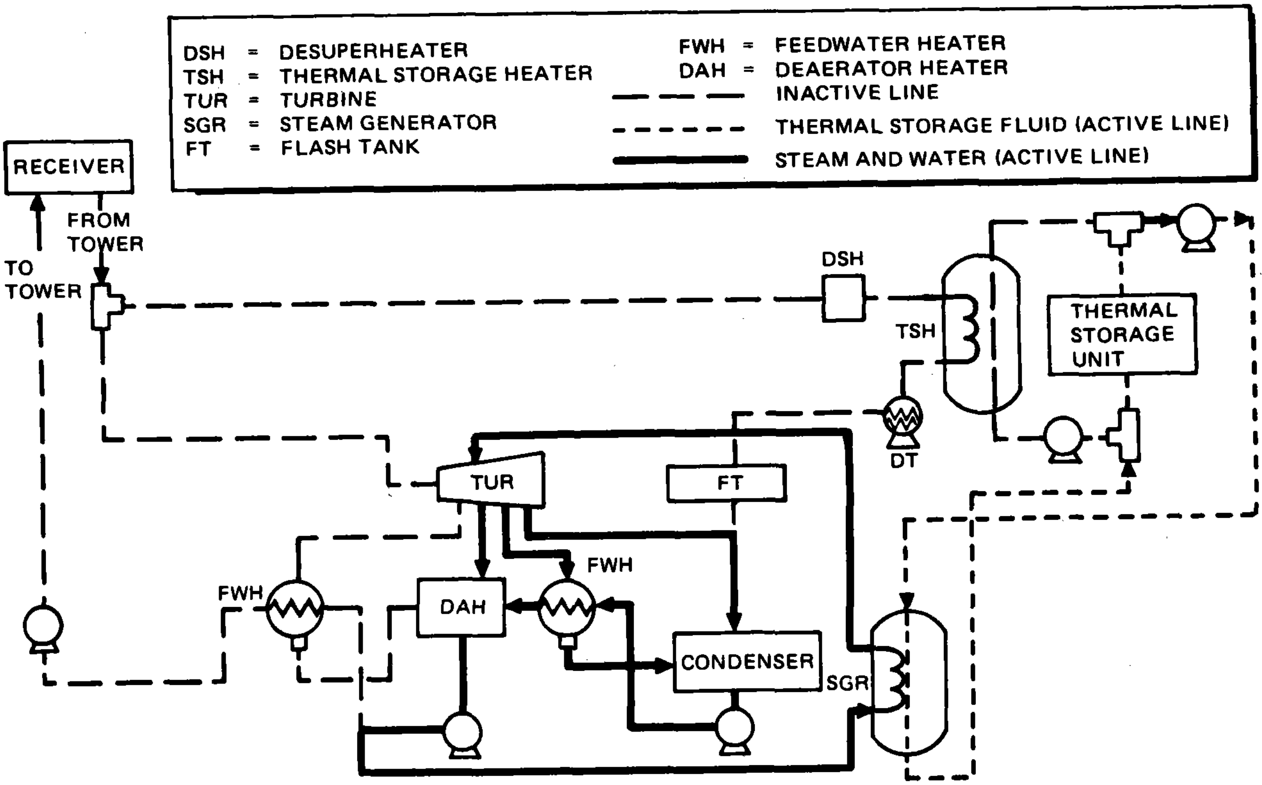

Figure 3-21. Plant Schematic for Discharge Only

water in the boiler. The water then travels into the boiler and is evaporated by the heat from the fluid traveling in the opposite direction through the tubes in the boiler. The preheater water control valve is modulated by the boiler water level measuring and control system to maintain a constant water level in the boiler. The steam generated then is conducted into the thermal storage superheater where its temperature is raised to 299°C (570°F). The superheated steam then exits this part of the system and is conducted to the low-pressure steam throttle on the turbine.

As the heat extraction proceeds, the charge monitoring system will indicate to the operator at regular intervals the charge state of the TSU. This indication will be very valuable from an operational standpoint in that the required changes in operating mode can be anticipated.

3.4.4 Intermittent Cloud Mode

When substantial variations in insolation are encountered, due to intermittent clouds, the operating mode shown in Figure 3-22 is employed, In this mode the TSS will be charged at the maximum rate. Operation of the generating plant can be maintained at approximately 70% of rated capacity, using some of the energy stored within the TSS. This operating mode requires a turbine capable of accepting two admission conditions.

When the collector field experiences intermittent cloud conditions, the turbine is switched from the receiver output to the TSU output. The operations which take place in the TSS are then a combination of those described in the previous sections under Charging Mode and Discharging Mode. The function of the components in the charging side of the TSS are completely independent of those in the extraction side of the system. They are, of course, connected at the TSU, but the TSU functions to isolate the two systems from each other because the TSU has such a high capacity compared to the flowrates in the two sides of the system. To illustrate this, it can be imagined that the system is initially in the charging mode, receiving full power from the receiver and all of the energy is being stored in the TSU. The extraction side of the system then can be started at any time and energy can be extracted from the TSU. However, upon closer analysis it can be seen that the heated fluid coming from the TSHs can pass directly into the extraction heat exchangers and the

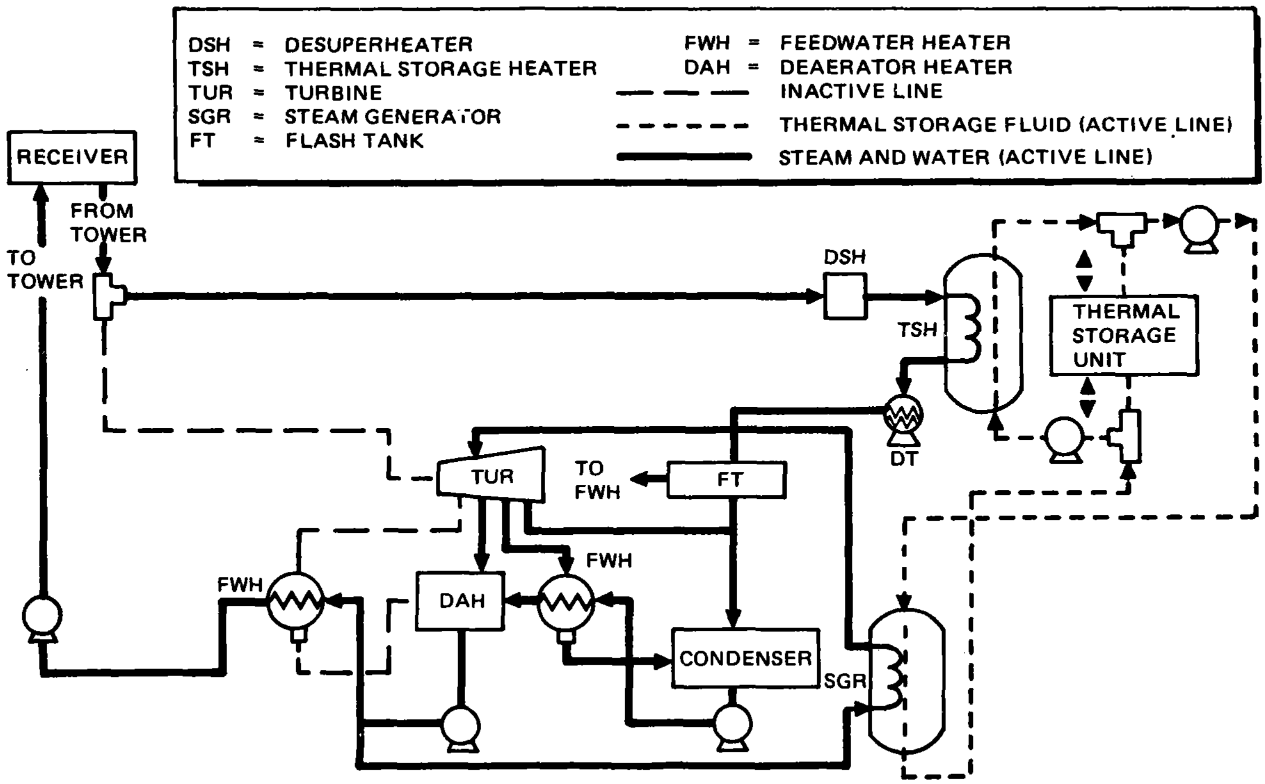

Figure 3-22. Plant Schematic for Operation with Intermittent Clouds

cooled fluid return directly to the charging heat exchangers without necessarily going into the TSU. The TSU automatically takes any differential which happens to exist in these two sets of flows. It acts as a buffer between a highly varying thermal input from a mirror field subjected to intermittent cloud conditions, and stands between this varying thermal source and the requirements of the turbine which are essentially a constant input of high-pressure, high-temperature steam. Thus, when the sun illuminates the collector field, the TSU itself is relatively idle since most of the energy is passed directly by it. When clouds obscure the sun, the TSU supplies the total energy to keep the turbine running at derated conditions. If the insolation is high most of the time, the TSU will slowly accumulate any excess energy not used by the turbine. If the insolation is lower, the TSU will be discharged slowly. Since the charge condition can be monitored easily, an impending, totally discharged condition can be anticipated and the cognizant personnel warned that the system may have to be brought down to a standby power condition. More details are given in Section 4.4.4.

3.4.5 Shutdown/Standby Modes

The definition of TSU shutdown is one in which the thermal input and the thermal output are both reduced to zero. Since the unit is meant to store energy it can never really be shut down in the normal sense of the word. For it to cool off in a fully charged condition would take many, many days, unless energy were actively extracted from it. The shutdown procedures are thus mainly limited to decreasing the charging and/or discharging rates to zero from whatever the values are when the command is received to shutdown. For additional details, see Section 4.4.5.

3.4.5.1 Shutdown Following a Charging Operation

Immediately after the decision to shut down is received, the steam from the receiver to the desuperheater will be decreased and ramped down to zero. The desuperheater controller and the charging controller will respond to this decrease in receiver steam supply by throttling down the water to the desuperheater and the fluid flow through the TSH's, with the objective of maintaining the temperature of the thermal storage fluid at the output of the TSHs at the design value of 316°C (600°F). When the turndown ratio of approximately 20 to 1 is reached, the steam supply from the receiver will be closed completely, the fluid charging pumps will be stopped, and the entire system will come to rest.

The only systems which will remain active at all times are the UMUs and, at various times, the fluid maintenance system. The ullage maintenance system must remain active constantly since the pressure in the ullage space must be kept within close limits, slightly above the outside atmospheric pressure, as described in Section 4.4.5.

3.4.5.2 Shutdown Following an Extraction

If the TSS is in the extraction mode when the command for shutdown is received, it is possible to stop the flow of fluid almost immediately by stopping the power to the pumps. A small amount of steam will still be generated due to the heat given off by the fluid which has been trapped in the tubes within the superheater, the boiler, and the preheater; however, this quantity of steam should be small.

3.4.5.3 Hot Standby Mode

The hot standby mode is one of three standby modes which can be defined; the others are idle mode and nighttime mode. The hot standby mode is one in which the charging and/or extraction heat exchangers are maintained in a hot condition so that they can be brought on-line immediately to accept steam from the receiver or furnish steam to the turbine. Energy from the TSU is used to maintain the extraction heat exchangers in the hot standby mode. This heat is furnished to the heat exchangers by a small pump which extracts fluid from either the top of the intermediate seal steam manifolds of the TSUs and pumps it through the discharge heat exchangers and then back into the bottom of the TSU. Bleed flow from the receiver downcomer can be used to maintain heat in the TSHs, if desired.

3.4.5.4 Standby Idle Mode

The standby idle mode is the condition of the TSU if for any reason the entire plant should be shut down for an extended period. Some provision must be made in such a case to provide for enough ullage maintenance gas in the form of pressurized nitrogen to maintain the pressure in the ullage spaces within the prescribed limits. Should the TSUs be required to be in the idle mode for days or weeks, the temperature within the tanks will drop very slowly as a result of heat loss through the insulation.

3.4.5.5 Nighttime Standby Mode

This mode is used at night after the main turbine has been placed in a standby mode, and a supply of seal steam is required to maintain a positive pressure in the turbine and to main the turbine seals at an elevated temperature. This is a substantial heat rate; 0.432 MWt for the Commercial Plant. Since a steam supply temperature of only 135°C (275°F) is required, it is not necessary to use high-grade heat for this purpose. Accordingly, heat-transfer fluid from the cool side of the TSU is used. Fluid from a special manifold a few feet above the main bottom manifold is pumped from the TSU's and into one of the steam generators (boiler and superheater section), after which it is returned to the bottom of the TSUs.

3.4.6 Emergency Modes

Emergency conditions in subsystems external to the TSS may impact the TSS, e.g., causing a sudden stop of thermal power to the TSS, or forcing the TSS to shut down rapidly. In general, the TSS lends itself very well to dealing with these sudden changes of steam input and/or power output requirements. No unusual problems are anticipated in meeting these requirements because all the components are passive except for the pumps.

Similarly, should an emergency occur in which the electrical power to the TSS should be severed, shutting down all pumps and allowing all valves to return to their "normal" positions, then it has been determined that no dangerous conditions will result, as long as the TSS is not physically damaged in such a way as to allow fluid to spill out onto the surrounding ground. Should spillage occur, the main danger is one of fire. A fire system has been provided, as well as an underground drainage sump into which fluid will flow naturally by gravity. If this fluid should be burning, it will be snuffed out as it enters the underground sump area due to lack of oxygen.

Should electrical power be cut off from the unit, then the entire system will be allowed to cool down gradually. The gases in the ullage space will contract and gaseous nitrogen will be required to fill this space in order to avoid collapse of the tank (due to a partial vacuum which would otherwise be created in the ullage space). This nitrogen gas supply will operate satisfactorily without the need of any kind of electrical power, since the nitrogen is stored under high pressure in a bottlebank and the gas is allowed to enter the ullage space under the control of a simple vacuum breaker valve.

Emergency conditions could be caused within the TSS, itself, if there were a massive failure of critical components or assemblies, e.g., a major piping or vessel failure, releasing hot fluid, or over- or under-pressurization of the TSUs. Such failures are extremely unlikely, and careful design and operational provisions effectively eliminate such events. However, if such a failure should occur, the subsystem has been designed to contain and minimize the effects of such a failure.

3.4.7 Transitions Between Operating Modes

This subject is considered to be the same for the Pilot Plant as for the Commercial Plant except for the numerical values of the time constants involved. Turn to Section 4.4.7 and Table 4-32.

3.5 REFERENCES

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3.2 System Analyses and Design. CDRL Item 1: Pilot Plant Preliminary Design Baseline Report, Volume 1, Book 2, McDonnell Douglas Astronautics Company, Report MDC G6040, UC-13, January 1976.

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Section 4
PILOT PLANT THERMAL STORAGE SUBSYSTEM DEFINITION

4.1 FLOWDOWN FROM COMMERCIAL PLANT

4.1.1 Design

Even though the heat storage capacity and maximum extraction and charge rates all are about an order of magnitude higher for the Commercial Plant than for the Pilot Plant, the same basic design concept is used for both plants. In both cases, a multidisciplinary approach is used in which fluid mechanics, heat transfer, structural, and control considerations are combined in an optimization of the thermal storage subsystem from the standpoints of performance and economics.

Existing technologies and commercial sources of supply are used for all components of the thermal storage subsystem, divided principally into (1) the heat storage tank, (2) the charging loop with its thermal storage heater, and (3) the extraction loop with its steam generator. The 10:1 capacity leads to designs making use of "building block" or modular components, with smaller numbers of components in the Pilot Plant and larger numbers in the Commercial Plant.

One constraint on component size stems from the limitations of transportation by rail, truck, and airplane. For instance, a rail car accommodates, roughly speaking, items up to 8 ft in width. Accordingly, any piece of equipment exceeding allowable dimensions must be constructed in pieces for assembly at the plant site. An exception to this may be the erection of solar power plants on the shores of bodies of water accessible to barge traffic which could carry freight of larger dimensions.

The more sophisticated items in the charge and extraction loops, such as pumps, heat exchangers, and steam generators, are fabricated in a shop where control of dimensions and tolerances to any desired degree is possible, and then transported intact to the site. Field construction is not an option. From a practical economic and logistic standpoint, extremely large shop fabricated components, for instance, pumps, are to be avoided in view of specialized and costly manufacturing methods and excessively long lead-times. Thus, there is a size constraint for shop-fabricated items.

The particular sizes of equipment and the 10:1 charge/discharge rate ratio from Commercial Plant to Pilot Plant results in standardization of pumps and heat exchangers at limiting sizes. For instance, where the Pilot Plant uses two heat exchangers in parallel for the thermal storage heater, the Commercial Plant uses five in parallel, each unit being of a size somewhat larger than that in the Pilot Plant. Such multiplicity of heat exchangers has become standard practice in the chemical processing industry, the largest user of high-temperature and otherwise sophisticated heat exchange equipment. The beneficial effects on operation of the use of multiple components in parallel include "floating design point" performance optimization (by suitable variation of the number of on-stream units and their operating conditions) and enhanced reliability (ability to maintain and repair units not on-stream).

The design of the thermal storage fluid piping in the charge and extraction loops is based on the usual considerations of minimum cost, involving an optimization from the standpoint of pressure drop, parasitic pump power and thermal insulation. The Pilot Plant piping system will serve as a model for the Commercial Plant piping. The Commercial Plant, of course, will have heat exchangers and steam generators in parallel, manifolded into larger piping which, in turn, will duct the fluid to and from the multiple heat-storage tanks through manifolds.

A similar but different limitation on size exists for components which are suitable for field construction, namely, comparatively thin-walled tanks and sheet metal components. The construction codes prevailing for tanks subjected to hydrostatic pressure from contained liquids and/or solids (e.g., API Standard 650) specify a limiting thickness for a steel plate, generally of the order of 4.4 cm (1.75 in). This, presumably, reflects the technological limits on being able to produce good steel plate only up to some limiting thickness. For a given design tensile hoop stress in the walls of a cylindrical tank with its axis vertical, a constraint is created whereby larger diameters can only be achieved at the expense of smaller heights. In this case, too, several identical tanks in parallel are dictated for the Commercial Plant while a single tank is adequate for the Pilot Plant.

The design of the packed beds and associated fluid manifolds and tank insulation will be practically identical for the Pilot Plant and Commercial Plant. The gravel and sand particle sizes will be identical in all the packed beds for dual medium storage. The flow manifold hole diameters and hole distribution across the tank cross-section will also be similar for different sizes. In the packed bed and fluid distribution areas, the SRE testing provided confirmation of the original design concepts, and they are applied with confidence to any larger heat-storage system consisting of any number of tanks.

The overall effect is that the Commercial Plant contains a multiplicity of pumps and heat exchangers which are manifolded and connected to a multiplicity of thermal storage tanks. In the Pilot Plant, twin pumps and heat exchangers are used in conjunction with a single thermal storage tank. Because the unit capacities of tanks, pumps, heat exchangers, filters, etc. differ at most by a factor of 2 to 1, the experiences with the Pilot Plant during construction and operation will be directly applicable to the Commercial Plant. This is especially important for the thermal storage tank where the experience gained during construction and operation of the Pilot Plant may suggest improvements and further economies for the Commercial Plant.

4.1.2 Fabrication and Installation

The modular features of the Commercial Plant, discussed above, demand no advances in the state-of-the-art of shop and field fabrication. The field fabrication of the thermal storage tanks with their diameters in the range of 15 to 30m (50 to 100 ft) will consist of welding together steel plate, previously rolled to the proper radius of curvature, and of length and width commensurate with the space in the transport medium to the site. Concurrent with the erection of the steel tank, the gravel and sand for the packed bed will be loaded with appropriate techniques to ensure the required void fraction. The packed bed inside the thermal storage tank contains three fluid manifolds, as well as penetrations for temperature instrumentation at various levels in the tank. The prccedure will be to interrupt the loading of the tank with solids at the appropriate levels in order to install these items, care being taken to maintain as uniform as possible a bed packing around the penetrations. Field cranes will set the tank roof or roof sections in place, after which the final step will be the application of the external insulation on the sides and roof of the tank. All these construction and installation procedures will be identical for the Commercial Plant and Pilot Plant.

The construction and installation of the interface equipment — thermal storage heater, steam generator, instrumentation and controls — will be scheduled concomitant with that of the steam loops and plant control system. The charge and extraction flow loops will be inspected completely to ensure that no openings or leaks exist. Then the heat-storage tanks and piping in both loops will be filled with the heat-transfer fluid at ambient temperature.

4.1.3 Operation

The operation of the Pilot Plant parallels the operation of the Commercial Plant in both the steady-state and transient portions. Startup, switching between operational charging and extraction modes, and shutdown are all similar in the two plants. The main difference between the plants, discussed in Section 4.1.1, is that all fluid flow and heat-transfer components in the Commercial Plant are made up of a multiplicity of units in parallel, whereas in the Pilot Plant only one unit or two units in parallel are used.

Because of the parallel unit feature it is possible in the Commercial Plant to vary the number of units onstream as well as their individual loading to produce a desired operational condition. Previous transient exercise of the corresponding components of similar unit capacity in the Pilot Plant, one or two at a time, will permit estimation of the various transient performance factors — time constants and delay times in the various modes — for the Commercial Plant.

Events in the heat storage tank depend on fluid velocity and bed packing, i.e., on strictly vertical variables, essentially independent of the diameter of the tank. Therefore, the charging and extraction performance of the tank in the Pilot Plant should be a good indication of the performance of the tanks in the Commercial Plant whose fluid velocities are in the same range but whose diameters are larger. This statement holds true whether one or more of the four storage tanks in the Commercial Plant are working in parallel.

Another operational area in which Pilot Plant experience applies to the Commercial Plant is the area of maintenance. The multiple flow loop components – pumps, valves, pipes and filters – will be closely the same in both plants except for differences in size. The maintenance history of the Pilot Plant components will provide data for the Commercial Plant. Of even greater importance are the Pilot Plant experiences with the operations affecting the heat transfer fluid: (1) ullage maintenance unit, which uses gaseous nitrogen, and (2) fluid maintenance unit, which uses heat and power to purify the fluid after degradation in use. The operation of these units will determine the actual rates of fluid degradation in diurnal cyclic use, the cost-effective purification steps, and the percentage of the heat-transfer fluid inventory which will need to be replaced on a continuous basis. The experience with the Pilot Plant will determine the logistics of fluid replenishment in the Commercial Plant.

4.2 SUBSYSTEM REQUIREMENTS

The thermal storage subsystem "buffers" the electrical power generating subsystem (EPGS) from excessive variations in insolation, and extends the plant's generating capacity into periods with low or no insolation. The general requirement on the thermal storage subsystem is that it provide a means of transferring to stored thermal energy a portion of the thermal output from the receiver subsystem and subsequently transferring stored thermal energy to steam in a form suitable for generating electrical power with the conventional turbine-generator in the EPGS. More specific requirements for the Pilot Plant thermal storage subsystem are given in the following subsections and in Appendix A.

4.2.1 Fluid Conditions

There are five fluid streams crossing the boundaries of the thermal storage subsystem, all water or steam flows which enter or leave the subsystem. These five streams are shown on Figure 4-1. The required fluid conditions and flow rate ranges for each of these five streams are summarized in Table 4-1.

It can be seen from Table 4-1 that there are two different sets of conditions required for exit steam from the steam generator, corresponding to: (1) steam to be supplied to the turbine for power generation, and (2) the much lower flow rate, lower temperature conditions of supplying nighttime steam for maintaining equipment seals.

All other fluid conditions and flowrates are derived from the requirements for the five basic streams, given in Table 4-1. Although not a requirement, per se, it was established early that the heat-transfer fluid in the extraction loop would cycle between 218°C (425°F) and 302°C (575°F).

4.2.2 Performance

The thermal storage subsystem is required to have an extractable storage capacity of at least 103.8 MWHt, which is composed of 7.5 MWHt to provide a turbine hot start and 96.3 MWHt to permit the turbine-generator to supply 7 MWe net (7.8 MWe gross) for 3 hr following turbine startup. This extractable capacity is to be available following a full charge and a 20-hr hold period. The required charging rates are 1.5 MWt (rated steam operation) to 30 MWt (derated steam operation). Required discharging rates are 3.1 to 32.1 MWt. The maximum allowable heat loss is 3 percent of extractable capacity in 24 hr, starting in a fully charged condition. The subsystem is required also to provide nighttime seal steam at a temperature of at least 149°C (300°F) and at a rate of 0.33 MWt for approximately 16 hr.

4.2.3 Operational

The subsystem is required to operate stably and safely in all normal and emergency operating modes. The major operating modes are: startup (both cold and hot), charging, discharging, intermittent cloud, shutdown, standby,

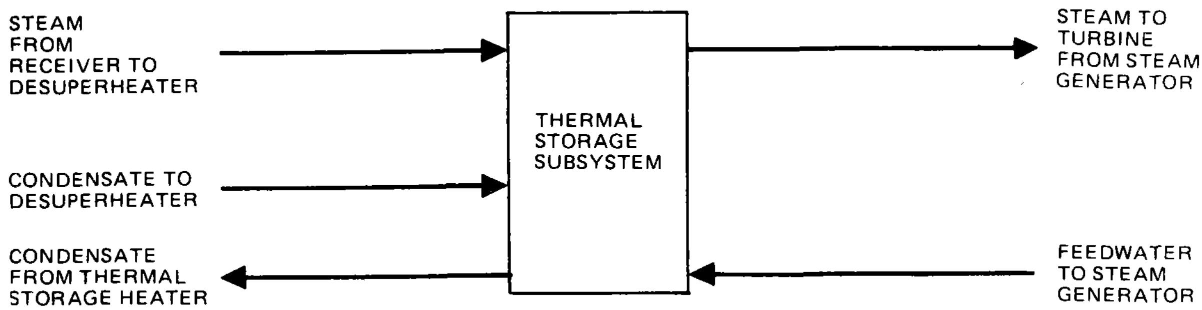


Figure 4-1. Process Streams Crossing TSS Boundaries

Table 4-1

PROCESS STREAM REQUIREMENTS FOR
PILOT PLANT THERMAL STORAGE SUBSYSTEM

Process Stream	Case 1: Max Charge with Rated Steam	Case 2: Max Charge with Derated Steam, including intermittent cloud operation	Case 3: Max Extract Energy from
Receiver steam to desuperheater inlet:			
Temperature, °C(°F)	510(950)	343(650)	No flow
Pressure, MPa(psia)	10.1(1465)	10.1(1465)	
Maximum flow, kg/hr(lb/hr)	47,300(104,000)	59,200(130,500)	
Minimum flow, kg/hr(lb/hr)	2380(5230)	2970(6525)*	
Condensate to desuperheater:			
Temperature	211(412)	No flow	No flow
Maximum flow	12050(26,500)		
Minimum flow	590(1295)		
Condensate from thermal storage heater:			
Temperature	246(475)	246(475)	No flow
Pressure	9.7(1400)	9.7(1400)	
Maximum flow	59,200(130,500)	59,200(130,500)	
Minimum flow	2970(6524)	2970(6525)*	
Feedwater to steam generator:			
Temperature	No flow	No flow	121(250)
Pressure	No flow	No flow	2.90(420)
Maximum flow	No flow	No flow	47,500 (104,700)
Minimum flow	No flow	No flow	4580(10,100)
Steam from steam generator:			
Temperature	No flow	No flow	277(530)
Pressure	No flow	No flow	2.76(400)
Maximum flow	No flow	No flow	47,500 (104,700)
Minimum flow	No flow	No flow	4580(10,100)

*Not limiting case for design

and emergency modes. These modes and operating characteristics are described in detail in Section 4.4.

4.2.4 Interface

The thermal storage subsystem has interfaces with the receiver subsystem and the electrical power generation subsystem (EPGS). The requirements in terms of process flows and conditions are given in Section 4.2.1. Additional physical requirements are that the piping, connections, and mounting fixtures will match those of: (1) the receiver downcomer, (2) the EPGS at the entrance of the turbine automatic admission port header, and (3) the outlet of the EPGS condensate loop.

Not only must the Thermal Storage Subsystem be controllable by manual commands from a control room operator, the subsystem must also provide information to and be responsive to automated commands from a computerized Master Control Subsystem. There are two pairs of major control commands: (1) start/stop energy storage, and (2) start/stop steam generation from stored energy. In addition to these major commands from master control, the interface must provide for throttling controls imposed by master control on the basis of subsystem status measurements sent to master control, and based also on variations in steam and water flow rates imposed on the subsystem by the interfacing subsystems. The subsystem control must respond to such flow rate modulations by metering heat-transfer fluid flows and subsystem functions. The TSS is required also to accept override commands, including complete control, by master control.

4.2.5 Environmental

The environmental requirements on the TSS are identical to those imposed on the total plant; these requirements are given in Volume II. The primary requirements which are of significance in design of the TSS are the earthquake forces, and the ambient temperatures and wind velocities (the latter two of primary importance in calculating heat losses and insulation requirements). Design requirements for horizontal earthquake loadings are 0.165g operational and 0.33g safe shutdown; vertical components are two-thirds of horizontal components. Temperature extremes for survivability are -30° to 60°C (-22° to 140°F). Conditions for heat loss calculation are 28°C (82.6°F) ambient temperature, 3.5 m/s (8 mph) wind speed at 10m elevation, and a velocity profile with velocity in m/s equal to $3.5(H/10)^{0.15}$, with height H in meters.

For design purposes, safety margins will be used that are commensurate with availability and performance requirements to ensure operation during and/or after exposure to the environmental conditions, as appropriate, for the 30-yr life of the subsystem.

4.2.6 Structural

All critical components of the TSS must be designed and installed such that the environmental and site conditions described in Section 4.2.5 do not include a dynamic environmental condition which exceeds the structural capability of the component. All components must be designed to withstand handling and hoisting inertial loads, as applicable, during fabrication, transportation, installation, and maintenance.

The thermal storage unit is required to be designed in accordance with API Standard 650 as modified by the ASME Boiler and Pressure Vessel Code, Section VIII for elevated temperature operation. All heat exchangers are to be designed, fabricated, and inspected in accordance with the ASME Boiler and Pressure Vessel Code, Section VIII. Piping is to be designed, fabricated, and inspected in accordance with the American National Standard Institute Code for Pressure Piping, ANSI B31.1.

Materials of construction throughout are to be selected to ensure compatibility with the process fluids at the maximum operating conditions.

4.2.7 Miscellaneous

The TSS is required to be designed to minimize safety hazards to operating and service personnel, the public, and the equipment. Electrical components must be insulated and grounded. All parts or components associated with elevated temperatures must be insulated against contact with or exposure to personnel. Any moving elements must be shielded to avoid entanglement, and safety override controls and interlocks must be provided for servicing. Isolation valves are to be provided on all major assemblies and on all interface utility lines to permit isolation and shutdown of assemblies and segments of the subsystem. Concrete and/or earth berms and dikes are to be provided to contain the maximum quantity of heat-transfer fluid which can be emptied from all above grade sections of the subsystem. Safety showers and eye-washes must be provided. Ladders, handrails, and platforms must meet OSHA standards. Fire-protection equipment must be provided.

4.3 DESIGN CHARACTERISTICS

Design characteristics are given and discussed in this section for the entire thermal storage subsystem, with a subsection devoted to each major assembly. Given for each major assembly are details of the requirements, design analyses and trade studies made in developing the preliminary design, and a description and discussion of the selected preliminary design.

Figure 4-2 is a process schematic diagram of the Pilot Plant thermal storage subsystem, showing all major assemblies and key process conditions. The second sheet of Figure 4-2 is a more detailed schematic, showing controls and all components. Drawing MDAC-075008-P-M1, in Appendix B of this volume, is the most detailed schematic.

4.3.1 Thermal Storage Unit

The function of the Thermal Storage Unit (TSU) is to act as a reservoir for solar thermal energy, which charges the TSU during hours of high insolation. At other times, when insolation is partially or completely unavailable, thermal energy is extracted from the TSU to produce steam for the electrical generation subsystem. This section deals with the requirements, design analyses and optimizations, and the design description of the TSU for the 10-MWe Pilot Plant. The nomenclature is in Section 4.3.1.4.

4.3.1.1 Requirements

The steady-state operating requirements to be met for energy charging and extraction are shown in Table 4-2. Design requirements for the TSU are listed in Table 4-3. The TSU design criteria for the 10-MWe Pilot Plant, developed to respond to the requirements of these two tables, are listed in Table 4-4. In keeping with the aim of building the TSU and associated equipment within existing technology, the maximum heat storage unit temperature

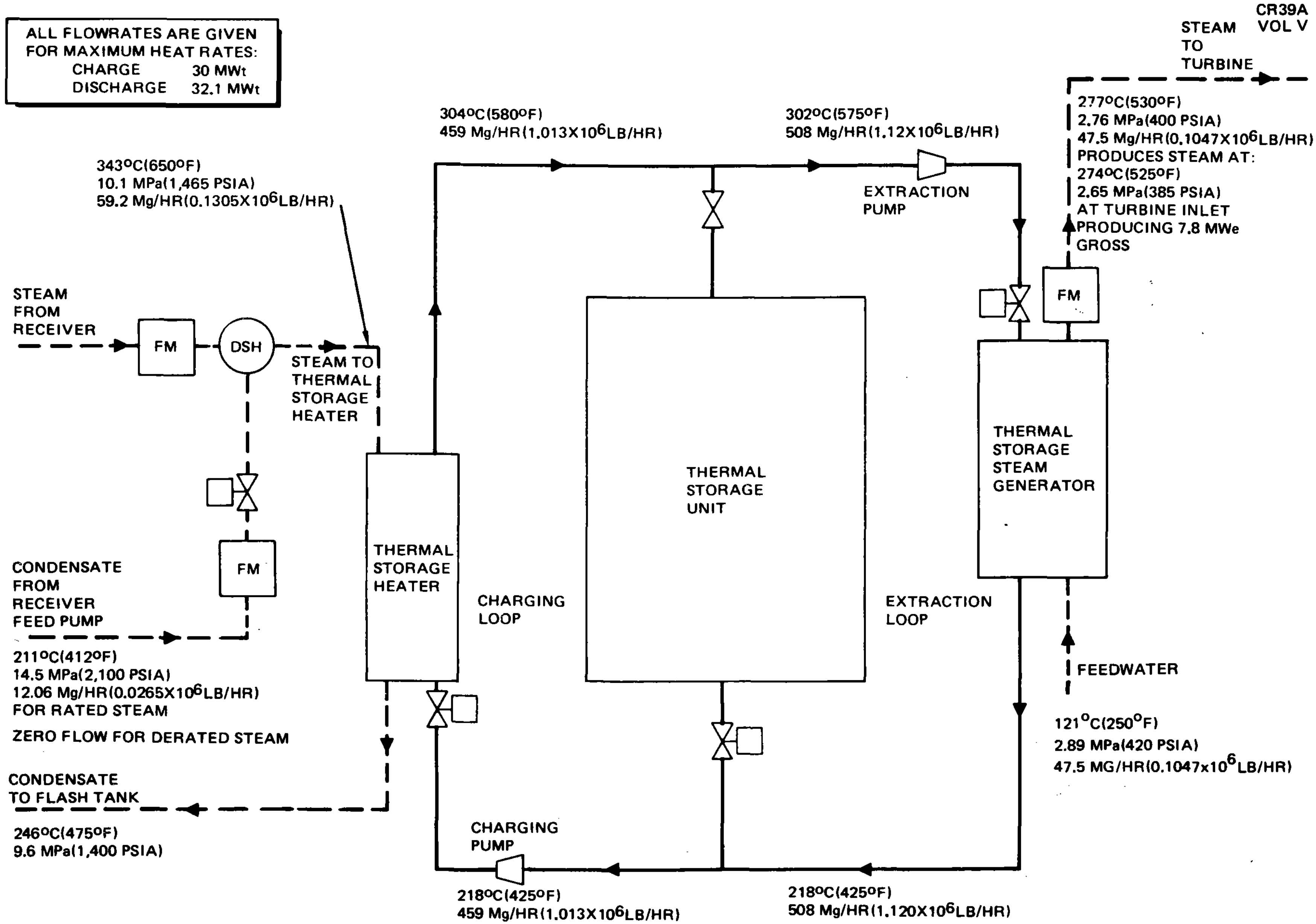


Figure 4-2 Schematic Flow Diagram of Thermal Storage Subsystem for 10-MWe Pilot Plant (Sheet 1 of 2)

THERMAL STORAGE SUBSYSTEM (TSS)

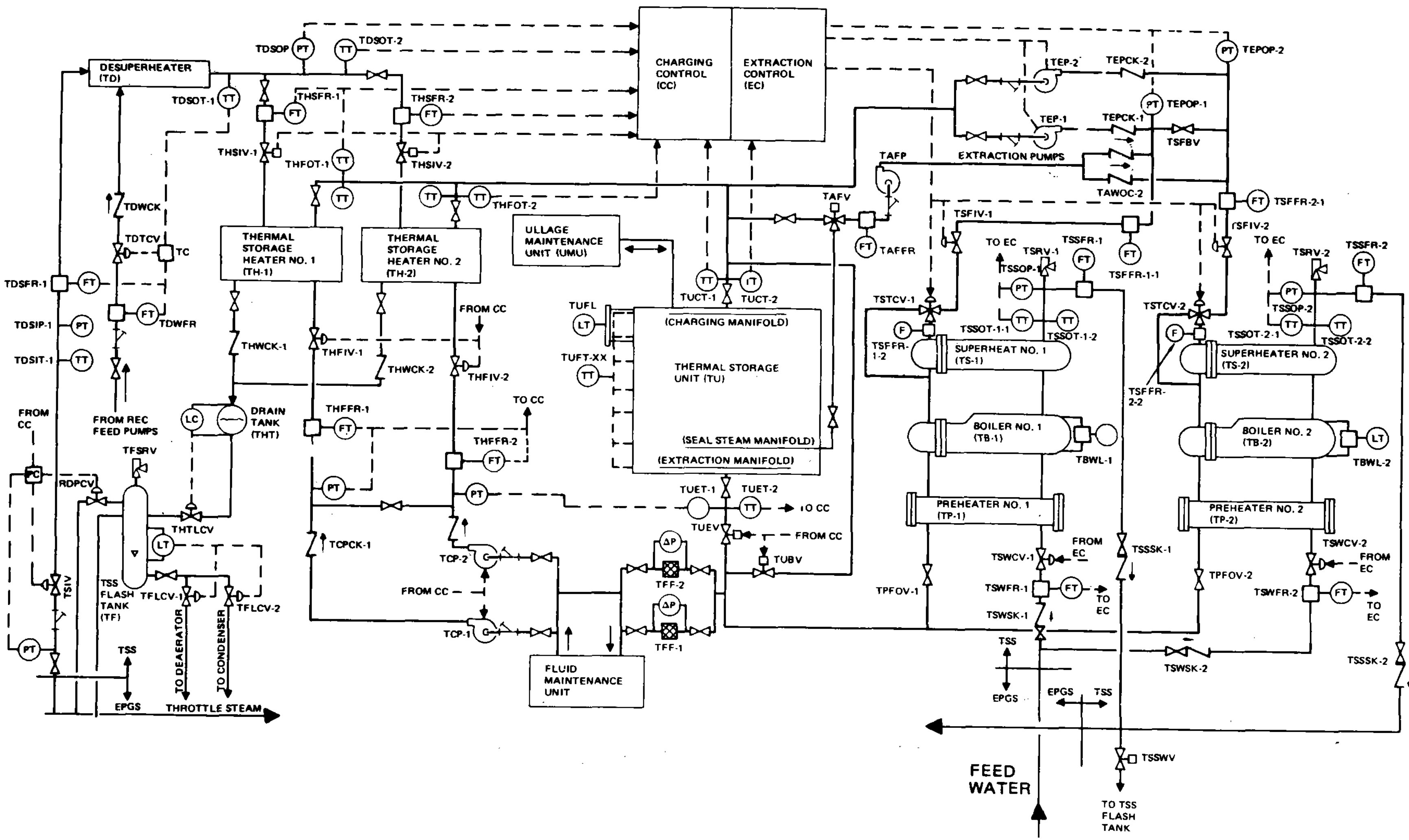


Figure 4-2. Thermal Storage Schematic (Sheet 2 of 2)

Table 4-2

10 MWe-PILOT PLANT THERMAL STORAGE REQUIREMENTS

- Design Storage Temperature: Max 302°C (575°F)
  Min 219°C (425°F)
- Extractable capacity after 20-Hr hold time:

Net	96.3 MWHt	(328.7×10^6 Btu)
Turbine Hot Start	7.5 MWHt	(25.6×10^6 Btu)
Total	103.8 MWHt	(354.3×10^6 Btu)

- Allowable degradation of thermal storage fluid temperature during extraction = 15°F (8.3°C), i.e., to 560°F (294°C)
- Losses during 24-hr hold: less than 3% of extractable capacity

		Charging	Extraction
• Thermal rates:	Max	30 MWt (102×10^6 Btu/hr)	32.1 MWt (109.6×10^6 Btu/hr)
	Min	1.5 MWt (5.1×10^6 Btu/hr)	3.1 MWt (10.6×10^6 Btu/hr)
• Time at max rate			3 Hr
• Hot standby demand, fluid at 302°-294°C (575-560°F)			0.02 MWt (68,260 Btu/hr)
• Nighttime seal steam, fluid at 219°-149°C (425-300°F)			0.33 MWt (1.11×10^6 Btu/hr)

Table 4-3

10-MWe PILOT PLANT TSU DESIGN REQUIREMENTS

- Design life (with routine maintenance) 30 yr
- Conditions for heat loss calculations

Ambient temperature	28°C (82.6°F)
Wind speed at 10m elevation	3.5 m/s (8 mph)
Velocity profile	$V_H = 3.5$ m/s $(H/10m)^{0.15}$

- Barstow soil bearing capacity

  1,500 psf (0.07 MPa) at 2 ft (0.61m) depth
  5,000 psf (0.24 MPa) at 5 ft (1.53m) depth
  10,000 psf (0.48 MPa) at 10 ft (3.05m) depth

  (More soils data in ERDA letter of 14 January 1977)

- Barstow annual rainfall: 4 in. (100 mm)

Table 4-4

DESIGN CRITERIA FOR TSU

- Cost-effective
- Use of existing technology
- Compatibility with all specified operating modes
- Selection of optimum storage media
- Structural design according to recognized vessel code(s)
- Consideration of thermal expansion tank and media
- Consideration of thermocline nonideality in packed bed
- Low pressure drop during fluid flow through bed
- Consideration of fluid flow nonidealities in TSU
- Inert gas ullage to prevent high-temperature oxidation of fluid
- Consideration of heat losses to environment
- Optimum use of insulation on tank walls

for the 10-MWe plant was chosen to be 302°C (575°F) even through the solid and liquid components selected are capable of steady-state performance at 316°C (600°F), a level to be attained in the 100-MWe plant. Additional requirements, in the field of safety, are contained in Ref. 4-1.

4.3.1.2 Design Analyses

Under this heading are presented the multidisciplinary considerations which, in combination, produced the proposed TSU design described under Section 4.3.1.3. These considerations are contained individually in subsections which are devoted (in order) to: dual-medium heat storage analysis, selection of dual heat transfer media, dual-medium thermal design, heat losses and insulation, fluid distribution and bed packing, and structural analysis.

The basic design concepts were those which appear in Ref. 4-2 and which were used in the construction of the TSU tested as a component of the SRE Thermal Storage System. Most of these concepts were validated in tests conducted by Rocketdyne during October-December 1976. As a result of the tests and because of the availability of additional information, certain design concepts were modified. The design analyses presented here, therefore, contain some material previously presented in Ref. 4-2 and some new material. In any case, they represent the current considered design rationale for the dual-medium thermal storage subsystem.

Dual-Medium Heat Storage Analysis

Thermal storage methods are often grouped into two categories: sensible heat and latent heat or phase change. A third category is sometimes added under thermal storage, namely reversible chemical reactions, which can include heats of solution and hydration. There has been considerable amount of work done on thermal storage; however, until the last few years, it was limited primarily to very-low-temperature applications (particularly heating of buildings) and/or tentative paper studies. Rocketdyne's continuing activity in the thermal storage field has included maintaining files of published information plus contacts with other current workers in the field. Some of the useful literature sources which summarize primary sources include the applicable section of ERDA's bibliography on solar energy, which covers material through 1975 (Ref. 4-3), an ERDA-funded "Survey of High Temperature Thermal Energy Storage" prepared by Sandia/Livermore and released last year (Ref. 4-4), recent survey papers by Swet (Ref. 4-5) on heat storage for buildings, Green, et al. (Ref. 4-6) on high-temperature storage, and Offenhartz (Ref. 4-7) on chemical methods of storage, and other survey reports of which Ref. 4-8 and 4-9 are good examples.

Some relevant conclusions from Ref. 4-6, which included a comprehensive survey of applications above 200°C, are given below (underlines added):

> "The concept of sensible heat storage represents the current state of thermal energy storage technology. <u>This type of storage system can be designed and engineered using available technology and materials with a relatively low risk.</u> Systems using water as the storage medium are limited in maximum temperature by the high vapor pressure, with the costs of these systems being greatly inflated by the need for high-pressure containment. The vapor pressure problem can be alleviated by the use of heat transfer oils or molten salts. These materials also offer the possibility of higher-temperature operation. If reductions in material costs can be realized, the oil or salt systems will become very attractive. The use of a solid storage material has the advantages of high-temperature operation, direct-contact heat exchange,

and very low material costs. Hybrid systems consisting of a packed bed of solid particles through which a heat transfer oil or molten salts is flowing have been shown to reduce significantly the cost of an all liquid system.

The continued development of heat-of-fusion storage systems currently needs identification of new, low-cost, high-temperature storage materials, development of high flow active heat exchangers, and design and construction of experimental storage systems for verification."

Rocketdyne has designed, constructed, and tested a thermal storage system based on the dual-medium, sensible-heat concept (patent pending) at the SRE level. This concept makes use of a low-cost stationary rock bed as the primary storage medium with a suitable liquid to transfer the heat in and out of storage (and to store part of the energy directly). This dual-medium type of system combines advantages of a low-cost solid with the flexibility, low pumping power, and moderate heat exchanger requirements of a liquid energy storage system.

Conceptually, in its simplest form, the system uses a bed (shown in the center of Figure 4-3) of an inexpensive solid (e.g., rock, ore, metal scraps, etc.). An appropriate high-temperature liquid fills the voids in the bed and flows through the bed to deposit or withdraw energy. Heat storage is effected in both the liquid filling the interstices between the solid bed particles as well as the solid particles themselves. The solid particles also assist the performance by inhibiting fluid natural circulation in the tank. By occupying approximately 75% of the thermal storage volume with an inexpensive solid, heat-transfer fluid costs are reduced nearly 75% below an all-liquid thermal storage concept.

In the cyclical operation, heating of the bed (charging) is achieved by removing lower temperature fluid from the bottom of the bed, heating it in a heat exchanger with the energy source fluid (shown in Figure 4-3 as steam from a solar receiver), and returning the fluid to the top of the tank. A fairly sharp, horizontal temperature divide (a thermocline) is maintained

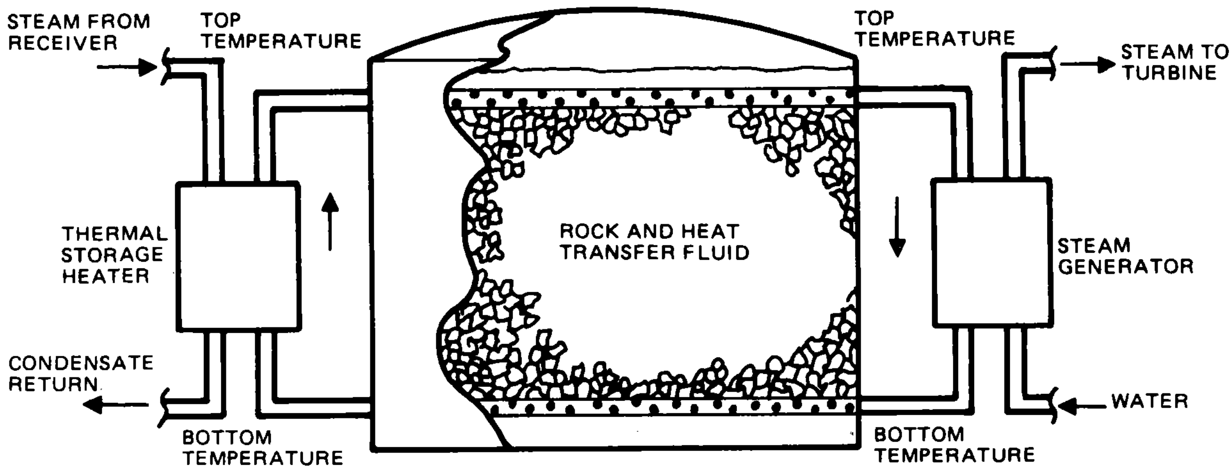


Figure 4-3. Dual-Medium Thermal Storage Concept

naturally between hot and cold fluid because of the lower density of hot fluid and the stabilizing effect of the solid bed. This thermocline moves downward through the bed from the top of the tank during charging and upward during extraction. When the storage unit is completely charged, all of the bed and the fluid is at the maximum temperature and the thermocline does not exist. The extraction loop uses the fluid to remove energy from the storage unit. This energy may be used to produce steam for power plant operation, or to heat feedwater or for other functions. The use of the thermocline principle further reduces subsystem costs by eliminating the necessity to have multiple tanks to segregate the hot and cold fluids.

The operating modes for the thermal storage subsystem discussed above involve cyclical heating and cooling of the bed, which defines a thermal regenerator for which theory was worked out some 50 yr ago, e.g., by H. Hausen and T. E. W. Schumann (See Ref. 4-10 and 4-11). Explicit analytical solutions were obtained for the temperature history in the fluid and in the solid by solving transport equations expressing the conservation of mass, momentum and energy, under several idealizing assumptions such as invariant fluid and solid properties. Table 4-5 gives a summary of the transport equations, together with initial and boundary conditions for heat storage and extraction, assumed to occur in unidirectional flow parallel to the axis of the packed bed. Also listed are the assumptions made by Hausen and Schumann. The symbols are explained in Subsection 4.3.1.4.

A complete model of a duel-medium thermal storage unit was developed by Rocketdyne, based on the Hausen-Schumann analytical solution. Subsequently, a variable mesh numerical integration method was developed at Rocketdyne in 1974 to treat more realistically solar energy thermal storage phenomena with high-temperature liquids exhibiting significant changes of properties with temperature, and also to permit investigation of successive storage and extraction cycles, including cases with nonuniform temperature over the charging or extraction periods. Special care was given to taking suitably small time and distance intervals in the vicinity of the thermocline, and the computation was set up as a digital computer program for convenient exploration of parametric changes. A large number of trade-offs and parametric design calculations were made for TSU's of various sizes.

Table 4-5 (Page 1 of 2)

DUAL-MEDIUM THERMAL STORAGE MODEL
(See Section 4.3.1.4 for Nomenclature)

1. Transport Equations for Thermal Charging and Extraction in Bed

Fluid Continuity: $\frac{\partial \rho_L}{\partial t} + \frac{\partial \rho_L v_L}{\partial x} = 0$

Fluid Energy: $f \rho_L C_{p_L} \frac{\partial T_L}{\partial t} + f \rho_L C_{p_L} v_L \frac{\partial T_L}{\partial x} = -h(A/V)(T_L - T_s)$

Solid Energy: $(1 - f)\rho_s C_{p_s} \frac{\partial T_s}{\partial t} = h(A/V)(T_L - T_s)$

Boundary Conditions: $T_L(0,t) = T_{L_I}(t)$; $V_L(0,t) = V_{L_I}(t)$
$X=0,\ t > 0$

Initial Conditions: $T_L(x,0) = T_{L_O}(X)$; $T_s(X,0) = T_{s_O}(X)$
$X \geq 0,\ t=0$
$V_L(X,0) = 0$

2. Assumptions in Energy Equations

- Negligible heat flow into or through tank walls
- Negligible axial heat flow through solid, fluid, and walls
- Negligible energy loss due to viscous heating
- One-dimensional flowthrough bed
  ($D_{particle}/D_{tank} << 1$, (Ref. 4-12)
- Uniform bed packing (all x, t)
- Constant cross-sectional tank area
- Infinite solid thermal conductivity

3. Methods of Solution

A. Schumann-Hausen Analytical Solution (Ref. 4-11)

- h, C_{p_L}, C_{p_s}, ρ_L, ρ_s not a function of temperature or velocity
- $T_{L_o} = T_{s_o}$ = Constant

Table 4-5 (Page 2 of 2)

DUAL-MEDIUM THERMAL STORAGE MODEL

- $V_L = V_{L_I} = \text{Constant}$
- $T_{L_I} = \text{Constant}$

B. Rocketdyne Variable Mesh Numerical Integration

- $V_L = V_{L_I} = \text{Constant}$
- Properties do depend on temperature and velocity
- Arbitrary boundary and initial conditions

Figure 4-4 shows heat extraction thermoclines in the TSU for a typical 10 MWe design over a 6-hr period, with cycling between 218° and 302°C (425° and 575°F). The thermoclines are seen to be steep and liquid can still be extracted at 302°C (575°F) after 5 hr of energy extraction; after 6 hr, the exit fluid temperature has dropped to 294°C (560°F). The temperature of the fluid leaving the TSU is shown in Figure 4-5. The exit temperature drop was limited to 8.3°C (15°F) i.e., 10% of the operating temperature amplitude. At the end of the 6-hr extraction period there is still some energy left in the tank (i.e., energy above the 218°C lower temperature). The TSU capacity must be sized to allow for this unavailable energy as well as other nonideal effects. This energy is "lost" only during the first charging cycle. During subsequent diurnal variations, the output energy is equal to input energy except for heat losses to the external environment.

The two main desiderata in dual-medium sensible heat storage are (1) to maximize the energy stored per unit volume, both in the liquid and in the solids, and (2) to minimize the bed void fraction, i.e., the fraction of the bed occupied by the liquid, since the cost per unit volume of fluid is an order of magnitude above that of solids.

In the solar application, the diurnal storage and extraction of heat takes place over a period of hours, so that even with heat transfer equipment of large dimensions (say, of the order of 30m (98 ft) the interstitial flow

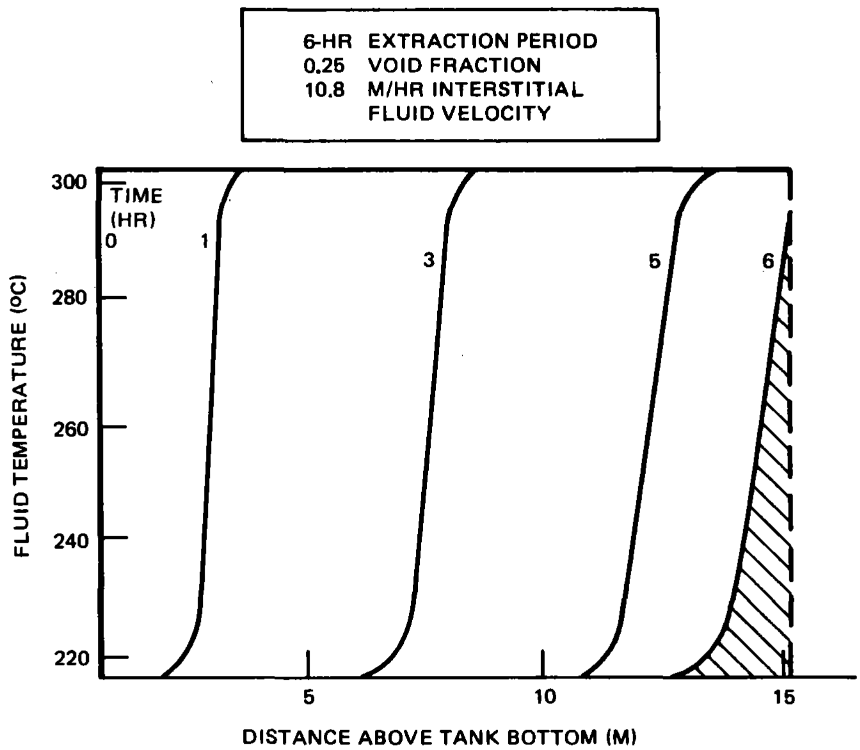


Figure 4-4. Energy Extraction Thermoclines for Typical 10-MWe Design

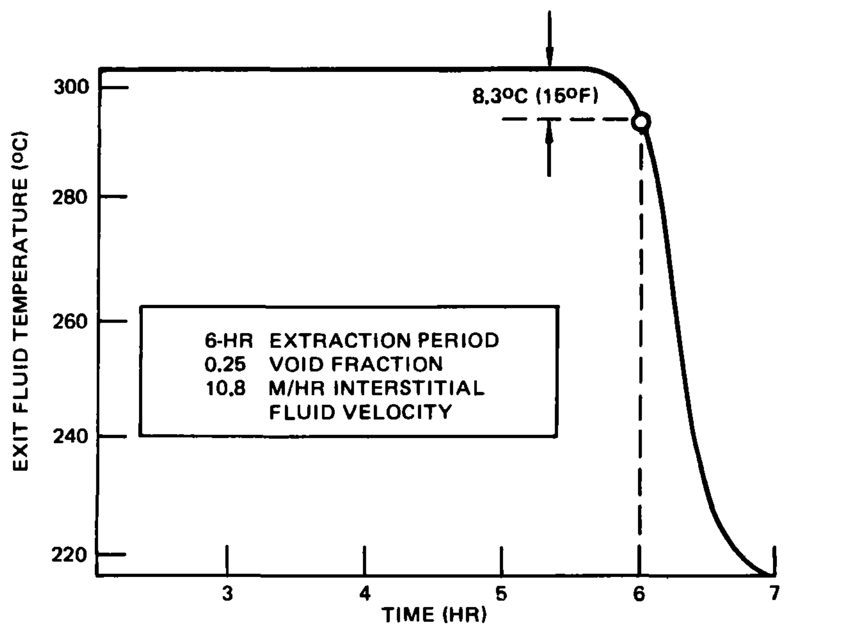


Figure 4-5. Fluid Exit Temperature During Typical 6-Hr Heat Extraction

velocities of a heat-transfer fluid are very small, of the order of 10m (33 ft) per hour. The flow regime through a dual-medium packed bed is thus always laminar making small solid particles desirable from the standpoints of (1) a larger convective heat-transfer coefficient (h) in the bed, because laminar heat transfer depends solely on geometry, and (2) providing a larger A/V (area-to-volume ratio) within the packed bed. The fluid/solid heat-transfer rates, e.g., in MWt (or Btu per hr), during charging or extraction increase with the product (h) (A/V). Figure 4-6 shows how a decrease in effective particle diameter beneficially reduces the fluid inventory.

In view of the laminar flow regime in the bed, the bed pressure loss is small, of the order of 0.7 to 2.1 KPa (0.1 to 0.3 psi), even when the void fraction is in the low 0.2 to 0.3 range, based on estimates from Reference 4-12.

Selection of Dual-Heat-Storage Media

It is desirable for the heat-transfer fluid as well as the particulate bed solids to have a high product of (specific heat x density) to maximize energy stored per unit volume of bed. Also, the relative cost of the dual-medium storage should be minimized in terms of dollars per unit energy stored. Table 4-6 compares various candidate fluid and solid media in terms of both criteria.

With the 316°C (600°F) maximum operational temperature requirement for the fluid, there are three principal candidates (all liquid at room temperature): Therminol 55, Therminol 66, and Caloria HT43. The table shows that Caloria HT43 is the most desirable from both the volumetric storage and cost standpoints. To determine comparative degradation, long-term tests were carried out on these fluids to evaluate thermal stability, compatibility with materials of construction, and fouling of heat exchangers. These tests are reported in Section 6.2. Briefly, Therminol 55 was found to have excessive weight loss due to volatilization and was eliminated from consideration. Caloria HT43 and Therminol 66 were found to have excellent stability and absence of fouling problems.

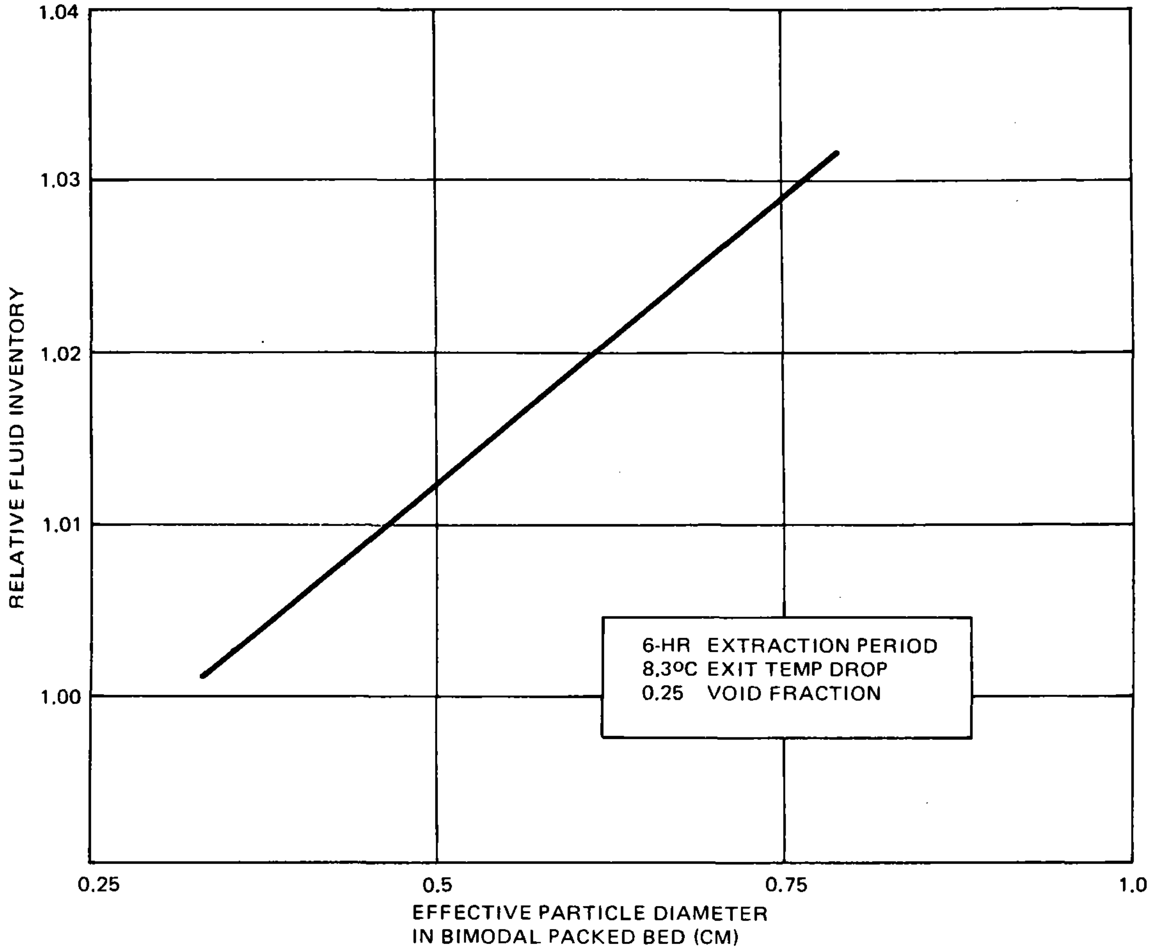


Figure 4-6. Effect of Particle Diameter on Tank Liquid Inventory

Table 4-6

THERMAL STORAGE MEDIA COMPARISON

Medium	Maximum Practical Temperature, °C(°F)	Volumetric Heat Capacity Megajoule/M^3-°C (BTU/Ft^3-°F)	Relative Cost* ($/KWth-hr)
Rock	$1,140^{+}$ ($2,000^{+}$)	2.26 (33.7)	0.31
Iron Ore	$1,140^{+}$ ($2,000^{+}$)	3.84 (57.3)	2.20
Jet Fuel	204 (400)	1.85 (27.6)	2.50
Caloria HT43	316 (600)	2.05 (30.6)	6.80
Therminol 55	316 (600)	2.04 (30.5)	10.00
HITEC	588 (1,000)	2.75 (41.0)	12.00
Therminol 66	343 (650)	1.96 (29.3)	33.00
NaK Eutectic	860^{+} ($1,500^{+}$)	0.66 (9.8)	38.00
Caloria HT43 plus Rock**	316 (600)	2.61 (32.9)	1.90
Therminol 66 plus Rock**	343 (650)	2.18 (32.6)	8.60

* For 83°C (150°F) temperature range

** With 0.25-void fraction.

The laboratory data on Caloria HT43 and Therminol 66 were translated to required fluid replenishment rates for typical diurnal thermal cycles, both for the Pilot Plant and Commercial Plant. For the Pilot Plant cycle (218°-302°C or 425-575°F) the annual replenishment rates, in percent of inventory per year, were 7.0 for Caloria HT43 and 6.1 for Therminol 66; for the Commercial Plant (232-316°C, or 450-600°F cycle) the corresponding percentages were 12.9 for Caloria HT43 and 30.9 for Therminol 66. It should be noted that these replenishment rates are conservative since no attempt was made during laboratory tests to capture and return volatiles to the system as well be done by the ullage maintenance unit during actual subsystem operation.

In view of the roughly 6-to-1 cost advantage of Caloria HT43, it is much less expensive both in initial cost and in makeup fluid cost than Therminol 66. Therefore, Caloria HT43 was selected as the heat-transfer fluid. Its availability in Pilot Plant quantities is assured, and with reasonable lead times Commercial Plant quantities will be obtainable.

In Table 4-6 rock and iron ore are listed as particulate solid materials. Rock (granite) is the cheapest and most cost-effective medium, even though iron ore (magnetite) has a higher volumetric heat capacity and thermal conductivity. A comparison showed a 10% cost advantage for rock over iron ore at the Pilot Plant size and about 35% at the Commercial Plant size. Therefore particulate rock was selected as the solid bed medium. Rock products come in all kinds of particle sizes and shapes, permitting a selection of optimum bed packing.

For rock and Caloria HT43 the volumetric heat capacities are approximately equal (see Table 4-6) at about 2.2×10^6 joules/m^3-°C (33 Btu/ft^3-°F). Thus, the bed will store the same energy independent of void fraction. This means that it is desirable to select solids for the lowest possible void fraction to minimize storage media cost through maximum space use of the low-cost solid portion. Traditionally, low void fractions are produced by using two or more particle size ranges, differing by about a factor of 10; the smaller particles fill the voids between the larger particles. Void fractions as low as 0.21 were obtained in Company-sponsored small-scale tests with just two size ranges, a commercial crushed granite rock and a coarse silica sand. In the SRE large-scale tests a similar bimodal mix was used consisting of 25.4 mm (1 in.) river gravel and No. 6 silica sand (1.5 mm or 1/16 in. typical particle size), and a void fraction of 0.28 was achieved on the first large-scale loading. A void fraction of 0.25 is judged to be attainable in the Pilot and Commercial Plant TSU's.

Dual-Medium Thermal Design

The Schumann-Hausen solution presented earlier in this subsection under "Dual-Medium Heat Storage Analysis" makes idealizing assumptions in order to permit a closed-form solution for solid and liquid temperature in the bed at any point at any time. The Rocketdyne stepwise computer solution relaxes the idealized requirements in that it allows properties (including solid and liquid densities and specific heats and in-bed heat transfer coefficients), to vary with temperature and velocity, and permits arbitrary boundary and initial conditions. However, there are additional nonidealities.

In early thermal design work in 1974 (Reference 4-2) corrections were estimated in the form of increments to be added to tank diameter and bed height to account for these nonidealities. Subsequently, well-instrumented SRE tests were carried out in which actual thermoclines and other parameters were measured during charging, extraction and hold periods, over a range of charge and extraction rates. The present thermal design is based on firm data from the SRE-TSU tests, which serve as the basis for correlations directly applicable to Pilot and Commercial Plant TSU design.

The SRE thermal storage unit is shown in Figure 4-7, with some of the design and performance features listed. Figure 4-8 is a photograph of the unit tested in the SRE program. Figure 4-9 and Figure 4-10, contain typical performance data for this storage unit, operating over the 219°-302°C (425°-575°F) temperature range planned for the Pilot Plant; these results are raw data, plotted on a computer CRT display.

Figure 4-9 shows a series of thermoclines in the TSU, each curve representing the temperature profile at a particular time. (The time key is shown in the table at the top of the figure.) At the start of the extraction test depicted,

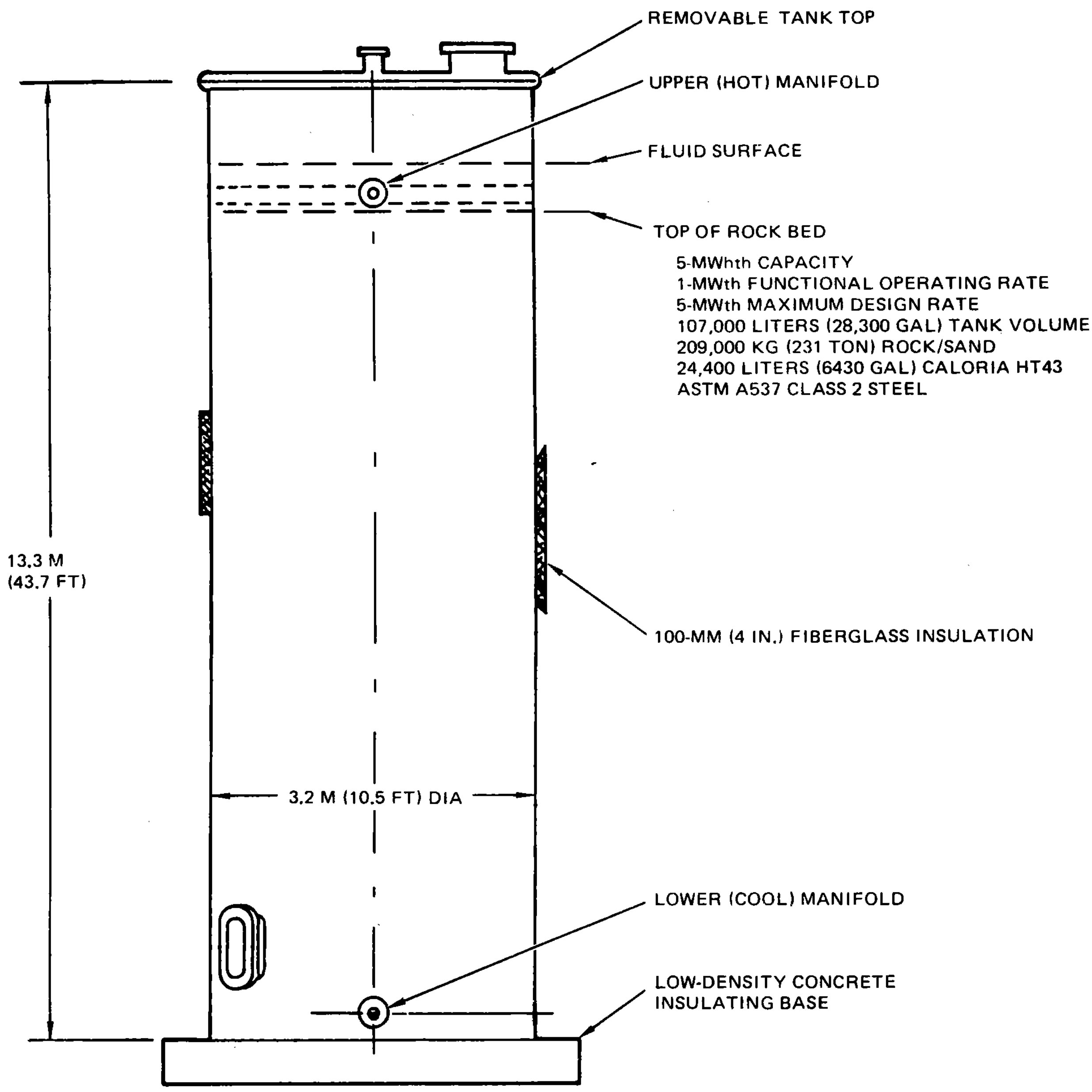


Figure 4-7. Dual-Medium Thermal Storage Unit (TSU) Built and Operated by Rocketdyne as Part of SRE

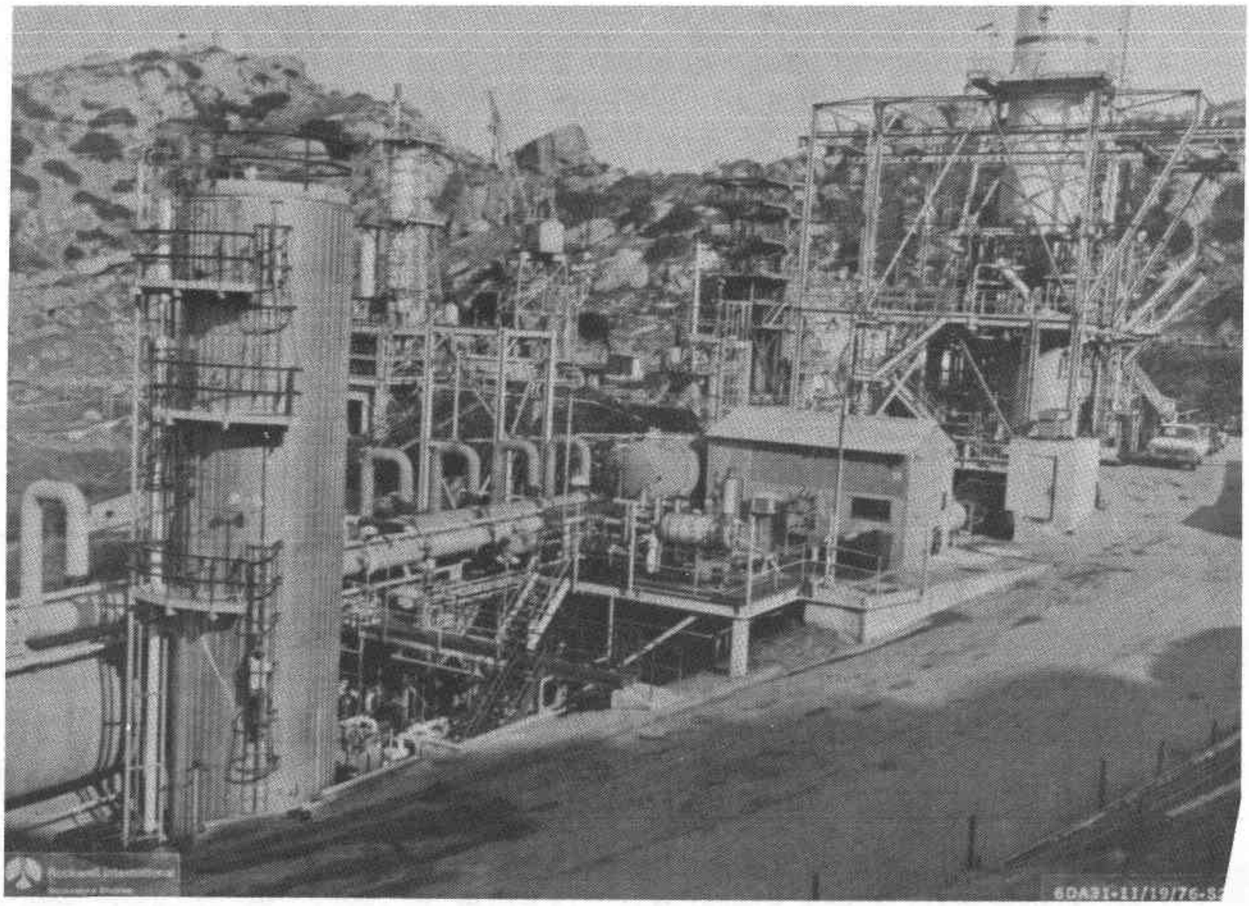


Figure 4-8. Dual-Medium TSU at Rocketdyne Santa Susana Field Laboratory

1 DAY 337 H.M.S 16.54.39	2 DAY 337 H.M.S 17.19.58	3 DAY 337 H.M.S 17.51.36
4 DAY 337 H.M.S 18.25.00	5 DAY 337 H.M.S 18.56.55	6 DAY 337 H.M.S 19.30.00
7 DAY 337 H.M.S 20.00.00	8 DAY 337 H.M.S 20.31.55	9 DAY 337 H.M.S 21.09.42

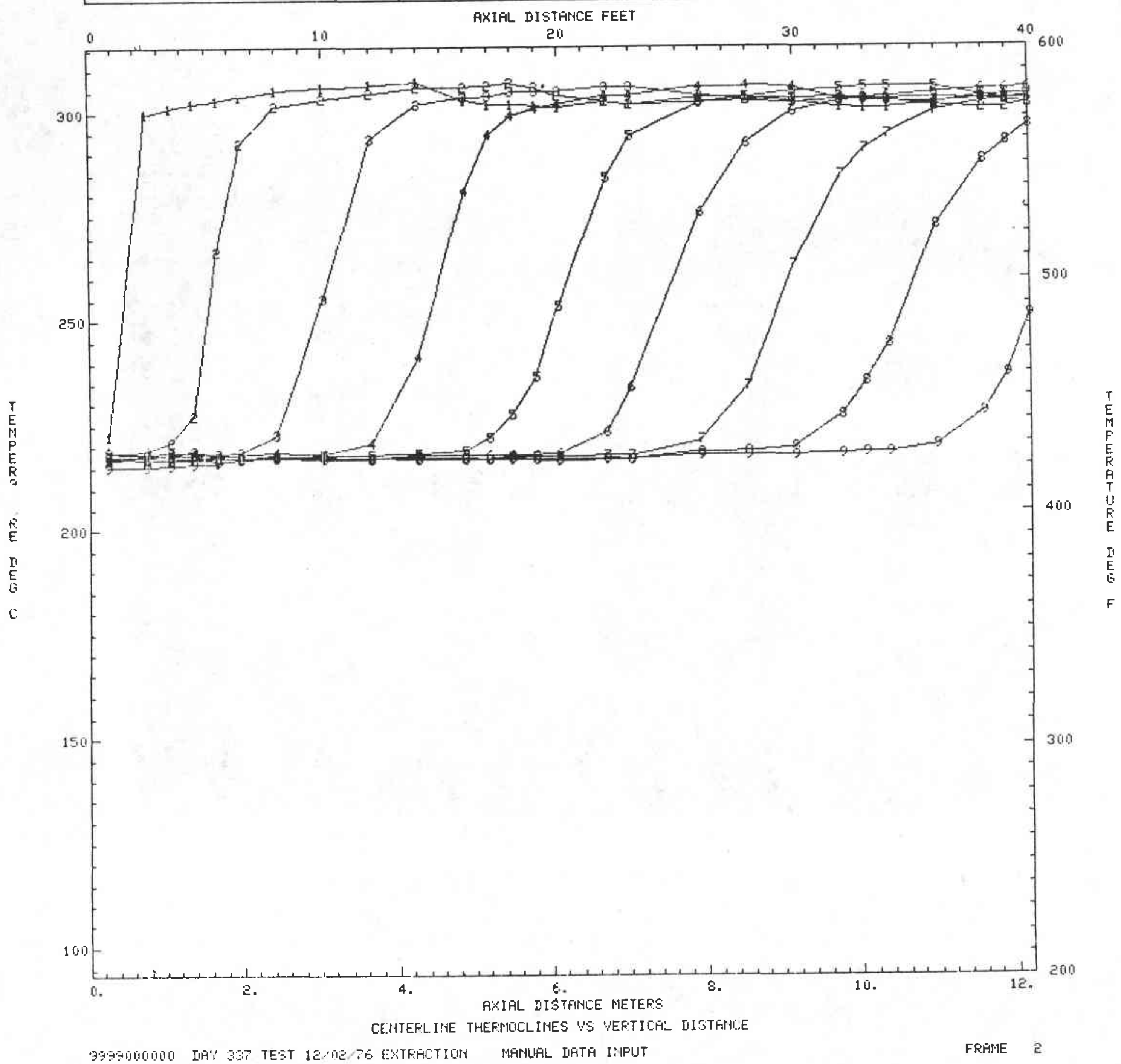


Figure 4-9. Typical Extraction Thermoclines (SRE - Run 32)

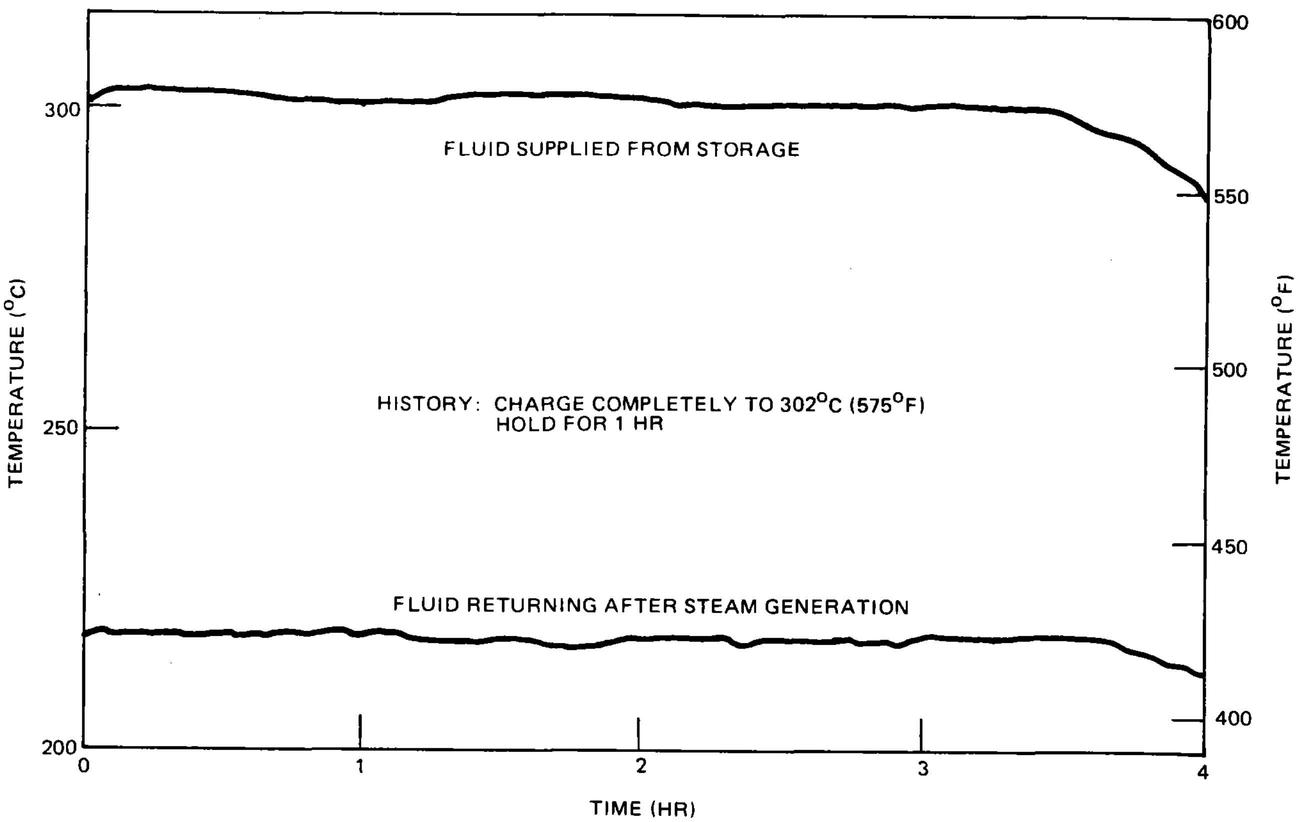


Figure 4-10. Typical Performance Data During Extraction

the entire TSU was charged to 302°C (575°F). Shortly after the start of extraction (curve 1) a small portion of the bottom of the TSU (zero axial distance is the bottom) had dropped to 219°C (425°F), but the remainder of the tank, above a sharp thermocline, remains at the top operating temperature, 302°C (575°F). As energy extraction continues, the thermocline moves upward in the TSU (e.g., curves 2 through 7), with the top of the TSU remaining at the upper operating temperature) until almost all of the stored energy is extracted. Then, the thermocline begins to "break through" the top of the bed and the top temperature begins to fall off (e.g., curve 8).

The temperature of the hot fluid delivered from the top of the TSU is shown in the top half of Figure 4-10. An expanded temperature scale is used to magnify any variations in fluid temperature. It can be seen that the temperature is flat, ±1°C (±2°F), which is well into the "noise level" for temperature measurements on such a process throughout the extraction until very near the end, when the temperature begins to tail off, as predicted by Rocketdyne's computer model of the dual-medium thermal storage system. A temperature of 294°C (560°F) has been selected as a practical cutoff point to terminate extraction and begin recharging this test; 5.1 MWHt of energy was delivered with fluid temperatures above 294°C (560°F), corresponding to a volumetric efficiency of 87% (i.e., 5.1 MWHt is 87% of the total stored energy in a fully charged tank between the temperature limits of 219°C (425°F) and 302°C (575°F). This performance is even better than estimated during the design for a unit of this size, and should increase as the TSU size and capacity increase, due to reduced heat loss because of more favorable tank surface area to volume ratios.

There are two types of nonideal characteristics of the thermal storage unit which require it to be larger than just enough to store the amount of energy to be extracted: (1) heat losses to the environment, and (2) unavailable energy. The first type represents all heat losses from the TSU, and expresses the amount of charged energy which is lost for extraction. The second type represents a portion of the stored energy which cannot be extracted in a given cycle, but which does not have to be recharged either, i.e., there is no net loss of energy from the subsystem. Upon the initial charging of the TSU, the unavailable energy fraction (typically of the order of 10 to 15 percent) of the total stored energy must be charged and stored, but, thereafter, it merely remains in the TSU, analogous to gasoline below the bottom of the fuel line in an automobile tank. These two types of nonideal characteristics are illustrated in Figure 4-11. It can be seen from the figure that both nonideal characteristics decrease in size, relative to the total stored energy, as the stored energy increases.

Two efficiencies are defined. The most important efficiency is the energy recovery efficiency (Q_E/Q_C in Figure 4-11), which expresses the fraction of charged energy that can be extracted on a continuous basis (typically 96 to 98% for Pilot and larger subsystems). This efficiency is the measure of performance that should be used when characterizing the thermal storage subsystem. Another, secondary, efficiency is the volumetric efficiency (Q_E/Q_T) in Figure 4-11), which is a measure of the size of TSU required to deliver a given amount of extractable energy. This efficiency (typically 85 to 90%) affects only TSU sizing and cost within the TSS, and has absolutely no effect on overall plant performance.

Dual-medium TSU performance is primarily a function of thermocline velocity and thermocline slope, as evidenced by Figure 4-9. The design method used matches the observed thermoclines which are representative of the nonidealities as to (1) velocity, which is mainly influenced by heat loss, by selection of a suitable "effective" solid specific heat C_s^* and (2) slope, which is mainly influenced by fluid-solid heat transfer, by selection of a suitable A/V. The Rocketdyne computer model was then used with these values of A/V and C_s^* to predict the Pilot Plant and Commercial Plant TSU thermoclines and thermal performance.

Figure 4-12 is a replot of Figure 4-9 (SRE extraction Run 32) with calculated curves for A/V of 24 and 120 ft^{-1} (79 to 394 m^{-1}), which bracket the slope of the observed curves. A value of A/V = 40 ft^{-1} (131 m^{-1}) is a close match. The calculated curves were based on a velocity-matching C_s^* less than the actual solid specific heat, as would be expected with heat loss. For a charge as expected with heat loss, C_s^* was found to be larger than the solid specific heat. Figure 4-13, an SRE charge (Run 35), is shown with calculated thermoclines for A/V = 24 and 120 ft^{-1} (79 and 394 m^{-1}) with A/V = 40 ft^{-1} (131 m^{-1}), again a good match. Lastly, Figure 4-14 shows an SRE run

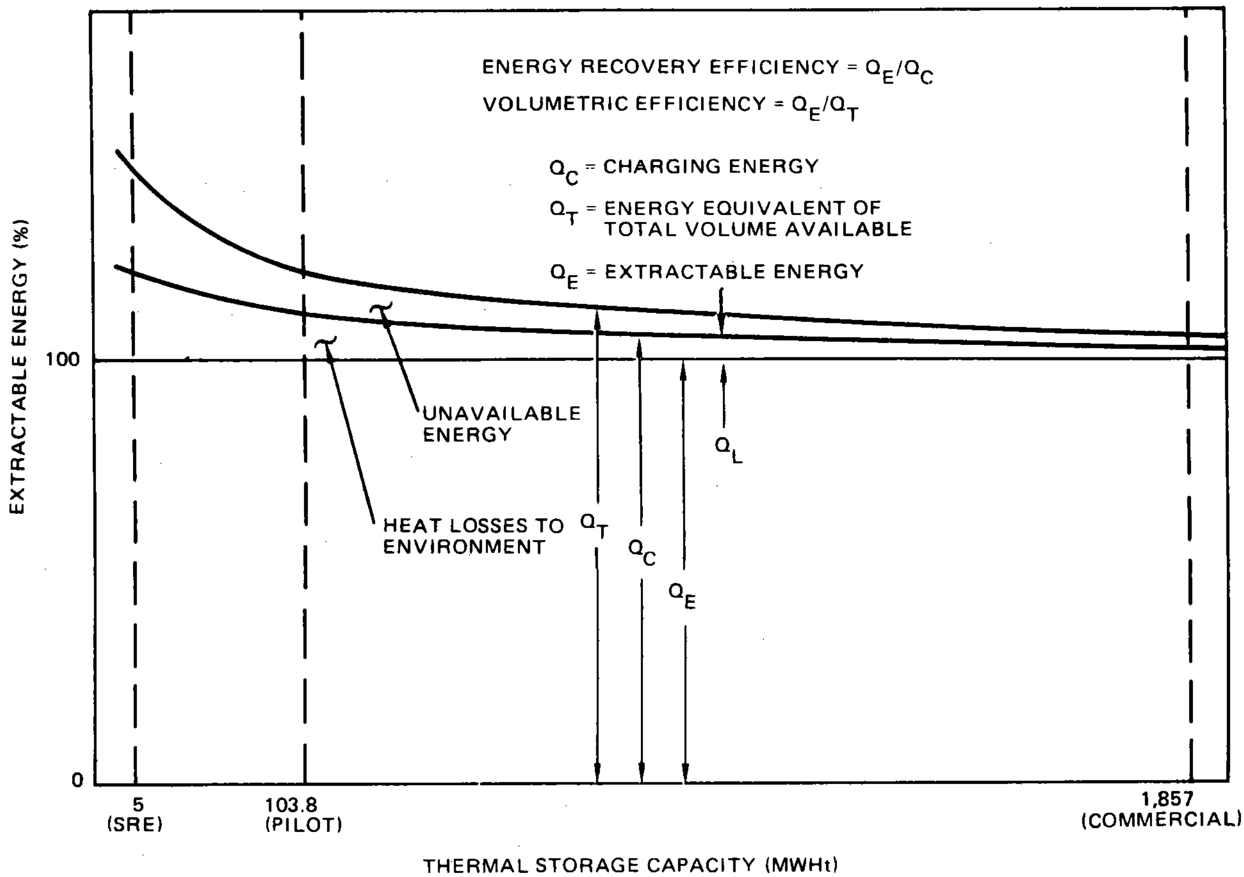


Figure 4-11. Thermal Storage Unit Non-Ideal Performance Characteristics

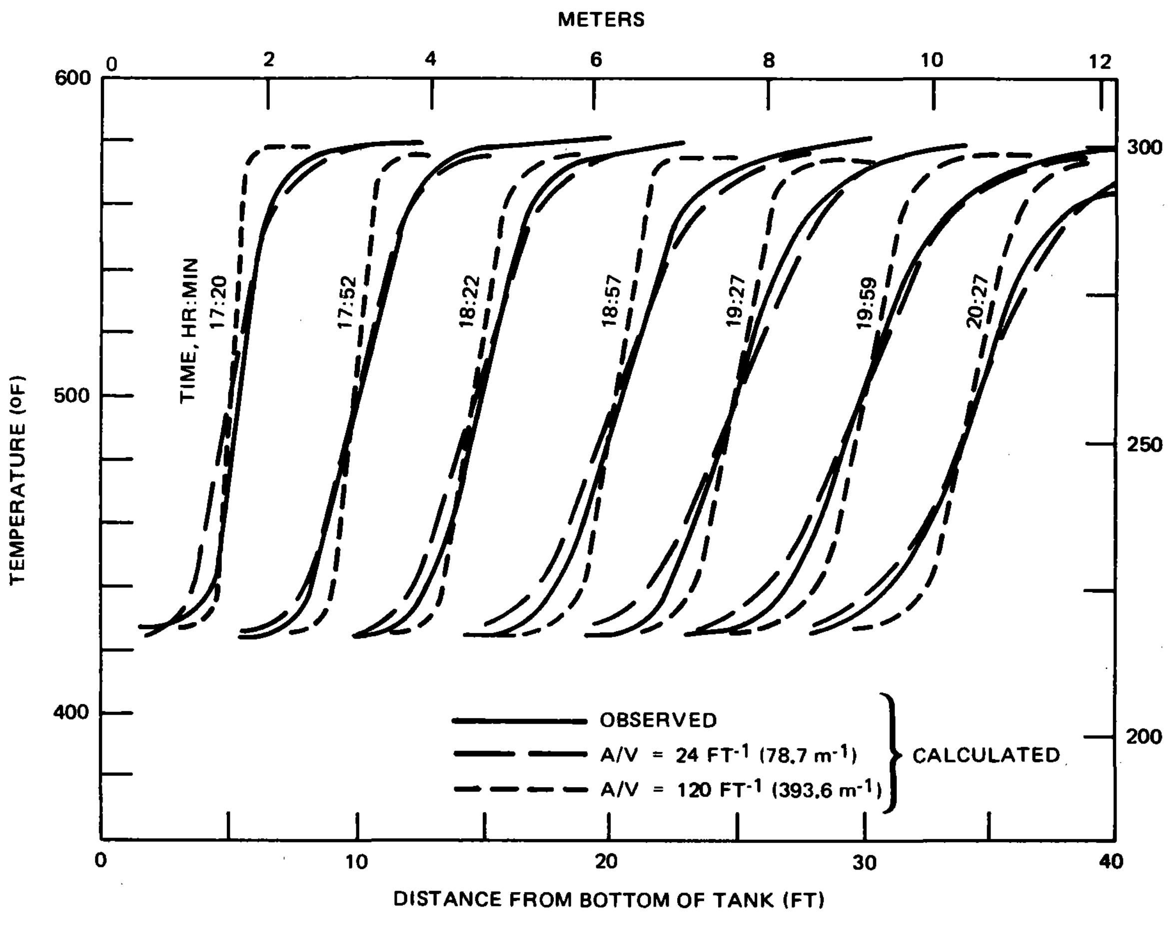


Figure 4-12. Observed vs Calculated Thermoclines for Extraction (SRE Run 32)

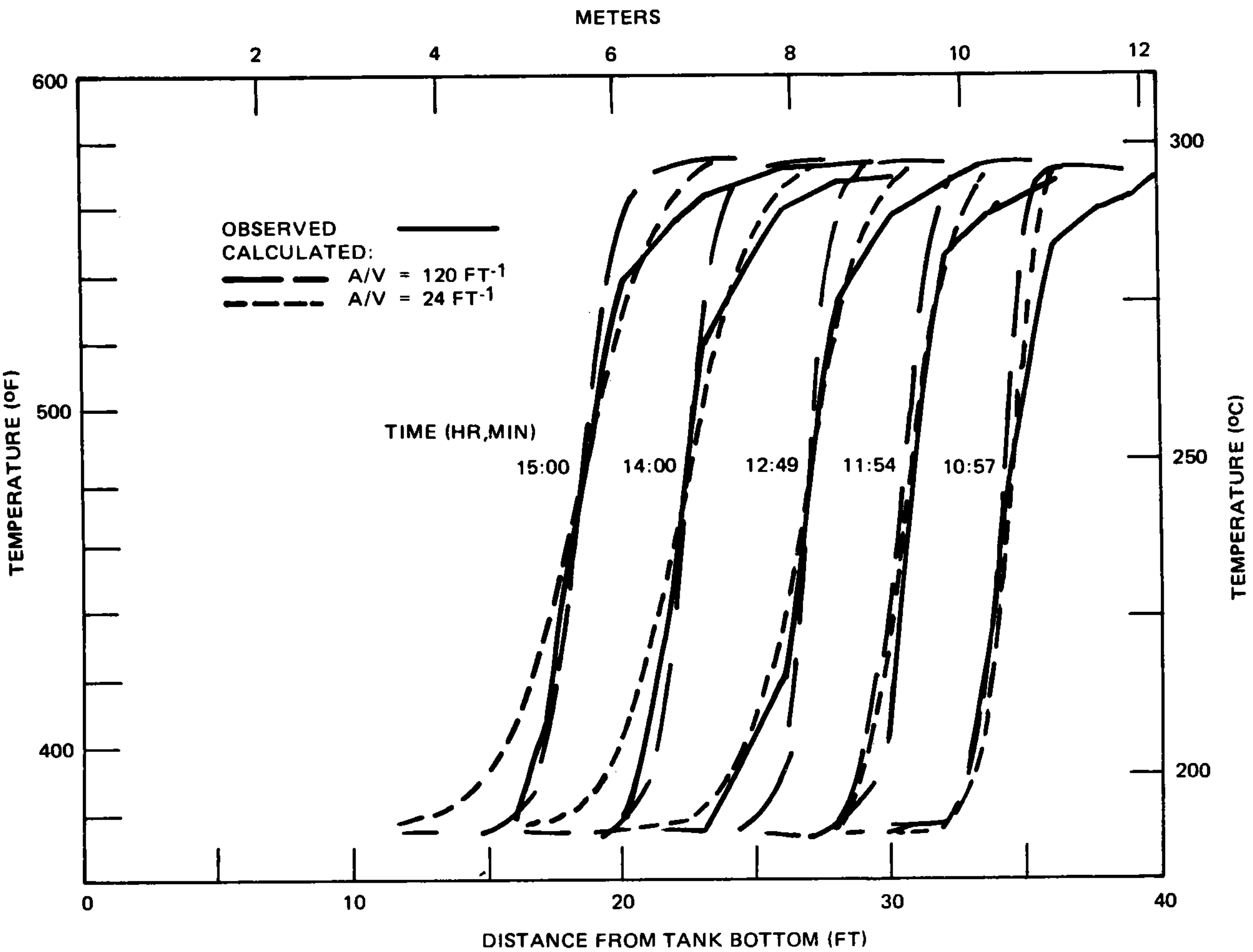


Figure 4-13. Observed vs Calculated Thermoclines for Charging (SRE Run 35)

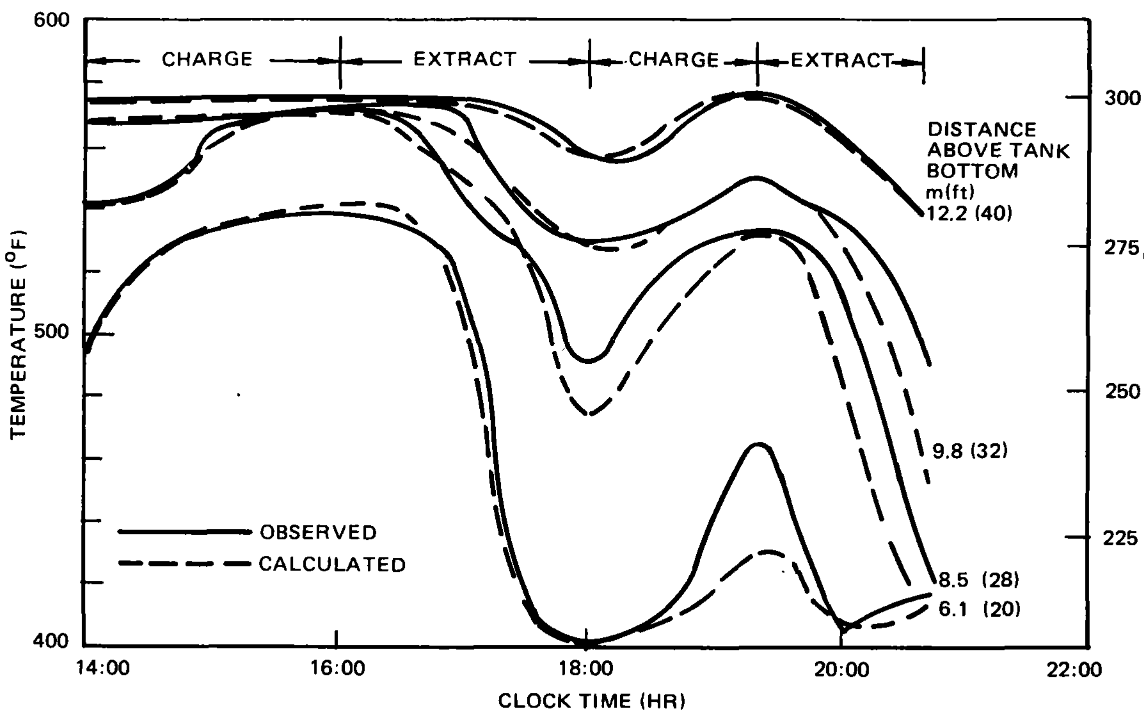


Figure 4-14. Observed and Calculated Bed Temperature Profiles During Double Charge/Extract Cycle (SRE Run 47)

devoted to a charge-extract-charge-extract double cycle (Run 47). The dashed curves were calculated by the Rocketdyne computer model with the values C_s^* and A/V found for charge and extraction, respectively, in the two previous figures. It is seen that the predicted curves lie very close to the observed ones, giving confidence to these design methods for realistic duty cycles.

The design methods described, which were based on actual measured SRE performance, were used to define the Pilot Plant TSU bed dimensions. An important requirement is the ability of the TSU to deliver the specified extractable energy over a range of extraction rates after a 20-hr hold at full charge. This requires a knowledge of the heat losses from the tank, not only during cyclic operation, but also when fully charged. The optimum height/diameter ratio for minimum heat loss from all tank surfaces (including the bottom) is unity, at which point the ratio of tank heat loss area to tank volume is a minimum. Such a "square" and well-insulated tank therefore represents an optimum shape. The design calculations indicated that both the height and diameter should be in the 12.2 to 15.2 m (40 to 50 ft) range for a single-tank Pilot Plant design.

To take maximum advantage of the SRE results, obtained with a bed height of about 12.5 m (41 ft), it was decided to make the Pilot Plant TSU bed height identical to that for the SRE. The question of the tank diameter remained to be answered after an evaluation of the SRE heat loss data and their application to the Pilot Plant design, as discussed in the next subsection.

Heat Losses and Insulation

A cylindrical TSU tank will exchange heat with the environment across all exterior surfaces: top, bottom, and cylindrical sidewall. The temperatures inside the tank will vary over the operating range 219°-302°C (425°-575°F), with different bed surfaces at different temperatures throughout a diurnal cycle. In particular, the top is always exposed to the maximum temperature 302°C (575°F) except when the full charge has been extracted from the bed. Similarly, the bottom is at the minimum temperature, 219°C (425°F) at all times except when the tank carries a full thermal charge. The upper portion of the sidewall will be close to the maximum temperature, and the lower portion close to the minimum, the dividing line being at the thermocline elevation, which varies during the day.

Heat loss through the top will be increased somewhat by any circulation in the tank ullage; however, the packed bed will eliminate any liquid circulation along the sidewalls so that the only internal heat transfer mechanism there is conduction. This is also the case for the bottom. The tank sides and top are exposed to the atmospheric conditions set forth in the requirements: 28°C (82.6°F) air temperature, with a wind velocity varying from zero at the ground to 3.5m/(8 mph) at a height of 10m(32.8 ft) and increasing with height. The resulting combined radiative and convective atmospheric heat transfer coefficients (varying with height) are then in the range from 5.7 - 22.7 $W/m^2 °C$ (1 - 4 Btu/hr $ft^2 °F$), using well-known forced convection correlations for airflow over a flat plate (for tank top) or across a cylinder (for sidewalls).

In view of the requirement for a 20-hour hold at full charge, adequate insulation of the top and sides of the tank is mandatory. In the SRE TSU a 10 mm (4 in.) thickness of fiberglass insulation (Owens-Corning Intermediate Service Board, or ISB for service up to 454°C (850°F) of thermal conductivity k_i = 0.05 Btu/hr ft °F(0.08 W/m°C) was used, with an outside corrugated aluminum weather cover, as may be seen in the photograph of Figure 4-9. During the SRE testing heat flux meter readings were taken at different operating conditions, including holds, at various points on the exterior of the TSU. Figure 4-15 shows measured values (shaded bands) for top and sidewalls with the 10 cm (4 in.) insulation thickness. The atmospheric conditions for the SRE, monitored at 3-hr intervals, were more severe for heat losses than those in the requirements, due to lower air temperatures.

Because of the heavy bolted flange construction of the top cover, which acted as a fin and short circuited the top insulation, there was excessive heat loss through the top in the SRE. It was decided to use 8-in. (20 cm) fiberglass insulation of the ISB or equivalent type for the top and sides of the Pilot Plant and Commercial Plant TSU's, and to specify a roof construction embodying an insulated truss structure. The resulting predicted heat flows are shown by the triangles for top and sidewall on Figure 4-15.

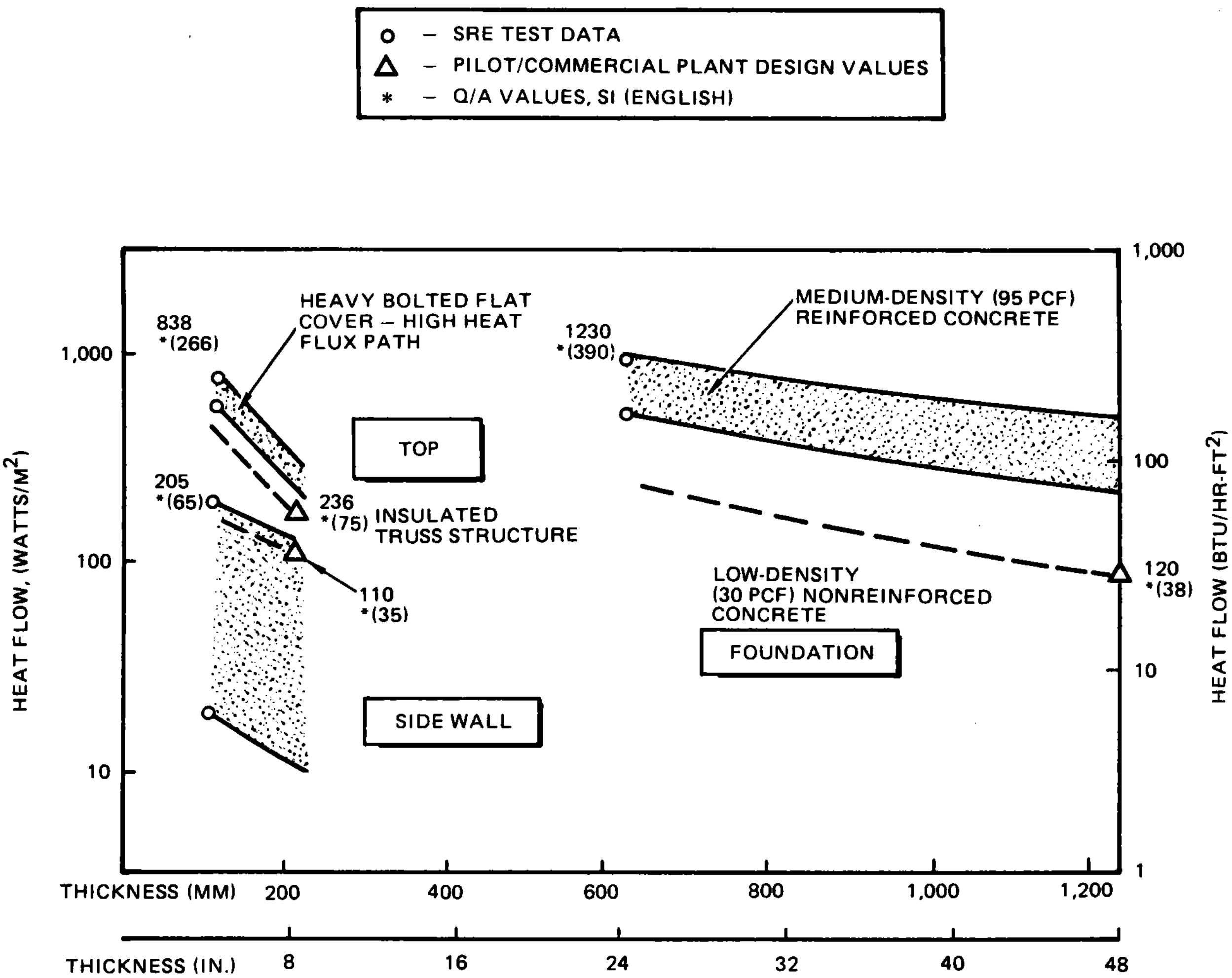


Figure 4-15. TSU Heat Loss Paths

In the SRE, a 61 cm (2 ft) thick medium density 1.52 g/cm^3 (95 lb/cu ft) reinforced-concrete foundation gave rise to large heat losses through the bottom of the tank (see shaded band on Figure 4-15); because the tank was bolted to the foundation, steel reinforcement provided high flux paths and the "deck" extending all around the tank acted as a cooling fin. In the Pilot Plant and Commercial Plant the foundation will be either low-density (0.48 g/cm^3) nonreinforced concrete, or dry soil of adequate bearing strength. The design heat loss value for the low-density concrete foundation is shown as a triangle on the right side of Figure 4-15. The corresponding number for dry soil should be close to that for the low-density concrete; the slightly higher thermal conductivity of the soil is offset by the absence of hold-down bolts which would be specified with the concrete foundation.

The combination of the component heat losses in the proper proportions, considering the area/volume ratio of the tanks in the SRE, Pilot Plant, and Commercial Plant designs, led to values for total heat loss plotted in Figure 4-16. The 24-hr heat loss for the Pilot Plant size is seen to be 2.6% of the energy extractable from the TSU. This may be taken as the 20-hr hold loss and must be allowed for in the TSU design. With the previously designed bed height of 12.5 m (41 ft), an inside tank diameter 15.2 m (50 ft) provided sufficient extractable energy after a 20-hr hold at full charge. A side effect of a 20-hr hold is a decrease of 1.7°C (3°F) in the average temperature of the bed contents. This means that in actual operation the bed must be fully charged at a temperature which exceeds the extraction fluid temperature, nominally 302°C (575°F). The selected 304°C (580°F) temperature of charging fluid will be ample to account for this 1.7°C decrease plus any other temperature losses in lines. For the Commercial Plant the percent heat loss is smaller, 1.8%, and the temperature decrease is below 1°C (1.8°F). The temperature of the outside aluminum weather cover will be in the neighborhood of 49°C (120°F), satisfying OSHA regulations. As the atmospheric conditions change, this temperature will change by a few degrees at most, but the heat loss rate will be virtually unaffected. This is a consequence of the extremely high resistance to heat flow in the fiberglass insulation, two orders of magnitude higher than that inherent in atmospheric radiation and convection.

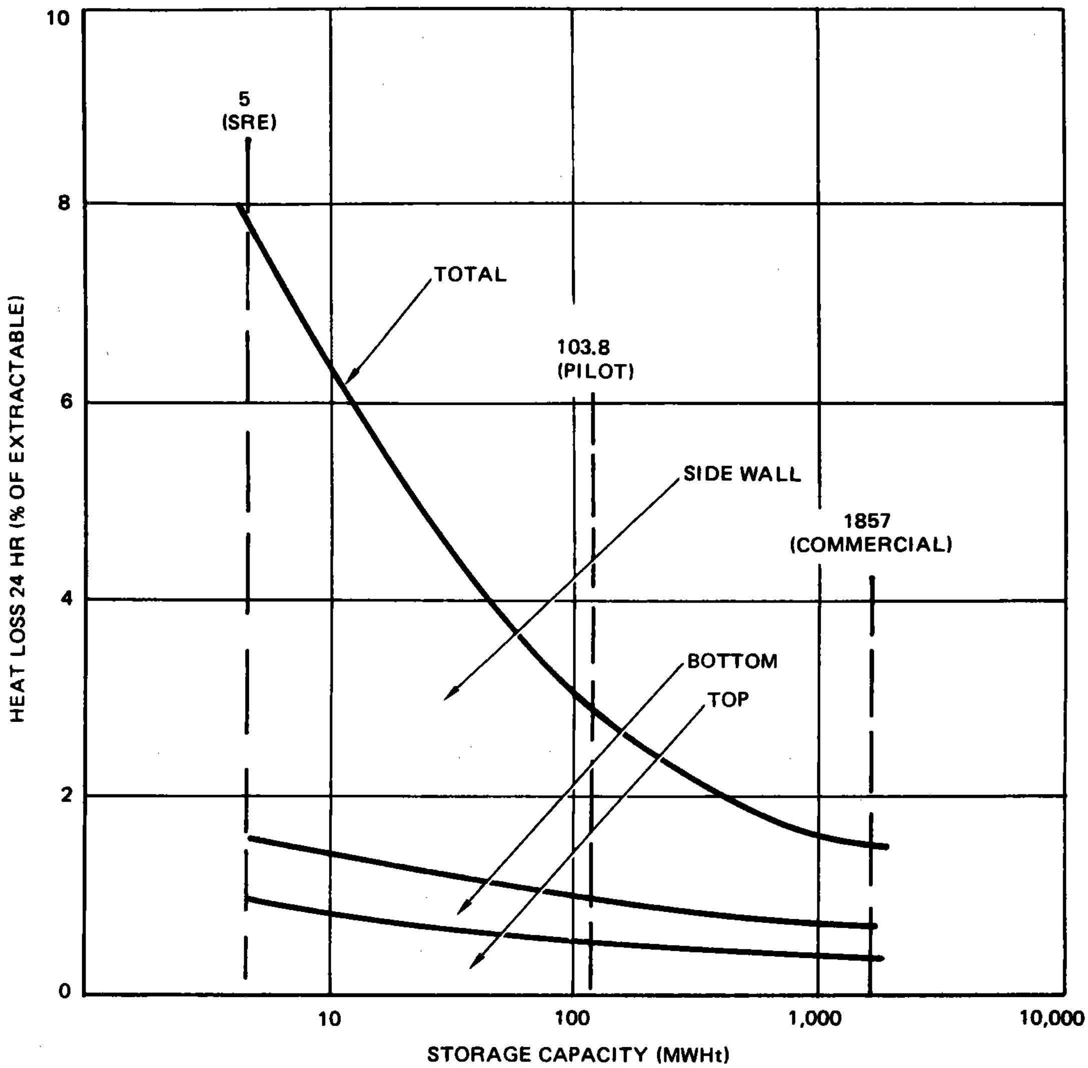


Figure 4-16. TSU Heat Loss Scaling

The insulation design presented above is for a single tank. For the Commercial Plant case four tanks in parallel will be used. A reduction in sidewall heat loss could result from clustering the tanks close to each other, so that the effective atmospheric conditions will be thermally less severe.

Fluid Distribution and Bed Packing

To realize the optimal utilization of the packed bed for heat storage, it is desirable for the thermocline separating the high- and low-temperature liquid to be as sharply defined as possible at all times during its travel up and down the tank. Further, flat radial velocity and temperature profiles across the tank are desirable to produce uniform liquid-solid heat transfer at all points in the bed. Most importantly, radial nonuniformities lead to incomplete heat extraction and degraded temperatures in the extracted liquid stream.

The velocity profile is a function of (1) the distribution of the liquid as it enters the bed through the appropriate manifold, (2) the flow paths through the bed which depend on the distribution of voids in the bed, and (3) the distribution of the liquid as it leaves through the other manifold.

Packed beds have been in use in the chemical process industry (columns for distillation, extraction, etc.) in water treatment (sand and multimedia filters), in heat treatment (pebble bed heaters), and other applications. Most of the named applications use only a narrow size range of particulate packing (unimodal packing). For unimodel packing it has been observed (Ref. 4-13), especially when the tank diameter/particle diameter ratio (D/d) is not very large, that the flow velocity is a minimum in the center and a maximum at about one particle diameter from the wall. At D/d = 30 the velocity nonuniformity is of the order of 20%, but at high D/d values (e.g., 100 or more) the velocity profile should be essentially uniform.

For the TSU, there is a bimodal distribution (25-mm gravel and 1.5-mm sand particles), a D/d of the order of 1,000, and a small void fraction (0.25, compared to 0.35-0.65 in standard commercial applications). The "wall" effect, which produces larger voids at plane boundaries where no interlocking with other particles is possible, should be small. However, this presupposes uniform density of the packed bed.

Where variations in packing density exist, the fluid may "channel," taking the paths of least resistance, usually on a local scale. With bimodal packing and low void fraction, the expectation is that channeling will be eliminated effectively if the initial bed installation is performed carefully. Tests with Rocketdyne's laboratory flow system and with the SRE subsystem have shown no significant channeling or problems with bed packing.

The liquid distribution manifolds must be designed to produce as close to flat velocity and temperature profiles as possible. Microscopically speaking, for heat storage the interstitial liquid flow is laminar, with Reynolds numbers (at 260°C or 500°F) in the 1 to 10 range, which is the "creeping" flow regime. This is predicated on a uniform distribution of liquid which flows through the small interstices between sand particles and through the larger ones between sand and gravel, and between gravel and gravel particles.

Manifolds have a finite number of liquid flow holes or ports and this number must be high enough so that at a short axial distance from the manifold there is a flow in most of the interstices of the bed. This is of special importance for an inlet manifold, and for the upward flow direction during heat extraction. The amount of radial spreading of liquid issuing from a hole in a manifold will vary with its orientation (cocurrent, crosscurrent, countercurrent) and with the liquid velocity through it. The latter increases with the pressure drop across the hole. In a discussion of fluidized beds, it is stated that to prevent channeling the pressure drop through the manifold should be at least 40% of that of the bed itself, and the holes should be small (Ref 4-12). Although the TSU is different (with a fixed bed and liquid instead of gas), similar considerations should hold. The extraction manifold design is less critical.

Holes for the SRE manifolds were disposed in pairs at the bottom of a network of pipes making up each manifold, and were drilled directly into the pipe walls at points 60 deg from the bottom of the pipes. Both the top and bottom manifolds were identical, with holes in the bottom only. The holes were sized for a pressure drop at typical flowrates of 50% of the pressure drop expected through the bed, in accordance with the recommendation in the last paragraph. Also, the hole size was such that only 1-3 percent of the sand could possibly fit through the holes. To further guard against sand ingestion, the bed in the close vicinity of the manifolds was laid with gravel only, to act as a barrier against infiltration. Further away, the regular gravel-plus-sand bed was resumed.

The SRE TSU was equipped with thermocouple instrumentation to determine radial and circumferential nonuniformities in the fluid flow and heat transfer within the bed. The observations were positive. First, evidence of silt movement into the manifolds was minor and the pressure drops in bed and manifold were low, of the order of 7 KPa (1 psi), as predicted. Some clean-out of dust from the rock bed was observed in SRE tests, but this appeared to be an initial bed-conditioning phenomenon, as discussed in Section 6.3.

Figures 4-17 and 4-18 show the flatness and uniformity of the thermoclines across cross sections of the TSU. Figure 4-17 shows the thermocline shape across the tank at 6.1m (20 ft) elevation, as measured by a series of 12 thermocouples spaced from one wall to the opposite wall along a diameter. Zero on the radial distance (horizontal) scale represents the tank centerline, and the two extreme points are at the two walls of the tank. Each curve represents the temperature profile at a particular time, as keyed in the table at the top of the figure. At the first four time slices, the temperature is at the top operating temperature 302°C (575°F). The profile is flat across the tank diameter, with slight curvature near the two walls, corresponding to heat losses through the walls (a phenomenon which has progressively less impact as the tank diameter increases for larger systems).

Curve 5 shows the temperature profile as the thermocline is passing through the 6.1m (20 ft) elevation. The small upward curvature near the walls corresponds to heat being transferred from the temporarily hotter walls to the bed. The last four time slices (curves 6 through 9) show the temperature at the lower operating temperature 218°C (425°F) after passage

1 DAY 337 H.M.S 16.54.39	2 DAY 337 H.M.S 17.19.58	3 DAY 337 H.M.S 17.51.36
4 DAY 337 H.M.S 18.26.55	5 DAY 337 H.M.S 18.56.55	6 DAY 337 H.M.S 19.31.55
7 DAY 337 H.M.S 20.03.55	8 DAY 337 H.M.S 20.31.55	9 DAY 337 H.M.S 21.09.42

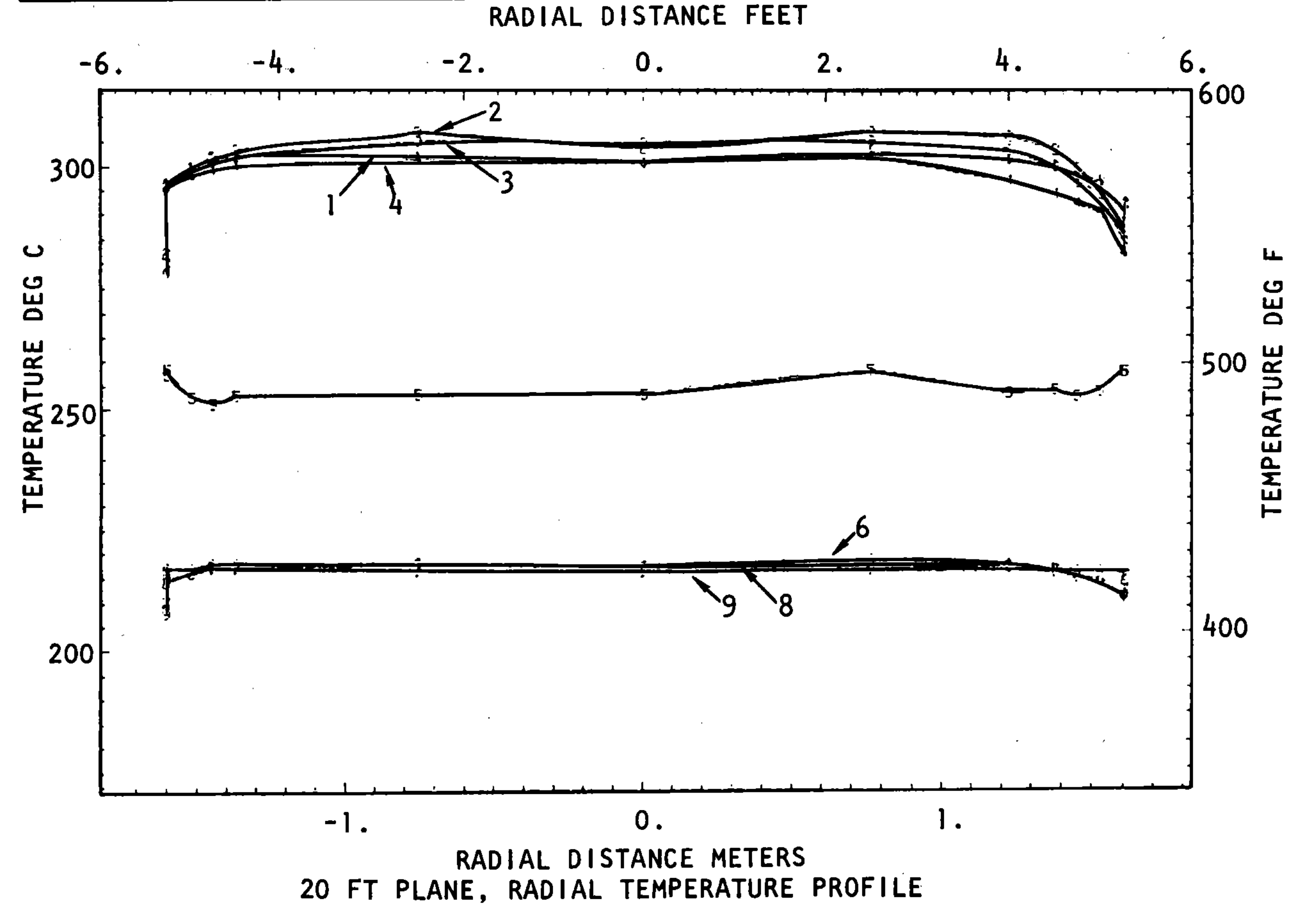


Figure 4-17. Thermocline Shape Across Tank Radius (SRE Run 32)

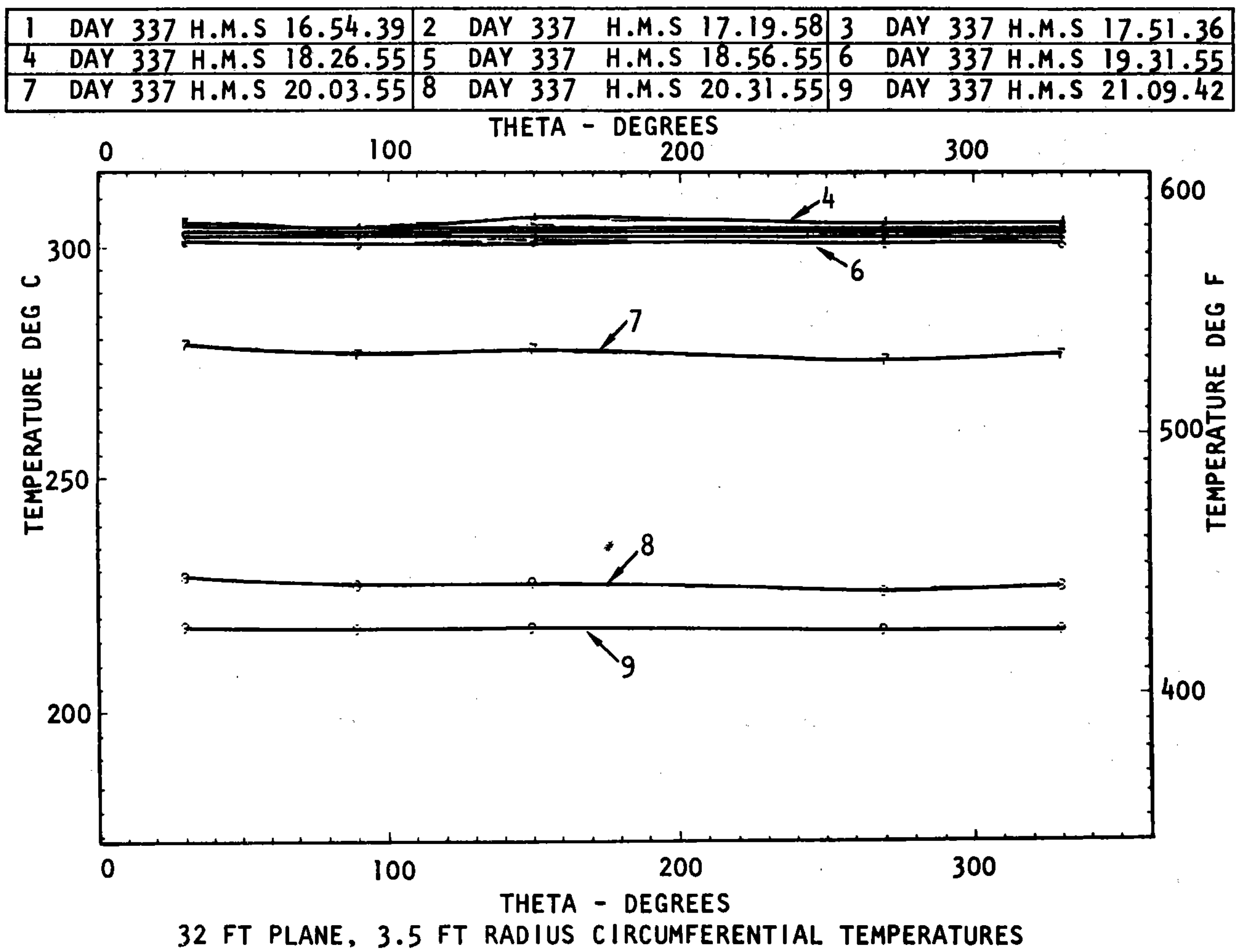


Figure 4-18. Thermocline Shape Over Tank Cross-Section (SRE Run 32)

of the thermocline. The temperature profiles can be seen to be very uniform across the tank, showing absence of channelling, "rat-holing," or other non-uniformities in fluid flow and heat transfer.

Figure 4-18 shows similar results for the flatness and uniformity of the temperature profiles across the bed cross section, in this case at 9.8m (32 ft) elevation. At this elevation there is a circular rake of thermocouples spaced over a full 360 deg at 1.1m (3.5 ft) radius. As with Figure 4-17, each curve represents a time slice. It can be seen that the temperature profiles are extremely uniform, even when the thermocline moves past this elevation.

Demonstration of such uniform thermoclines in a large-scale dual-medium system is especially positive evidence of the practicality of the concept and the simple bed-loading techniques used to install the rock bed in the TSU. There is almost no influence of the TSU walls beyond a short distance; therefore, the performance of the TSU should scale rather directly to larger units (which have even less influence on the walls). This result greatly increases the confidence in scaling up the results to Pilot Plant and subsequent Commercial Plant thermal storage subsystems.

In the light of the excellent SRE experience detailed above, the Pilot Plant and Commercial Plant manifold and bed design will involve some cost savings. The total fluid flow area through the holes per unit tank cross-sectional area will be maintained at the same value as in the SRE TSU. However, there will be a cost saving because the manifold piping will be approximately halved, and the number of holes to be drilled will be reduced by a factor of four. At the same time it is expected that the thermocline uniformity and flatness will remain unimpaired and excellent.

Structural Analysis

Detailed structural analysis of the TSU tank is necessary to prove the 30-yr life capability with combined load of rock and fluid when operating over the required daily cycling temperature variation, 218° to 302°C (425° to 575°F) for Pilot Plant and 232° to 316°C (450° to 600°F) for Commercial Plant.

The TSU structural analysis presented herein has been derived from considerations of soil mechanics, applications of the ASME unfired pressure vessel code, the American Petroleum Institute welded oil storage tank code, and the SRE test data.

The initial structural model used the traditional concepts derived from soil mechanics which resulted in an "active load" and a "passive load". With the assumption of a 30-deg angle of internal friction for the rock fluid mixture, the most severe condition was evaluated to be the passive pressure of the rock pushing on a tank wall with a hydrostatic pressure equivalent to a fluid with a specific gravity of 5.02. Application of this principle resulted in conservative wall thicknesses and severe constraints of tank height versus diameter relationships. It was uncertain whether the passive load would ever develop and how to model properly the TSU tank with thermal cycling charge/extraction stress variations.

Further analysis has revealed an approach to determination of stress values which are still conservative, but at the same time result in thinner wall sections and provide a more precise definition of the stresses resulting from the rock in combination with the thermal cycling. This model has evolved from analysis as applied to large grain elevators and similar structures. It is a well-established procedure that provides analysis for the rock load on the wall and is presented in the structural design handbook by Ketchum, Ref. 4-14.

Wall Stress Mechanism – The new structural approach postulates the following series of events related to the cycling heating and cooling of the tank walls. Initially, the TSU tank is filled with rock and sand and the resulting force on the wall is the active load. The active load results from the normal gravitational pull and spreading of the rock downward, and its value depends upon the angle of repose of the solids mixture. As the TSU is heated, the rock and tank walls expand. Because the coefficient of expansion of steel is higher than the rock-sand mixture, a gap will occur between the bed and the tank wall. During the first heating the tank will expand more than the rock bed and the tank wall load may actually decrease. If it is assumed that the gap is filled eventually by rock and sand, then (by soil mechanics definitions) the force on a wall at constant temperature is again the active load. In order for this to occur, the rock and sand must rearrange itself in the bed to fill the gap and the net result will be a settling of the bed to a new position. This settling may occur very little or it may occur up to the maximum extent possible which, by definition, results in application of the "active load."

In the case of the SRE tank, measurements of the bed height indicate that little or no settling took place after 3 mo of continuous operation in which approximately 25 heating and cooling cycles were imposed. However, if it is assumed that the bed settles the maximum amount, imposing the active load on the wall at the maximum temperature, then, as the system is cooled, an additional stress will appear because the tank wall contracts more than the rock bed. If it is assumed that the rock bed yields only by shearing upward to the top of the bed, then a stress in the wall can be computed, which represents the most severe condition that can posibly exist. The stress on the wall increases as the temperature decreases and this can be calculated accurately based on the temperature coefficient of the expansion of the bed and wall. During reheating of the TSU (charging) the bed and wall will expand back to the original condition of active stress that occurred at the maximum temperature, and complete relaxation of the temperature imposed stress will occur. During subsequent charging and extraction cycles the stress on the wall will vary but it can be no higher than that postulated since it is controlled by the basic material properties.

The stress model is similar to the problem of an elastic circular thin ring around an inelastic solid cylinder combined with a tensile stress preload. If the inner inelastic cylinder has a coefficient of expansion smaller than the outer elastic cylinder, the tensile stress in the elastic ring will increase when the temperature is decreased.

The model thus predicts thermal stresses definitely on the conservative side and the results are applicable to the full 30-yr life of the TSU. It is evident from this approach that the so called "ratcheting" cannot occur with a continuous and unending increase in stress with time. Ratcheting may be observed, in fact, but at most to build up the stresses to the limiting design values that will result from this analysis.

The following section describes the mechanism in more detail.

Initial Loading – After the initial loading of the tank with the rock and heat transfer fluid the pressure at the wall is defined as the active pressure (P_a) and is given by the following relationship:

$$P_a = K_a W_B H$$

$$K_a = \tan^2\left(45° - \frac{B}{2}\right)$$

This expression provides the combined active and fluid load on the wall where K_a is a coefficient of active pressure, W_B is the buoyant density in the fluid, H is the bed height of the tank wall section of interest, and the angle B is the characteristic angle of internal friction of the solids. The total pressure on the tank wall is then the sum of P_a and the normal hydrostatic pressure exerted by the fluid, $W_B H$. The angle B represents the friction angle or angle of repose and increases with increasing relative density of solids as well as increasing with the sharpness or angularity of the grains. The value of B can be determined experimentally by performing a shear test; representative values of B for sand (from Reference 4-14) are as follows:

Condition	Angle of Internal Friction for Sand (Deg): Round Grains, Uniform	Angle of Internal Friction for Sand (Deg): Angular Grains Well Graded
Loose	28.5	34
Dense	35	46

The angle of internal friction was assumed to be 30 deg for the fluid plus rock mixture. Figure 4-19 (Sheet 1) indicates the various stages and conditions in the thermal cycle. Diagram A depicts the load on the wall of the unit resulting from the active and hydraulic components at room temperature.

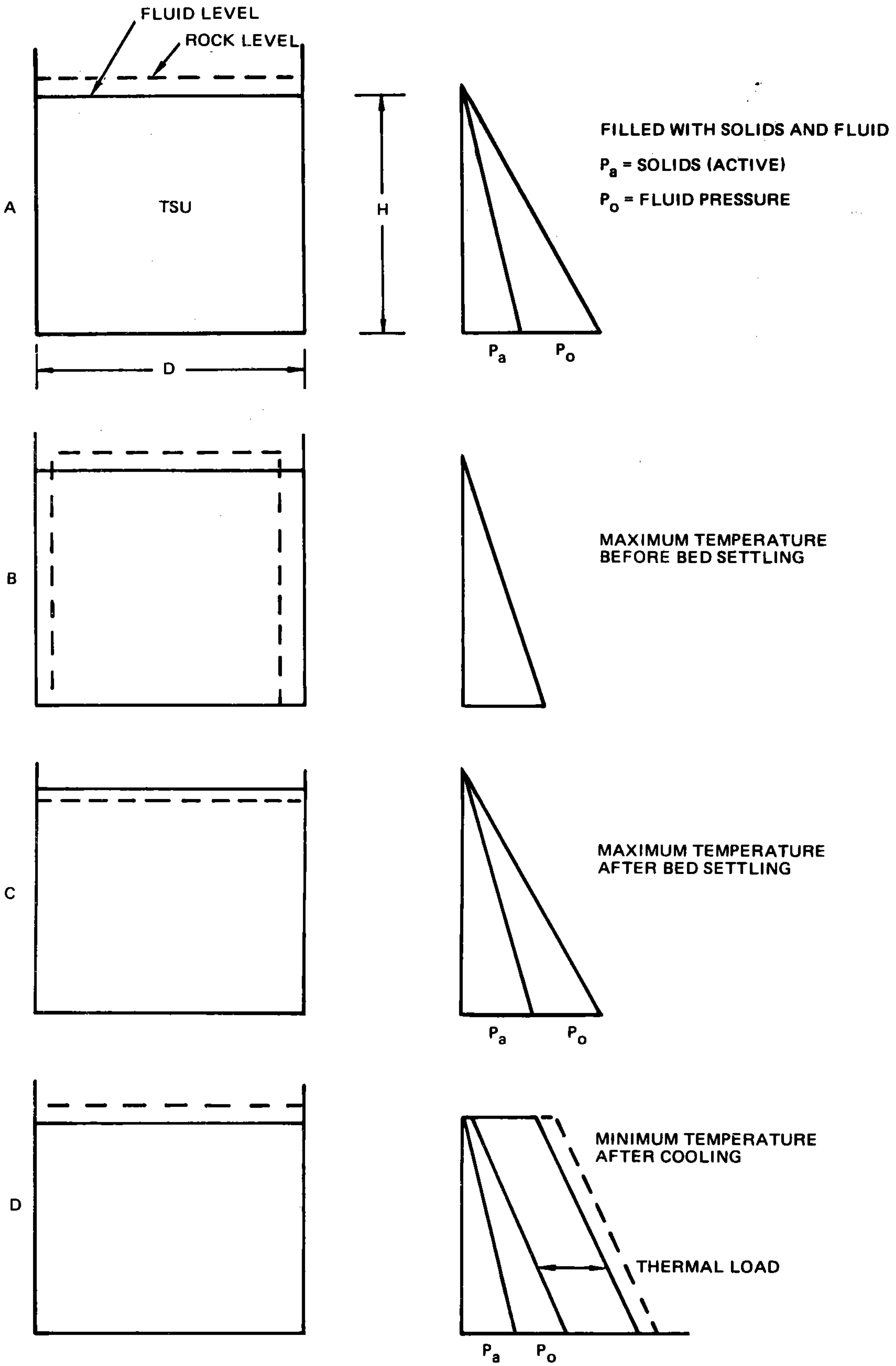


Figure 4-19. Structural Load Model for TSU Tank (Sheet 1 of 2)

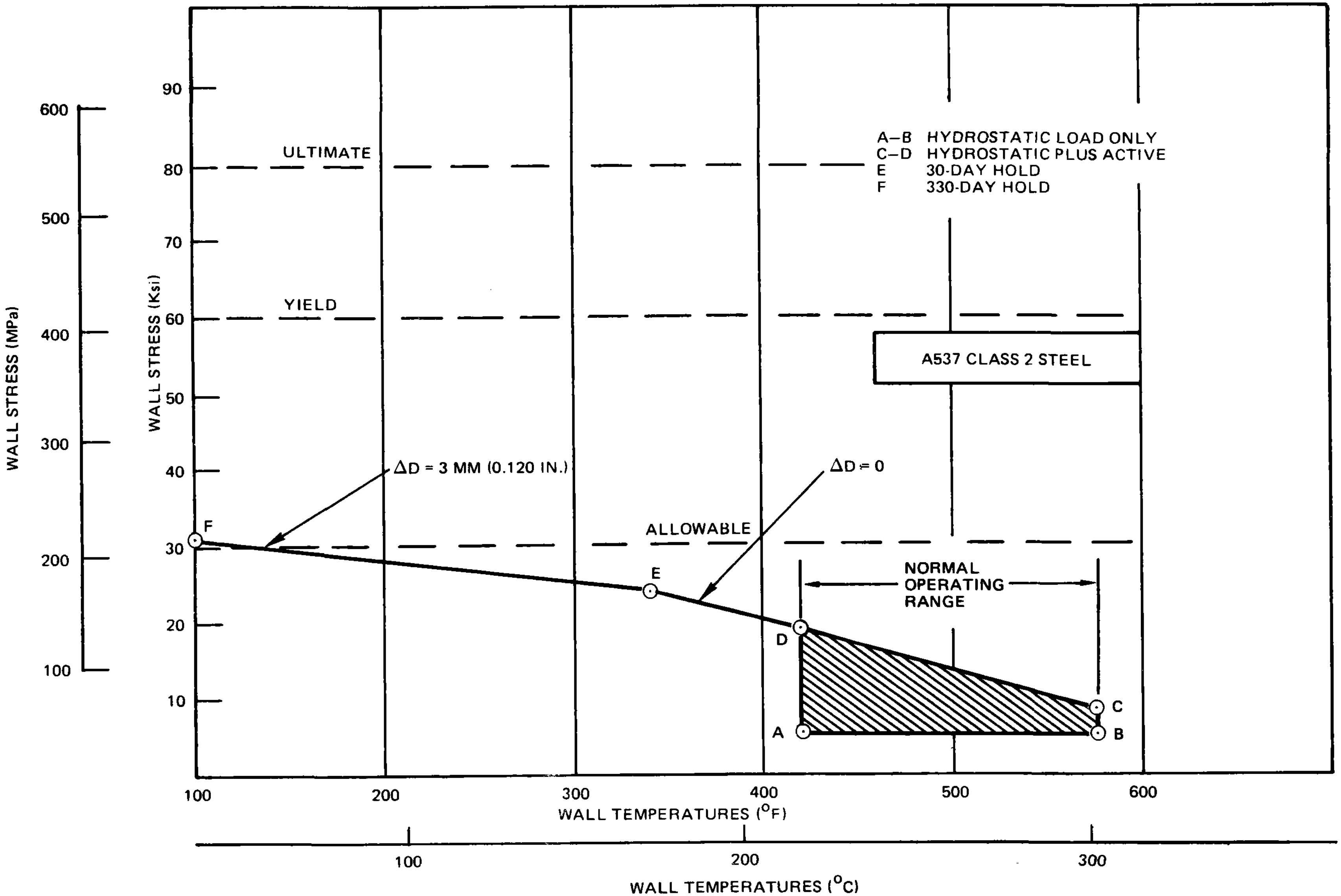


Figure 4-19. TSU Tank Structural Characteristics (Sheet 2 of 2)

The rock and the fluid present a pure hydrostatic load which varies with the height and effective density of the rock fluid combination. The resultant stress on the tank wall is given by the conventional hoop stress relationship and depends upon the wall thickness, the diameter of the tank, and the height H.

Loads During Tank Heating – During heating, the tank wall expands away from the bed because of the differential expansion of the two materials, as noted in the following:

Material	Linear Coefficient of Expansion (e)
Solids	7.92×10^{-6} °C^{-1} (4.4×10^{-6} °F^{-1})
Steel	11.7×10^{-6} °C^{-1} (6.5×10^{-6} °F^{-1})

When heated to the maximum temperature the result is a gap between the bed and the wall of approximately 8 mm (0.32 in.) and is depicted as Diagram B in Figure 4-19. If this gap were retained, there would be no added stress on the wall at the elevated temperatures and, in fact, the only source of load would be the fluid. However, it may be expected over a long period of time after many heating or cooling cycles that the rock and sand material will fill the gap. When this occurs, the worse case is that the bed will be packed from wall to wall at the maximum temperature with the same void fraction as when originally filled. The resulting load is again the active load plus the fluid load as in Figure 4-19, Diagram C.

Loads During Heat Extraction – When heat is extracted from the TSU, a thermal stress will be imposed on the wall depending upon the reduction in temperature and the relative elastic properties of the wall and rock bed. The thermally induced stress will be determined by the difference in the coefficient of expansion of the bed and wall, the temperature excursion, and the amount of deflection of the wall and rock bed. The wall deflection is established by the elastic modulus of steel. The bed is not an elastic medium, but a deflection value can be assigned to it that is representative of relative local movement near the wall that can be expected to occur with pressure exerted by the wall. The thermal stress at temperature T_{min} is,

$$F_t = E(e_{steel} - e_{bed})(T_{max} - T_{min}) - E\,\Delta D/D$$

where E is the elastic modulus for steel, e is the linear thermal coefficient of expansion, T_{max} is the upper temperature at which the gap between the bed and the wall is zero, and ΔD is the amount of radial contraction of the bed with respect to the wall. The total stress is the sum of the active, the fluid, and the thermal loads, as depicted as Diagram D in Figure 4-19.

Material of Construction – A comprehensive review of steels available as well as applicable code designations was made early in the present contract and is covered in detail in Reference 4-2. Structural values for various steels are given in Table 4-7. In investigating tank fabrication it was determined that, for relatively heavy wall stress limited structure, the higher-strength steels are the most economic to use. The TSU tank built for the SRE system tests used A 547 Class 2. No fabrication problems were encountered with this choice and at this point in time it is the choice for the Pilot and Commercial Plants. In addition to giving stress limits, the API Standard 650 code has a lower minimum gage limit as given in Table 4-8.

Pilot Plant TSU Tank Design – From the thermal considerations in the previous subsections, a single tank, with a bed height of 12.50m (41.0 ft), a diameter of 15.24m (50.0 ft), with a 0.25-void fraction, using Caloria HT43 was designed. Solids are nominally 25 mm (1 in.) gravel and 1.5 mm (1/16 in.) silica sand in a 2:1 (gravel-sand) ratio.

Bed and tank height were established considering soil-loading capacity of the Barstow area and the SRE test results. As noted in Section 4.2.1.1, soil-bearing capacity at the Barstow site varies from 0.072 MPa (1,500 psf) at 0.61 m (2 ft) depth to 0.48 MPa (10,000 psf) at 3.05m (10 ft) depth. For a bed 12.5m (41 ft) high with a total weight of 4,530 Mg (4,990 tons) solids plus 448 Mg (493 tons) Caloria HT43 (equivalent to 138,700 gal or $525m^3$ at 21°C, (70°F)), the total weight is 4,980 Mg (5,483 tons). The bearing pressure, for a 15.24m (50 ft) diameter, or $1,822m^2$ ($1,963\ ft^2$) bearing circle area, is then 0.270 MPa (5,586 psf). Allowing for extra

Table 4-7

ALLOWABLE STRESSES IN API STANDARD 650

API Reference Section	API Maximum Thickness (in.)	Material	S_d(psi) First Course Room Temp	S_d(psi) First Course 600°F	S_d(psi) Upper Courses Room Temp	S_d(psi) Upper Courses 600°F	Hydrostatic S_t(psi) First Course	Hydrostatic S_t(psi) Upper Courses
Basic	0.5	A36	21,000	21,000	21,000	21,000	21,000	21,000
Basic	0.5	A283 Grades C&D	21,000	-	21,000	-	21,000	21,000
Basic	0.5	A285 Grade C	21,000	16,980	21,000	16,980	21,000	21,000
Appendix D	1.5	A36	21,000	21,000	21,000	21,000	23,000	23,000
Appendix D	1.5	A283 Grades C&D	21,000	-	21,000	-	23,000	23,000
Appendix D	1.5	A285 Grade C	21,000	16,980	21,000	16,980	23,000	23,000
Appendix G	1.75	A573 Grade 70	26,300	-	28,000	-	28,000	30,000
Appendix G	1.75	A537 Class 1	26,300	25,500	28,000	27,150	28,000	30,000
Appendix G	1.75	A633 Grades C&D	26,300	-	28,000	-	28,000	30,000
Appendix G	1.75	A678 Grade A	26,300	-	28,000	-	28,000	30,000
Appendix G	1.75	A537 Class 2	30,000	29,660	32,000	31,640	32,000	34,300
Appendix G	1.75	A678 Grade B	30,000	-	32,000	-	32,000	34,300
Appendix G	1.75	A537 Class 1 @75,000 psi	28,125	27,280	30,000	29,100	30,000	32,150
Appendix G	1.75	A537 Class 2 @85,000 psi	31,875	31,520	34,000	33,620	34,000	36,430

Table 4-8

MINIMUM SHELL THICKNESS
(From API Standard 650)

Nominal Tank Diameter (Ft)	Minimum Nominal Plate Thickness (in.)
Smaller than 50	3/16
50 to 120 (excluding 120)	1/4
120 to 200 (inclusive)	5/16
Over 200	3/8

Notes:

1. The nominal thickness of shell plates refers to the tank shell as constructed. The thicknesses specified are based on erection requirements.
2. Nominal tank diameter shall be the centerline diameter of the shell plates, unless otherwise specified by the purchaser.

weight of the steel tank and roof (1% or less of the total bed load) results in a bearing pressure of 0.273 MPa (5,700 psf). This requires that the ground be excavated 1.8m (6 ft) to support the tank weight. The result is a sump area in case of inadvertent fluid leakage.

Computation of the total pressure at the bottom of the tank according to the active load principal assumes that the angle of internal friction, B, in the packed bed is 30 deg.

Coefficient of active pressure $K_a = \tan^2 (45° - \frac{B}{2}) = 0.333$ at 191°C(375°F)

W_B, buoyant density of the bed solids $= (1\text{-}f)\ (\rho_s - \rho_1)$
$= 0.75\ (168\text{-}45.6) = 92.2\ \text{lb/ft}^3\ (1477\ \text{Kg/m}^3)$

P_o, hydrostatic fluid pressure $= \rho_L H/144 = 45.6 \times 41/144 = 12.97$ psi (0.089 MPa)

P_a, active pressure of bed solids $= K_a W_B H/144 = 0.333 \times 92.2 \times 41/144$
$= 8.74$ psi (0.060 MPa)

$P_w = P_o + P_a = 12.97 + 8.74 = 21.71$ psi (0.15 MPa)

F_a, tensile hoop stress in tank wall due to active pressure for the lower coarse tank wall thickness (t_{max}), using conventional thin wall cylinder formula becomes:

$$F_a = \frac{P_w(D/2)(12)}{t_{max}} = \frac{21.71 \times 25 \times 12}{0.75} = 8,680 \text{ psi (60 MPa)}$$

The thermal portion of the stress at cooldown is independent of wall thickness, but the total stress is a function of thickness. Increasing wall thickness serves to reduce the portion of the total stress due to the active load but does not affect the component of thermal stress.

For the Pilot Plant, a wall thickness of 19 mm (0.75 in.) for the lower course was selected to provide a substantial margin from the yield and ultimate values for the A537 Class 2 steel.

Within the normal operating range the minimum temperature is estimated to be 191°C (375°F)*. The most conservative value of ΔD is zero, which results in a thermal stress at this condition of:

$$F_t = E(e_{steel} - e_{bed\ solids})(T_{max} - T_{min})$$

$$F_t = 29 \times 10^6 \times 2.1 \times 10^{-6} \times 200 = 12,180 \text{ psi (84 MPa)}$$

$$F_{tot} = 12,180 + 8,680 = 20,860 \text{ psi (144 MPa)}$$

For A537 Class 2 steel, the API values are:

F_u (ultimate) = 80,000 psi (551 MPa)

F_y (yield) = 60,000 psi (413 MPa)

F_o (allowable) = 30,000 psi (207 MPa)

Thus, the maximum stress during normal operation, 20,860 psi (144 MPa), is well below the design allowable of 30,000 psi (207 MPa).

*191°C(375°F) represents a TSU temperature drop of 27.8°C(50°F) below the minimum operating temperature, which requires several weeks of inactivity.

The worst possible stress condition that could occur would result from cooling the TSU to ambient temperature. It can be expected that, for large temperature excursions, higher wall stresses may result in some localized movement or readjusting of the sand and rock adjacent to the wall. If this occurs, then a logical minimum amount of movement is on the order of the size of the smallest sand particle, nominally 1.5 mm (0.06 in.). If this value is used, then ΔD becomes 3 mm (0.12 in.) and the induced thermal stress due to extreme cooldown from 302°C (575°F) to 38°C (100°F) is:

$$\text{Thermal stress, } F_t = E(e_{steel} - e_{bed})(T_{max} - T_{min}) - E\,\Delta D/D$$

$$F_t = 29\text{x}10^6 \text{ x } 2.1\text{x}10^{-6} \text{ x } 475 - 29\text{x}10^6 \text{ x } 0.12/600$$

$$F_t = 23,100 \text{ psi } (159 \text{ MPa})$$

The total stress (thermal plus pressure), F_{tot}, upon cooldown becomes:

$$F_{tot} = 23,100 + 8,680 = 31,780 \text{ psi } (229 \text{ MPa})$$

This stress is close to the allowable and provides a true stress margin of over 28,000 psi (192 MPa) below the yield and 48,000 psi (330 MPa) below the ultimate. These values are considered adequate and safe since they represent a considerably higher margin between allowable stress and yield and ultimate than many lower strength steels listed in Table 4-7.

The relationship of the various stress excursions is shown in Sheet 2 of Figure 4-19. Points A and B represent the stress level on the tank wall with hydrostatic loading only, which is the minimum load on the wall. After a period of recycling, the active load may be added to the hydrostatic load. The worst case assumes that the active load will be imposed at the maximum temperature, 302°C (575°F), and that thermal contraction only of the bed will occur (i.e., ΔD=0), which results in the operating line C-D. The region ABCD thus represents the limits of the possible operating envelope with normal daily charging and extraction.

Since occasional downtime is inevitable, Point E in Figure 4-19, Sheet 2 represents the temperature and stress level after 30 days of inactivity, at which point the temperature is estimated to be 171°C (340°F). Line E-F represents the increase in stress resulting from an ultimate cooldown to 38°C (100°F). The stress value at Point F assumes that the bed contracts an average value of 1.5 mm (0.06 in.) in the region adjacent to the wall on each side some time during the cooldown period.

The stress model and the selected values of temperature are very conservative. It is unlikely that the TSU would ever cool down to 38°C (100°F). To reach this temperature will take about 330 days of continuous nonoperation. This much inactivity would indicate complete abandonment of the Pilot Plant.

A complete listing of the Pilot Plant wall thicknesses based upon the above analysis is given in Figure 4-20.

Tank Foundation

Another factor affecting the tank height selection, in addition to thermal and heat loss considerations, is the allowable bearing pressure on the ground on which the tank is placed. The alternatives considered are to set the tank on the soil directly or to set it on a concrete base. Many large storage tanks are simply set on dry prepared soil and this has proved to be satisfactory. It has been found that tanks bolted to solid concrete foundations have experienced failures at the bolt interface and bolts sometime have pulled out of the concrete foundation, in other cases, ruptured at the bolt location during severe earthquakes.

Discussions with the Los Angeles Department of Water and Power, a large municipal utility which has constructed numerous large liquid tanks, revealed that the most satisfactory method of setting tanks based upon experience is to shape the tank bottom into a conical surface (analogous to the bottom of a champagne bottle) with the grade angle about 1 or 2%. This has been proven to be the most satisfactory mounting. During an earthquake the slight cone angle is stabilizing in preventing lateral movement. Vertical movement cannot be restrained and tank damage is not sustained providing the pipe connections are flexible enough.

Perhaps the most severe earthquake structural problem in all-liquid enclosed tanks is roof failure resulting from liquid sloshing. The long span and depth

SHELL COURSE SCHEDULE (ASTM 537 CLASS 2 STEEL)				
COURSE	HEIGHT M (FT)		PLATE THICKNESS MM (IN.)	
1 (BOTTOM)	1.83	(6)	19.0	(0.75)
2	1.83	(6)	16.3	(0.64)
3	1.83	(6)	13.5	(0.53)
4	1.83	(6)	10.7	(0.42)
5	1.83	(6)	7.9	(0.31)
6	1.83	(6)	6.4	(0.25)
7 (TOP)	2.44	(8)	6.4	(0.25)

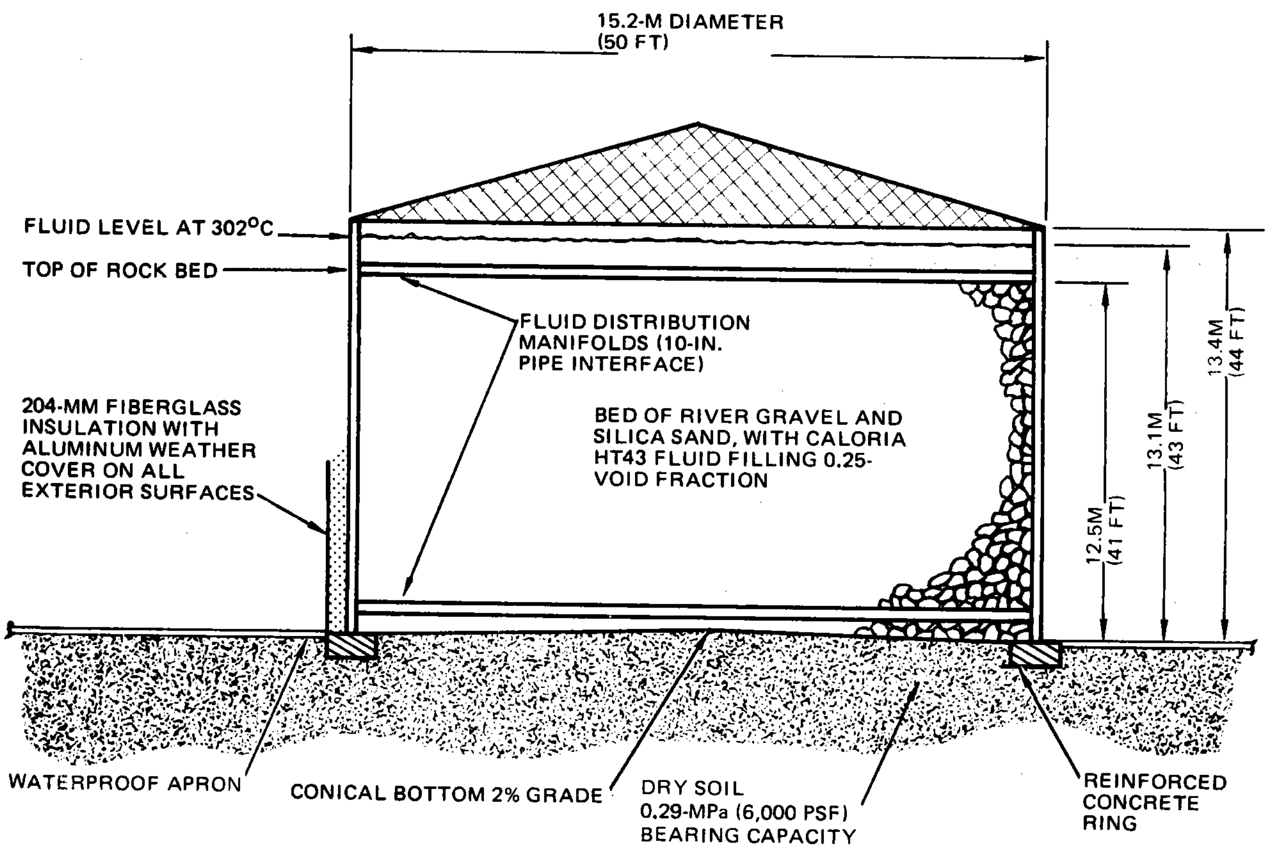


Figure 4-20. Design for 10-MWe Pilot Plant Thermal Storage Unit

of tank produces a wave motion during an earthquake that will rupture the roof-wall joint and warp the tank, in some cases requiring major rebuilding. The Los Angeles Department of Water and Power knows of no method to avoid this because of the extremely high pressures induced. However, in the dual-medium tank, the very small amount of free liquid above the bed will preclude this failure mode.

The Barstow soil report, quoted in Section 4.3.1.1, indicates an increase in bearing pressure with depth. The foundation design, then, consists of excavating to a few feet, sufficient to attain the bearing pressure commensurate with the tank height previously determined from stress limitations, and providing the tank with a conical bottom. The excavated volume encircling the tanks acts as a containment dike for any spillage of heat transfer fluid.

4.3.1.3 Design Description

The Pilot Plant TSU tank has been designed to meet or exceed all structural and safety requirements of the Solar Central Receiver Power Plant, as well as the requirements set forth in Section 4.3.1.1. The dimensions and other characteristics of the selected single tank design appear in Table 4-10 and Figure 4-20.

Table 4-10 (Page 1 of 2)

DESCRIPTION OF DESIGN FOR 10-MWe PILOT PLANT THERMAL STORAGE UNIT

TSU Configuration

One cylindrical tank, axis vertical, installed above ground, supported on dry soil of 0.29 MPa (6,000 psf) bearing strength by excavation to 1.83m (6 ft) below grade.

Tank Dimensions (Heights Measured at Circumference)

Inside diameter 15.25m (50 ft)
Overall height 13.4m (44 ft)
Packed bed height 12.5m (41 ft)
Free fluid surface height 13.1m (43 ft) at 302°C (575°F)
Effective height of top manifold 12.4m (40.6 ft)
Effective height of bottom manifold 0.23m (0.75 ft)
Effective height of seal steam manifold 0.9m (3.0 ft)
Tank cross-sectional area 182m^3 (1963 ft^2)

Table 4-10 (Page 2 of 2)

DESCRIPTION OF DESIGN FOR 10-MWe PILOT PLANT THERMAL STORAGE UNIT

Tank Thermal Performance

Extractable capacity after 20-hr hold time: 103.8 MWHt
Design storage temperatures: maximum 302°C (575°F)
minimum extraction 294°C (560°F)
minimum 219°C (425°F)
Thermal rates: maximum charge 30 MWt, maximum extract 32.1 MWt
minimum charge 1.5 MWt, minimum extract 3.1 MWt
Heat losses during 24-hour hold less than 3% of extractable capacity

Solid Storage Medium

25 mm (1 in.) river gravel and 1.5 mm (1/16 in.) No. 6 silica sand in 2:1 ratio, with 0.25-bed void fraction.
Superficial bed volume per tank 2447m^3 (86,400 ft^3)
Weight of solids per tank 4,530 Mg (4,990 tons)
Total weight of solids (all tanks) 4,530 Mg (4,990 tons)

Liquid Storage Medium

Caloria HT43 heat transfer fluid
525 m^3 (139,000 gallons) at 21°C (70°F)
Two manifolds, each with 5,900 holes of 3.1-mm (0.125 in.) diameter uniformly spaced over cross-section
One seal steam manifold

Tank Structural Details

Fabricated of A537 Class 2, structural steel with field-welded construction
Upward conical bottom plate 6.35-mm (0.25 in.) thick, 2% slope
Plate thickness for 1.83m (6 ft) high shell courses varies from 19mm (0.75 in.) at bottom to 6.35mm (0.25 in.) at the top
Roof is single skin with trusses, 1-in-12 pitch conical
Roof and sides covered with 204 mm (8 in.) fiberglass blanket insulation with corrugated aluminum weather cover

Interfaces (Flow Penetrations)

Caloria HT43 piping for primary thermal charging and extraction: top and bottom manifold, each 10-in. pipe
Caloria HT43 for night-time seal steam, 1.5-in. pipe in sidewall at 0.91m (3 ft) height
Nitrogen gas for ullage gas blanket 7-in. pipe into roof.

4.3.1.4 Nomenclature

A/V = Area/volume ratio for heat transfer in bed

B = Angle of internal friction in packed bed

C_{PL} = Fluid specific heat

C_{PS} = Solid specific head

C_s^* = Effective solid specific heat

D = Nominal diameter of tank

D_p = Effective particle diameter

E = Young's modulus

e = Coefficient of thermal expansion

f = Packed bed void fraction (volume of voids divided by total bed volume

F_a = Tensile wall stress

F_d = Allowable design stress

F_u = Ultimate strength

F_y = Yield strength

FS_u = Factor of safety on ultimate strength

FS_y = Factor of safety on yield strength

G = Specific gravity of liquid

H = Tank height

H_T = Total tank height

h = Heat-transfer coefficient

K_a = Coefficient of active pressure

P = Pressure

P_a = Active pressure of bed solids

P_o = Hydrostatic fluid pressure

P_w = Total pressure on tank walls = $P_o + P_a$

Q_E = Extraction rate

Q_s = Total extractable heat energy from tank

SRE = Subsystem research experiment

T_b = Minimum fluid temperature

T_t = Maximum fluid temperature

T_L = Fluid temperature

$T_{LO}(x)$ = Initial fluid temperature distribution

$T_{LI}(t)$ = Incoming fluid temperature

T_S = Solid temperature

TSU	=	Thermal storage unit
$T_{S_O}(x)$	=	Initial solid temperature distribution
t	=	Time
t_{max}	=	Maximum tank wall thickness (lowest course)
U	=	Actual (not superficial) fluid velocity
V_L	=	Actual fluid velocity
$V_{L_I}(t)$	=	Actual fluid velocity at x = 0
W_B	=	Buoyant density of bed solids
X	=	Axial distance in tank (normally vertical)
X	=	The distance of the variable design point from the bottom of the course under consideration
ρ_L	=	Fluid density, kg/m^3
ρ_S	=	Solid density, kg/m^3

4.3.2 Ullage Maintenance Unit

4.3.2.1 Requirements

The purpose of this unit is to provide a controlled-pressure, oxygen-free gas atmosphere above the fluid in the thermal storage unit (TSU). It is necessary to have an oxygen-free gas above the heat-transfer fluid surface to prevent fire hazards and long-term oxidation of the fluid. The ullage pressure must be controlled within a moderately narrow band to avoid underpressurizing or overpressurizing the tank. When the fluid (Caloria HT43) is heated from 218° to 302°C (425° to 575°F), the volume expands by about 9.5%. If the gas in the ullage space were not released, the pressure would rise above the allowable upper limit. The gas released must be replaced or stored and returned to the ullage space during the cooling cycle since an equal amount of gas is required to prevent the pressure from going below the allowable lower limit as the fluid cools and contracts.

These functions must be accomplished in as simple a manner as possible with minimum capital, maintenance, and operating costs. The design also must permit reliable operational control and have the flexibility and measurement capabilities needed for the 10-MWe Pilot Plant. Data obtained with the Pilot Plant will be used both for demonstrating performance at the 10-MWe level, and to permit realistic design scale-up to commercial size systems.

The major specific requirements on the ullage maintenance unit are as follows:

A. The gas in the ullage space must not react with the fluid. Therefore oxygen and air must be excluded from the entire system and an inert gas such as nitrogen or carbon dioxide or a reducing gas must be substituted.

B. A control system with suitable backups must be provided to ensure that the pressure in the ullage space will remain within the gage pressure range of zero (i.e., at the ambient absolute pressure) to 2 KPa (0.30 psi). A pressure below ambient is not tolerable because it might permit air to enter the tank.

C. Since the emerging gases will be hot and will carry with them vapors given off by the fluid, they must be cooled and the condensate either returned to the tank or properly disposed of. Cooling the gas is required to accommodate storage of the gas in one form or another.

D. The products of thermal degradation of the fluid are mostly low molecular weight hydrocarbons (along with a small amount of high molecular weight substances). These volatile fractions will evaporate into the ullage space and must be disposed of without creating a pollution problem. These degradation products are produced at a rate dependent upon the time-temperature history of the fluid and must be removed as produced, either by stripping them from the ullage gases or by removal in the fluid maintenance unit (described in Section 4.3.3).

4.3.2.2 Analysis/Trades

Many types of ullage maintenance units could be considered. The following alternatives were examined and evaluated:

A. Nonrecovered Gas Systems
   - Using GN_2
   - Using CO_2
   - Using an inert gas generator

B. Condensible Vapor Systems
   - Using water
   - Using a hydrocarbon such as heptane or hexane
   - Using other compounds

C. Recovered Gas Systems

- Storage at TSU pressure (gas holder)
- Storage of compressed gas

Summaries of the description and evaluation of each of these types of units are given in the following subsections.

Nonrecovered Gas System with Nitrogen

This unit (Figure 4-21) provides for venting the cooled gases during a heating cycle with the addition of fresh nitrogen during the cooling cycle. Nitrogen is obtained from a liquid nitrogen tank with an evaporating heat exchanger. Low boiling thermal degradation products of the fluid which are not condensed are vented to the atmosphere unless they are conducted through an activated carbon adsorption bed (not shown) where they are extracted and ultimately recovered. This concept removes the low boiling fractions vaporizing from the fluid, and the gas returning to the ullage space is completely devoid of these low boilers. Recent tests conducted at Rocketdyne giving thermal degradation rates have shown that it would be impractical to remove all of the low boiler products from the vented gases using a carbon adsorption bed, due to the relatively large quantities produced.

Commercial nitrogen gas may contain as much as 0.1% oxygen unless special purification is provided. If an ullage maintenance unit is used which recycles the inert gas, this small amount of oxygen can be tolerated because it reacts with the fluid on initial use only, i.e., the system is self-purifying. This is not true in a recovered ullage gas unit, where the 0.1% oxygen is resupplied each time a cycle is completed. A simple approximation would be that after 100 cycles, the oxygen supplied is equivalent to refilling the ullage space once with 10% oxygen gas. Since the fluid is used repeatedly while the inert gas is not, the oxygen impurity effect would be cumulative.

The above unit has the following two major disadvantages: (1) high operating costs, and (2) the possibility that the heat transfer fluid will slowly oxidize.

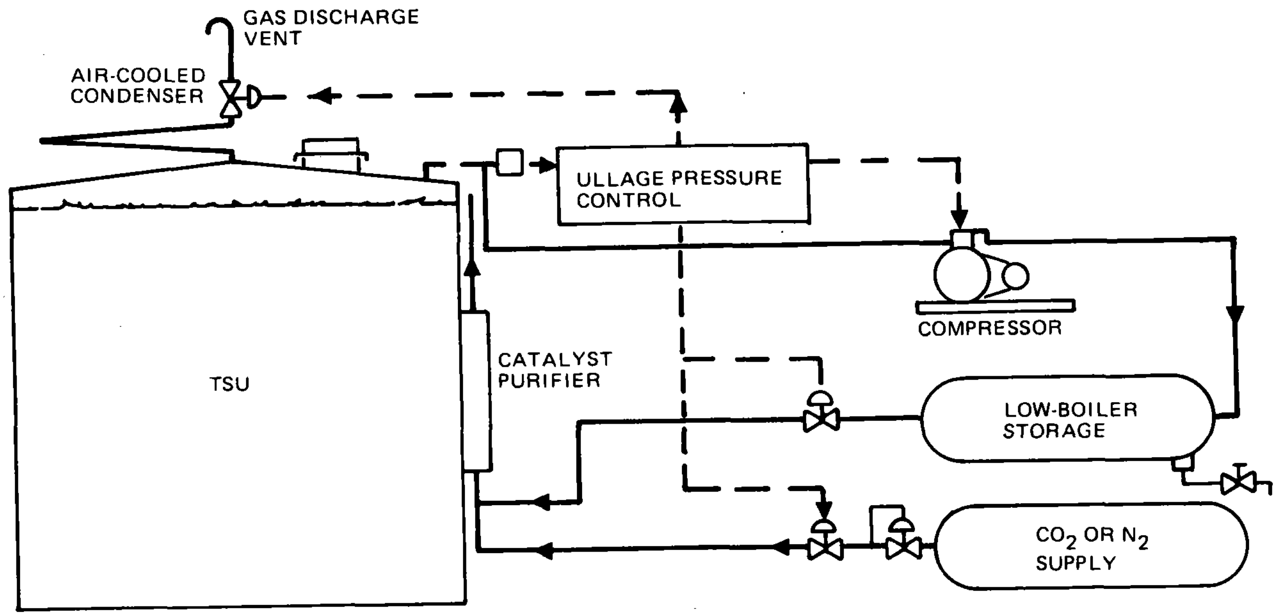


Figure 4-21. Candidate Ullage Maintenance Unit (Unrecovered Gas Type)

Nonrecovered Gas Unit with Carbon Dioxide

This unit would be conceptually identical to the nitrogen unit described in the previous subsection. Similar considerations which are mentioned above for nitrogen are applicable except for differences in the raw material cost.

The oxygen content of commercial carbon dioxide has not been established. A possible solution for both nitrogen and carbon dioxide is to pass the mixture over a heated catalyst to react the trace oxygen with the low boilers, giving CO_2 and H_2O (see Figure 4-21). The carbon dioxide itself should be very stable at the maximum temperatures encountered on the thermal storage subsystem. The main disadvantage is the high operating cost of the carbon dioxide which must be procured.

Inert Gas Generator Unit

In this unit the fluid maintenance unit (FMU) (described in Section 4.3.2) is used to remove all of the degradation products from the heat-transfer fluid. The volatile products of fluid degradation (Figure 4-22) are pumped into a degradation product storage tank and then into an inert gas generator where they are burned. The heat created by burning these byproducts then is used to run the FMU. The inert gases produced are used in part to provide ullage gas to the TSU during a thermal extraction cycle. During a thermal charging cycle, when the fluid is expanding, the gases coming from the ullage gas space at the top of the TSU are conducted into the inert gas generator along with a supply of air and a supply of the degradation products. These three substances are mixed and burned in the inert gas generator, thus incinerating any vapors which have been swept away from the ullage gas space. All of the degradation products are burned to form water and carbon dioxide which are expelled up the stack as inert flue gas.

This scheme has many advantages. However, it would require the development of a special inert gas generator which could accept the vented ullage gases and which would have a special heat exchanger to provide heat to the fluid maintenance system. This system is, therefore, not recommended for the present Pilot Plant design. However, it may be available for use in the commercial design because a greater time span is available for analysis, design, and construction. Also, data from the Pilot Plant regarding the

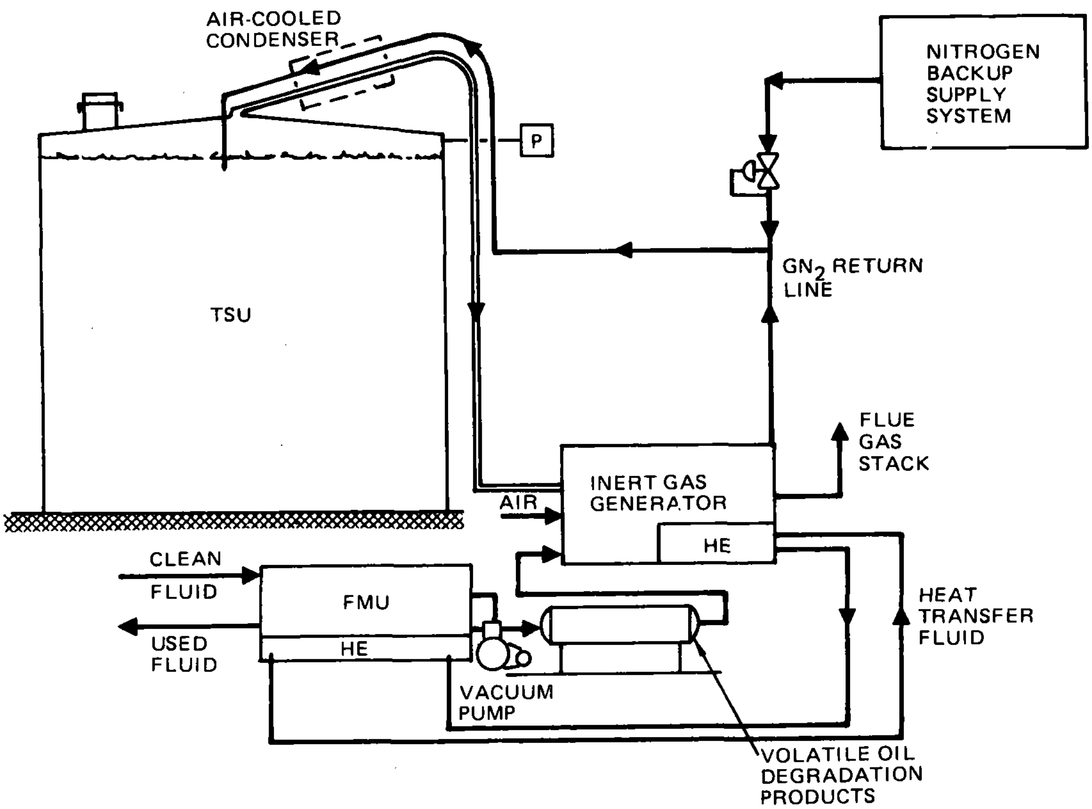


Figure 4-22. Simplified Schematic Diagram of an Ullage Maintenance System Using an Inert Gas Generator

rate of production of degradation products will be available for use in making an engineering analysis and an economic tradeoff to determine the relative desirability.

This system also suffers from the same objection as that described in the previous section in that the gas product by the inert gas generator is not completely free of oxygen (unless extreme precautions are taken).

Condensable Vapor Unit

This method uses an ullage gas which is condensed by compression during charging of the TSU and is boiled off to re-enter the ullage space as gas during discharging of the TSU.

The ullage maintenance system chosen for the Pilot Plant design incorporates some of the features of this system as explained later. The method would be an excellent choice except that the condensible gas would be swept out with the noncondensible degradation products of the heat-transfer fluid when they are vented or removed. A separation step would, thus, also be required to recover the ullage gas. This would be prohibitively complex and costly.

Recovered Gas Unit (at TSU Pressure)

Figure 4-23 shows a variable volume, constant-pressure gas holder which acts automatically to receive excess gas during the heating cycle and to return it during a cooling cycle. This type of unit has several advantages, principally the following:

A. Inherent simplicity of design; the gas holder acts as an automatic over-pressure relief valve.
B. Samples of low boiling material can be obtained by stripping them from the spent activated carbon.
C. It uses less electrical power than the proposed Pilot Plant design.

The principal disadvantages are as follows:

A. The gas holder would have a large volume and take up a fairly large space.
B. The rate with which degradation products are produced would make their removal using activated carbon economically prohibitive.

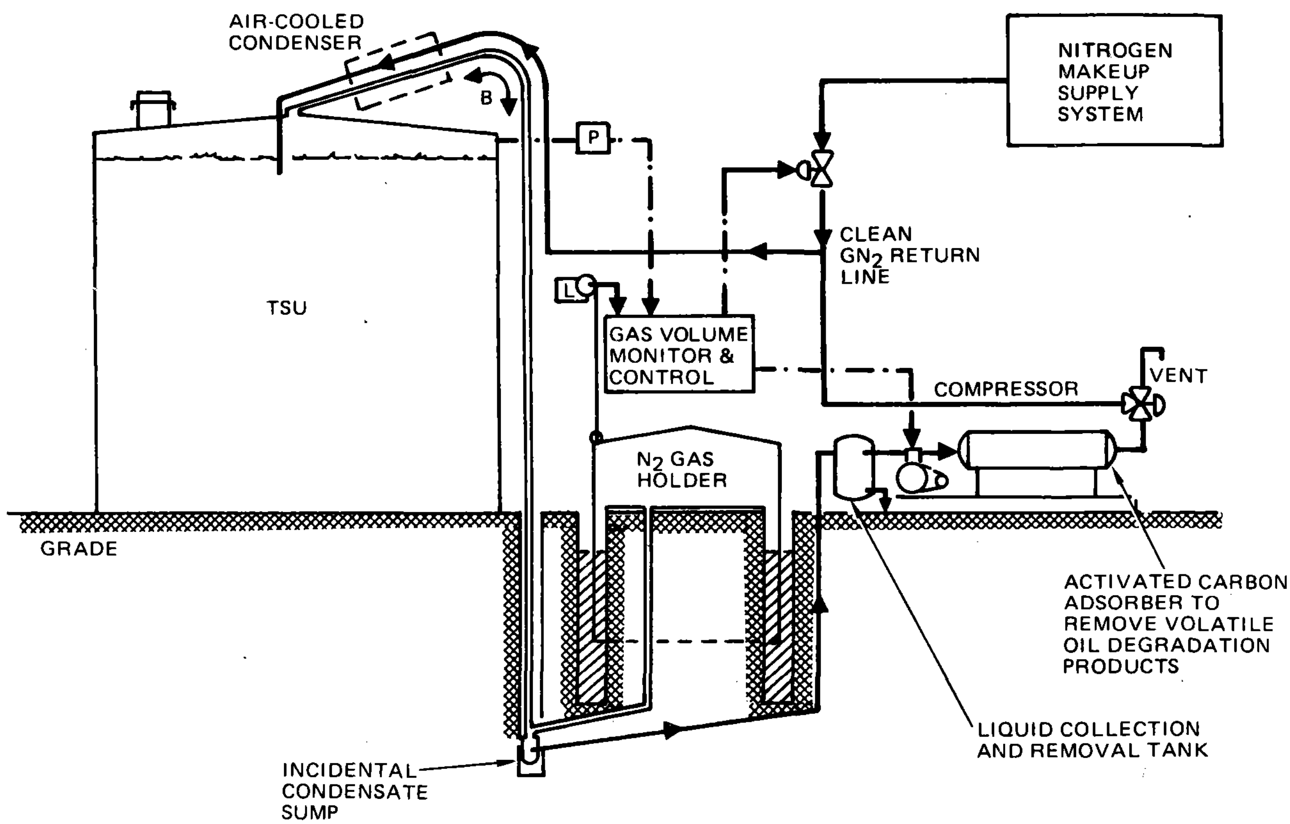


Figure 4-23. Candidate Ullage Maintenance Unit (Gas-Holder Type)

Recovered Gas Unit (With Compressed Gas Storage)

This type of ullage maintenance unit (Figure 4-24) was selected for the Pilot Plant design. It is described in detail in Section 4.3.2.3.

Rocketdyne has conducted long-term degradation rate experiments on Caloria HT43 and has operated a TSU as part of the SRE effort. The resulting data were used to aid in evaluation of candidate ullage maintenance schemes.

One of the key factors in choosing the ullage system is the relatively high rate at which gaseous degradation products collect in the ullage space and the method of disposal of these materials. Each of the above proposed ullage maintenance systems must deal with this problem, and, in addition, must satisfy the more obvious requirement such as maintaining the proper pressure in the TSU ullage space, etc. The method chosen deals simply and directly with the disposal problem.

Another factor is the operational flexibility afforded to make measurements and observations to be used later in scaling up to the commercial-size plant. Subsequent re-evaluations will be made, based upon the data collected from the Pilot Plant tests. The unit provides sufficient positive pressure to allow gas flow rates to be measured and provides accessible points where condensed liquid and gas samples can be taken conveniently.

4.3.2.3 Design Description

The ullage maintenance unit for the 10-MWe Pilot Plant design is of a type which recovers the ullage gas, compresses it for storage, and reuses it for each daily storage cycle. Details of the unit and its operation are given in this section.

General Description

The ullage maintenance unit is shown in conceptual form in Figure 4-24. The unit controls the pressure and vapor quality in the ullage space of the tank. Air is excluded since it will accelerate the heat-transfer fluid degradation rate, and could in extreme instances create a hazardous condition.

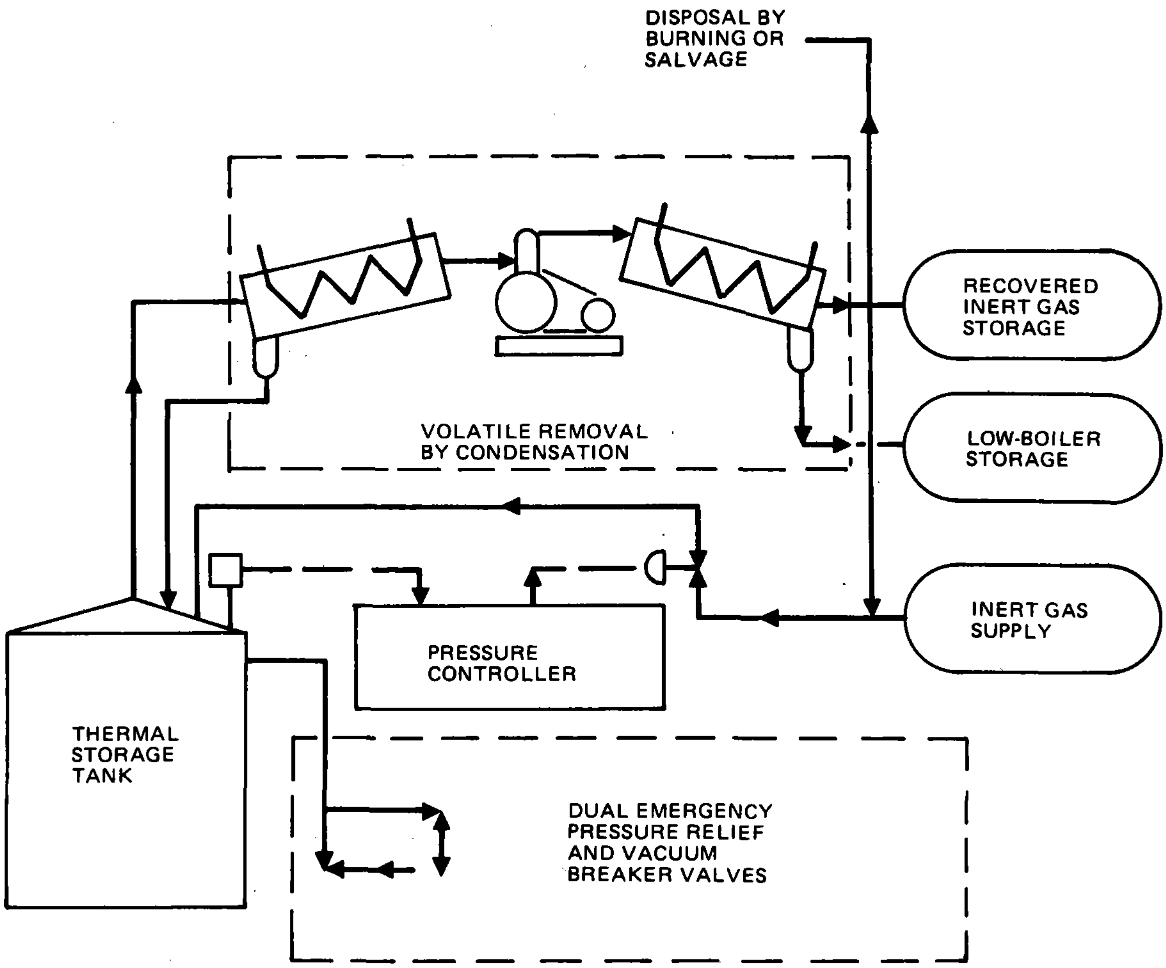


Figure 4-24. Conceptual Schematic Diagram of Ullage Maintenance Unit for 10-MWe Pilot Plant Preliminary Design

An inert ullage gas is added and vented as the fluid in the system expands and contracts during thermal cycling. Nitrogen was selected as the ullage gas for its economy and ready availability.

The ullage control unit features the following:

A. Ullage pressure automatically maintained at 10 cm of water above ambient atmospheric pressure.

B. A redundant system of emergency pressure and vacuum relief valves to protect the tank in case the pressure control system is unable to handle the necessary reduction or increase in gas flow into the ullage space.

C. Ullage gas storage at a working pressure of 1.2 MPa (175 psig) and with a gas volume of approximately 79.3 m^3 (2,800 ft^3) at 20°C and 1 atm, which is more than sufficient gas for one complete thermal charge of the TSU.

D. Measurement of inert gas flow into the TSU ullage space to allow monitoring and recording of system operation; TSU tank pressure and liquid level are also measured and recorded.

E. The major portion of the degradation products of the heat transfer fluid are of a volatile nature. The ullage maintenance system thus performs a dual function: (1) of maintaining an adequate supply (and pressure) of ullage gas in the ullage space, and (2) of removing fluid degradation products from the ullage space. It is anticipated that the ullage gas compressor and subsequent storage tank will remove by condensation all of the volatile degradation products except, for example, hydrogen methane, ethane, here defined as noncondensable (for purposes of this design). The anticipated rate of production of the noncondensables will be great enough to require that they be vented either by flaring to the atmosphere or by disposal in some more appropriate energy-saving manner. It is estimated that these noncondensable gases will be produced by the system at an average rate of 57 standard cubic meters per day (2,000 standard cubic feet per day).

Initially, the ullage space will be filled with gaseous nitrogen; however, as the ullage gas is reused over and over again the concentration of nitrogen in the ullage gas will decrease steadily due to the fact that it is lost to the atmosphere along with the noncondensables which are flared each day. This nitrogen gas is replaced by noncondensable vapors which are fed back into the ullage space during periods when the system is in a thermal discharge condition. Thus, eventually the ullage space in the TSU will be filled with noncondensable products of fluid degradation only and almost all of the nitrogen will have disappeared. An objection to using these gases in the ullage space is that they are flammable. However, adequate safeguards against their possible escape have been taken. Venting takes place only for burning in a makeup fluid heater or a remote flare tower in an emergency. The entire system, except for the ullage gas storage tank, is kept at a low positive pressure of only a few inches of water column above the surrounding atmospheric pressure. Any leakage is thus bound to be outward and extremely slow if present at all. If rapid venting is required or if rapid addition of inert gas is required, these operations can be performed in a safe manner by a system which has complete redundancy.

The oxygen content of the ullage gas will be monitored continually and an alarm will be given should the oxygen level reach to within striking distance of the flammability limit.

It is very doubtful that oxygen diffusing into the system (provided such were possible against the internal pressure) will ever reach more than trace concentrations because the degradation products of the fluid will contain some molecules having unsaturated chemical bonds which will be reactive enough to slowly react with any trace amount of oxygen, thus immediately removing them and preventing their accumulation.

The alternative to the above scheme is to vent the noncondensables each time the system is charged and to replace these gases with nitrogen gas as the thermal storage unit is discharged. This should maintain the nitrogen concentration in the ullage gases at a high enough level so that, should they escape from the system, they will form nonflammable mixtures with air in <u>all proportions</u>. At present, it has been decided that the decrease in hazard afforded by replacement of the noncondensables with nitrogen does not outweigh (1) the cost of the large amount of nitrogen gas used daily for replacement purposes, and (2) the increased difficulty of disposing of the degradation products which are now mixed with large amounts of N_2 gas. On the other hand, should the use of the extra gaseous nitrogen be justified, this will require virtually no change in the system configuration except to provide for sufficient liquid nitrogen each week to supply that which is consumed. Another factor, however, must be taken into consideration; noncondensable gases produced each day by the system will not be nearly so readily usable for burning or for disposal for some useful purpose because they will contain large amounts of gaseous nitrogen, which is not the case when the present system design is used.

Operating Description (Normal Storage Cycle)

In the following description, it will be assumed that the system is in operation and has come to a standard cyclic condition. Beginning at the start of a heating cycle, the heat-transfer fluid expands and raises the pressure in the ullage space above the outside ambient pressure as indicated by the differential pressure-measuring transducer and indicator DPI-18 (all components are referenced with numbers shown in Figure 4-25). The pressure controller uses this signal to turn on the compressor, C-80, provided this differential exceeds the first upper pressure limit. Gas and vapors are then pumped up through a simple air-cooled heat exchanger, HE-70, and down through a second one HE-71, (which is also the compressor intake manifold) into the compressor inlet. The "exchangers" are just lengths of uninsulated pipe. The fluid vapors are quickly condensed in HE-70 and the condensate runs by gravity back into the TSU tank, TA-1. The remaining gas, consisting of the inert gas combined with low boiler products of heat-transfer fluid degradation, passes down through a vertical pipe labeled HE-71 which further cools the gas to a temperature close to ambient; any further condensation is collected at the bottom in a sump tank and is removed periodically by opening a valve, HV-21. This is done when the liquid level reaches a given height as indicated by the sight-glass, SG-105, or signaled by the "high-level" liquid level probe (Table 4-11 has parts description).

The compressed gases are conducted into tank, TA-2, which is tall and narrow, allowing the heat of compression to be lost easily to the surrounding air, and allowing those low boilers which will condense on the walls to

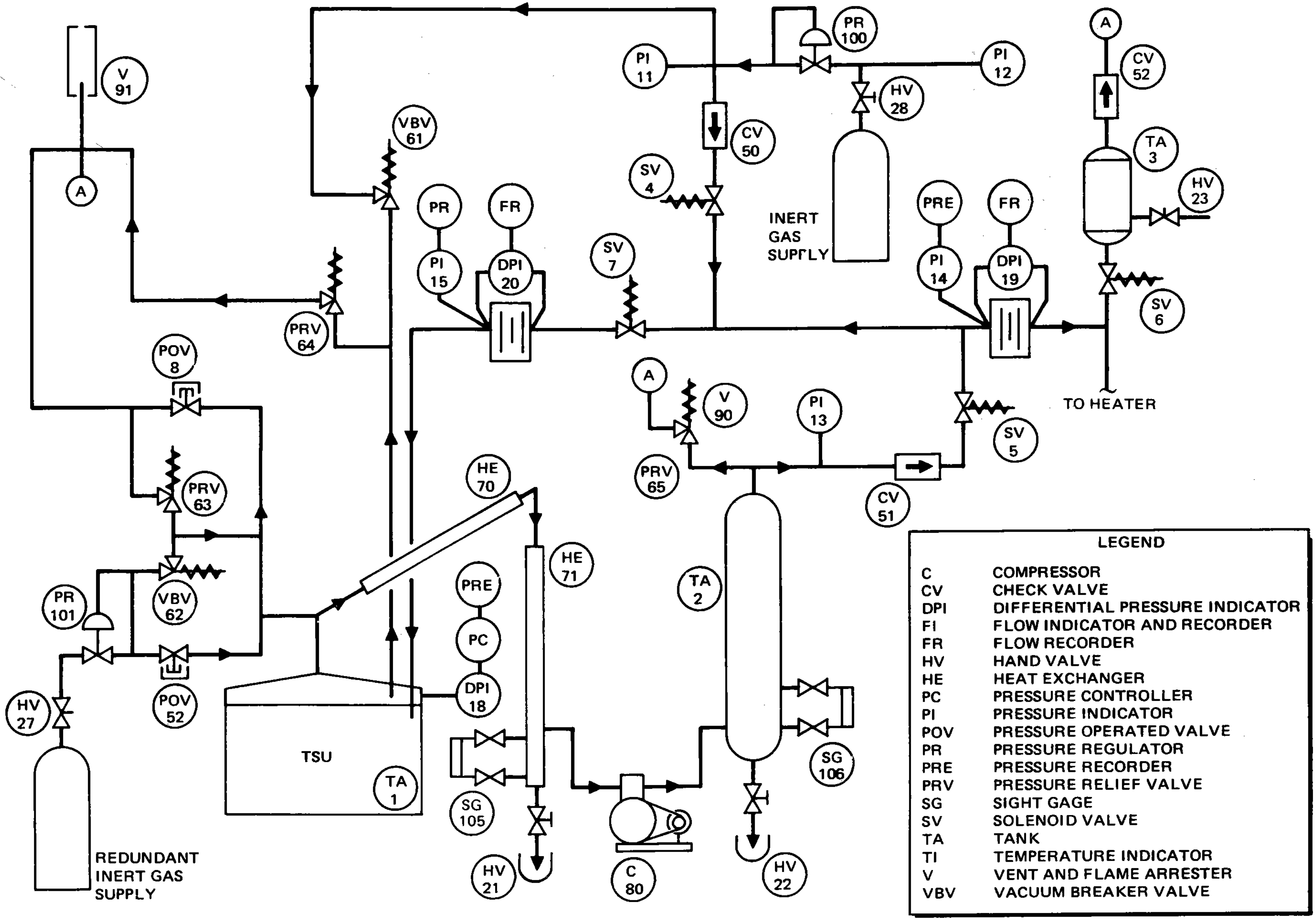


Figure 4-25. Ullage Gas Maintenance and Pressure Control System Preliminary, Design for 10-MWe Pilot Plant

Table 4-11 (Page 1 of 3)

PRELIMINARY COMPONENT LIST FOR ULLAGE MAINTENANCE UNIT

Component Number	Designation (Figure 4-25)	Function	Range, Size, and Comments
1	TA2	Thermal storage unit tank	15.2m ID by 13.4m high cylindrical tank
2	TA2	Inert gas storage tank	3785 liters (1,000 gal)
3	TA3	Gas sample storage tank	Short piece of larger diameter pipe having about 4 liter volume
4	SV4	Inert gas supply shutoff	1/2-in. NPT solenoid valve
5	SV5	Inert gas storage shutoff	1/2-in. NPT solenoid valve
6	SV6	Inert gas vent valve	1/2-in. NPT solenoid valve
7	SV7	TSU-gas isolation valve	1/2-in. NPT solenoid valve
8	POV8	Automatic pressure relief valve	3-in. diameter with Bettis actuator and 4-way solenoid valve
11	PI11	Pressure indicator, gas supply	0-1.38 MPa (0-200 psi) ordinary Bourdon pressure gage with long line
12	PI12	Pressure indicator, gas supply	0-34.5 MPa (0-5,000 psi) ordinary Bourdon pressure gage with long line
13	PI13	Pressure indicator, gas storage	0-1.38 MPa (0-200 psi) ordinary Bourdon pressure gage with long line
14	PI14	Pressure indicator for flow measurement	Remote indicating and recording
15	PI15	Pressure indicator for flow measurement	Remote indicating and recording

Table 4-11 (Page 2 of 3)

PRELIMINARY COMPONENT LIST FOR ULLAGE MAINTENANCE UNIT

Component Number	Designation (Figure 4-25)	Function	Range, Size, and Comments
18	DPI18	Differential pressure indicator	Recorder needed; sensing line must be equipped with a small inert gas bleed to provide a non-condensing temperature barrier
19	DPI19	Differential pressure indicator	With recorder
20	DPI20	Differential pressure indicator	With recorder
21	HV21	Condensate shutoff	0.86 MPa (125 psi) maximum working pressure
22	HV22	Condensate shutoff	1.38 MPa (200 psi) maximum working pressure
23	HV23	Gas sample shutoff	0.86 MPa (125 psi) maximum working pressure
27	HV27	Inert gas supply shutoff	34.5 MPa (5,000 psi) maximum working pressure
28	HV28	Inert gas supply shutoff	34.5 MPa (5,000 psi) maximum working pressure
50	CV50	Check valve	Standard type
51	CV51	Check valve	Low release pressure
52	POV52	Automatic pressurizing valve	Identical to POV8
61	VBV61	Vacuum breaker valve	
62	VBV62	Vacuum breaker valve	
63	PRV63	Pressure relief valve	
64	PRV64	Pressure relief valve	

Table 4-11 (Page 3 of 3)

PRELIMINARY COMPONENT LIST FOR ULLAGE MAINTENANCE UNIT

Component Number	Designation (Figure 4-25)	Function	Range, Size and Comments
65	PRV65	Pressure relief valve	
70	HE70	Heat exchanger	To be constructed from 3-in. diameter, schedule 40 pipe
71	HE71	Heat exchanger	To be constructed from 3-in. diameter, schedule 40 pipe
80	C80	Compressor	2.24 KW; 0.34 m^3/min. at 1.2MPa (3 hp, 12 SCFM at 175 psi)
90	V90	Vent and flame arrestor on storage relief	
91	V91	Torch vent	To be constructed and wired to pilot ignite auto
100	PR100	Pressure regulator	Standard unit used for pressure reduction on high-pressure gas bottles
101	PR101	Pressure regulator	Standard unit used for pressure reduction on high-pressure gas bottles
105	SG105	Sight glass	Standard sight glass 1m (3 ft) high with isolation valves
106	SG106	Sight glass	Standard sight glass 1m (3 ft) high with isolation valves

collect at the bottom where they are periodically removed by opening solenoid valve, HV-22. A sight-glass and liquid level probe indicate the liquid level. If required, these valve operations can be automated easily; this decision will depend upon the relative rate of low boiler production which will be determined from test data. The pressure in TA-2 will build up to 1.2 MPa (175 psig) by the end of the heating cycle; this is ensured by correct sizing of the tank. Should the pressure increase above this as indicated by pressure indicator PI-13, the excess will be released automatically (by the control system) through opening of valves SV-5 and SV-6. The quantity of gas thus released will be obtainable from the recording of the differential pressure indicator, DPI-19, and the composition of the gas can be obtained by sampling the gas in tank TA-3. When the heating cycle ends, the pressure will drop below the first upper pressure limit of 11.4 cm (4.5 in.) of water column and the controller will stop the compressor.

When the cooling cycle begins, the contraction of the fluid will cause the pressure in the ullage space to drop below the first lower pressure limit which will cause the controller to automatically open valves SV-5 and SV-7, allowing gas in TA-2 to flow through the check valve, CV-51, and through the flow measuring orifice, DPI-20, into the TSU ullage space where it will bring the pressure back up to an acceptable value. The pressurized gas in tank TA-2 will thus be used up, reaching a level of about 0.14 MPa (20 psig) by the time the end of the cold cycle is reached. Should the pressure fall below this value at any time, new gas from the inert gas supply will be supplied automatically by opening SV-4 for as much time as is required. However, if tank TA-2 is sufficiently large and if no gases are being vented, then new inert gas additions should not be necessary. A reserve of gas exists as liquefied gas at the bottom of the ullage storage tank in that as the pressure in the tank is reduced, this liquid will be revaporized. This liquefied gas is composed of some of the degradation products of the thermal storage fluid.

A new heating cycle can now begin and the above steps repeated.

Should any malfunction occur so that the ullage pressure might climb above the first upper-pressure limit to reach the second upper-pressure limit, then the controller would automatically open POV-8, causing the ullage gas to be vented through a vent stack equipped with a torch and ignition device. A torch is needed at the top of the vent since the cooling due to flow through the stack will reduce the temperature far enough to eliminate any chance of autoignition at the exit point. The flash point of Caloria HT43 is 204°C (400°F), autoignition temperature is 404°C (759°F). Should the pressure still increase slightly to a level above the third upper-pressure limit, then the pressure relief valves, PRV-63 and PRV-64, will release and vent the ullage space through the same stack. These, and other pertinent ullage pressure levels, are summarized in Table 4-12.

The above redundancy, and automatic operational procedure, is duplicated for the first, second, and third lower-pressure limits by the vacuum breaker valves, VBV-61 and VBV-62, and by the control system. A redundant inert gas supply is provided for vacuum breaker VBV-62 to ensure that enough inert gas is available to relieve any low-pressure condition by allowing the entrance of inert gas only.

There is an additional design consideration wherever the ullage maintenance system lines connect with the TSU. At these points vapors from the fluid will condense continually on the inside, colder wall of the pipe near the junction with the tank. These lines are thus constructed with a slight slope toward the tank to allow condensed fluid to return to the tank and to keep it from collecting at the other ends of these lines.

The compressor intake manifold has a number of sensors which are not shown in Figure 4-25. These are an oxygen content monitor sensor and a moisture content monitor sensor. In addition, a liquid-level switch is located just above the sight glass on the sump tank. A similar switch is placed just above the sight glass of the ullage gas storage tank, TA-2. (The above items are called out in Figure B-2 in Appendix B.)

Table 4-12

SUMMARY OF TSU ULLAGE PRESSURE LEVEL IMPLICATIONS AND SAFETY FEATURES

Meaning or Action Indication of Ullage Pressure Level	Pressure in TSU Ullage, cm(in.) of water (gage)
Tank Structure Design Yield Point	40 (16)
Tank Test Pressure	20 (8)
Higher High-Pressure Warning by Digital Data Logger	17.8 (7.0)
Pressure Relief Valve Setting	17.8 (7.0)
High-Pressure Switch Alarm	15.2 (6.0)
High-Pressure Warning by Digital Data Logger	14.0 (5.5)
Upper Limit of Nominal Operating Pressure	14.0 (5.5)
High-Pressure Relief Switch Closes	11.4 (4.5)
Nominal Pressure	10.2 (4.0)
Low-Pressure Supply Switch Closes	8.9 (3.5)
Lower Limit of Nominal Operating Pressure	7.6 (3.0)
Low-Pressure Warning by Digital Data Logger	6.4 (2.5)
Low-Pressure Switch Alarm	5.0 (2.0)
Vacuum Breaker Valve Relief Setting	2.5 (1.0)
Lower Low-Pressure Warning by Digital Data Logger	2.5 (1.0)

4.3.3 Fluid Maintenance Unit

4.3.3.1 Fluid Life and Maintenance Requirements

The fluid maintenance unit comprises one-half of a system designed to keep the heat-transfer fluid in satisfactory condition for the continual operation of the thermal storage subsystem (the other half is integrated into the ullage maintenance unit). The basic functions of the fluid maintenance unit are: (1) the removal of very low-volatility compounds (the polymerized material formed over long periods of time in the hot fluid via pyrolysis) and solids, and (2) the addition of fresh fluid to maintain a constant fluid inventory. The results of the prequalification tests, discussed in Section 6.2, have led to the choice of Caloria HT43 for the Pilot Plant and provided a rate equation for the fluid loss by volatilization of species produced by thermal cracking. Although the SRE fluid degradation tests covered a period equivalent to 2 yr of Pilot Plant operation, measurements of kinematic viscosity and gel permeation chromatograph analyses (GPC) have indicated no significant polymerization rate in the Caloria (Section 6.2.2).

The fluid maintenance unit will be required to:

A. Keep the level of polymerized materials below a given value. An acceptable level for Caloria HT43 is probably at least 10 to 20%; a requirement of 10% was used.

B. Remove suspended solids from the circulating fluid. The filters will be located in the thermal storage main flow line, upstream of the charging pumps.

C. Provide fresh heat-transfer fluid to make up for materials removed from the circulating fluid.

D. Provide virtually automatic operation using a simple direct method which requires relatively inexpensive equipment.

E. Use state-of-the-art technology and, as far as practicable, use existing commercial components.

F. Use only a modest quantity of parasitic electrical power and thermal energy.

4.3.3.2 Design Analyses

The design analyses for the fluid maintenance unit depend critically on the thermal stability characteristics of the heat-transfer fluid. While the fluid prequalification tests have provided useful information for the establishment of design parameters, it is still true that the existing data are limited and, in some areas, only qualitative. Information gleaned from heat-transfer fluid suppliers and from firms with experience in the fabrication of oil cleanup systems have also been useful. Pertinent background information on fluids is given in the following sections and a section on design trades follows.

Fluid Thermal Stability

Hydrocarbon heat transfer fluids, such as Caloria HT43, degrade with time via pyrolysis, which includes thermal cracking and polymerization, and (in the presence of air) oxidation. The oxidation of these fluids is so rapid at the TSU operating temperatures that it should be obvious that the maintenance system design must not permit contact of the fluid with air.

The dearth of information on long-term thermal stability of the fluids, alone and in contact with various solids, instigated the fluid prequalification tests described and analyzed in Section 6.2. Thermal stability tests conducted on Caloria HT43 and Therminol 66 were used to obtain rate equations for the fluid weight loss by volatilization of low molecular weight species produced by thermal cracking. These rate equations for Caloria HT43 and Therminol 66 were then used to determine the fluid-loss rate by volatilization for the Pilot Plant TSU operating over a specified temperature-time cycle (Figure 6-13) in which the top and bottom temperatures were 302°C (575°F) and 218°C (425°F). The calculated fluid loss rate over a 1-yr period (330 cycles) was 7.00% for Caloria HT43 and 6.07% for Therminol 66. Given the much higher cost of Therminol 66, it was concluded that the design of the thermal storage system should be predicated on using Caloria HT43. In view of the extensive AECL data on Therminol 66 decomposition (Ref. 4-15) which indicate a much lower rate of volatile loss than was found in this study, it is believed that the solids present in a number of the tests catalyzed the decomposition of Therminol 66.

There is not adequate information available on the rate of formation or solubility of polymeric species in Caloria HT43 as a function of molecular weight and temperature. The thermal stability tests (Section 6.2.2) provided some highly qualitative information on the formation of polymerized material from the results of gel permeation chromatography (GPC) analyses and measurements of the kinematic viscosity. The GPC analyses, performed by Sandia-Livermore (Ref 4-2), and kinematic viscosity measurements were conducted on samples of heat-transfer fluid removed from heated flasks at 1,000-hr intervals. Tentative results from the GPC analyses indicated that there was no significant amount of polymerization in the bulk oil. No indication was given of how much polymerization took place in either Caloria HT43 of Therminol 66. Measurements of the kinematic viscosity of Caloria HT43 at 1,000-hr intervals (Figure 6-3) gave no indication of fluid thickening that would signal an increasing concentration of polymeric species.

Commercial experience with lubricating oils shows that routine cleanup procedures will permit reuse of the oil almost indefinitely. Experience with rehabilitation and cleanup of lubricating oils and fuels used in large commercial and jet engines facilities is directly applicable to the cleanup of the thermal storage fluid. These processes use distillation to remove low boiling-point fractions in combination with filtration to remove solids. The lubrication-oil reclaiming industry recently developed several new processes which use solvent extraction to remove various additives and metals. Another new method adds recycled naphtha combined with a mild caustic (sodium carbonate) to the oil, causing the carbonaceous material to separate. After centrifugation, the oil is vacuum fraction distilled.

Design Trades

An analysis of design alternatives or trades in connection with the fluid maintenance unit must be based upon the best available information on (A) the degradation rate of the fluid, the fraction of the degradation products that are lost as volatiles, and the fraction of degradation products that will accumulate in the fluid as polymerized, high-boiling materials and residues, and (B) the maximum concentration of polymerized materials that can be tolerated in a practical system.

The actual degradation or pyrolysis rate of Caloria HT43 is unknown. The fluid tests, described in Section 6.2.2, measured only the rate at which highly volatile, low molecular weight species were lost from the system. The rate equation obtained for the percent weight loss of Caloria HT43 as a function of temperature is (with T in °K and R in weight percent/hr):

$$R = 5.3 \times 10^{10} \exp\left[-17650/T\right] \qquad \text{4-1}$$

Using Equation 4-1 and a specified temperature-time operating cycle for the Pilot Plant TSU (Figure 6-13), the steady-state fluid weight loss for one year (330 cycles) was 7.00%, which is pessimistic since no benefit was assumed from volatiles recondensing in the UMU.

Information from GPC analyses and kinematic viscosity measurements on fluid samples subjected to temperatures of 302°C (575°F) for 5,000 hr (which is equivalent to 2.02 yr of Pilot Plant operation at a top temperature of 302°C (575°F) using the TSU operating cycle given in Figure 6-13), indicated only that there was no significant amount of polymerization. The evidence implies that the removal of high-boilers from the Pilot Plant fluid would not be required for about one or two years.

In the absence of quantitative information on the rate of formation and accumulation of polymer residues, the rate of residue formation has been estimated to be about 5% or less of the total weight loss by volatilization calculated using Equation 4-1. For a 567,800-liter (150,000 gal) inventory of fluid in the Pilot Plant, Equation 4-1 predicts a loss of 120 l/day (31.7 gal/day) by volatilization, or a 6 l/day (1.58 gal/day) residue production rate. The waste stream removed from a fluid treatment system will be assumed to be 50% residue material. Thus, 12 l/day of 50% residue must be removed per day to maintain a steady-state value of high boiler in the fluid.

The amount of fluid to be processed daily would be determined by the rate of production and the tolerable steady-state concentration of polymerized materials. Little information is available that would permit the assignment of a limit for the steady-state concentration of polymerized materials. The supplier suggests that Caloria HT43 would be unsatisfactory for further use, and, hence, require refurbishing, if the precipitation number (ASTM Test D91-61) is greater than 0.5 and if its viscosity was greater than twice, or less than half, the viscosity of the fresh fluid. It is not known how well these tests can be related to tube fouling.

The lower the concentration of polymerized materials that can be tolerated by the system, the greater will be the amount of fluid that must be treated to maintain a steady-state condition. To be realistic, the maximum steady-state concentration of polymerized matter to be maintained in the bulk fluid by the maintenance unit was set at 10%. The 6 l/day polymer production rate, estimated for a 567,800-liter (150,000 gal) fluid inventory, would require that 60 l/day (15.9 gal/day) of fluid be processed. If the fluid maintenance system is operated for only 8 hr per day, 7.5 l/hr (2.0 gal/hr) must be processed to maintain a steady-state condition.

It would be entirely practical to simply remove 60 liters of fluid each day, thus filling a 15,000-liter (4,000 gal) tank truck every 215 days. This fluid could be reprocessed elsewhere and the reusable fraction returned. Since 60 l/day of clean fluid would have to be added to the system from storage, the amount of make up fluid on hand and the necessary storage volume would increase by 15,000 liters. Because of loading and unloading charges, transportation costs, and fluid processing costs, it will be cheaper to include a fluid-refurbishing process in the fluid maintenance unit. Moreover, since it is the objective of the Pilot Plant study to gain experience that will permit sound judgments to be made when the commercial-size system is built, it would be to the advantage of the program to include a heat-transfer fluid refurbishing process in the Pilot Plant system even if outside processing initially appeared to be cheaper.

As stated previously, the upper limit of 10% for polymer concentration, is believed to be realistic. Increasing the allowable concentration of polymeric material will decrease the size of the fluid processing unit required. The limitations on the upper value of polymer concentration may be affected by the following considerations, in addition to the primary requirement of freedom from heat-transfer tube fouling:

1. The viscosity increase at working temperatures requires greater pumping power.
2. If system shutdown and cooling are required, the room temperature viscosity should not be so great as to make a restart impossible.
3. High concentration may increase or decrease the degradation rate; the effect of high boiler concentration on decomposition is unknown.

Of the several possible methods available for application to the problem of removing high boilers (vacuum distillation, steam distillation, solvent extraction, and catalytic hydrocracking), only vacuum distillation processes have been considered here. Vacuum distillation is rather simple and straightforward; processes for oil cleanup that incorporate vacuum distillation are commercially available. The other three methods would involve some research and development expenditures to check them out for Caloria HT43. Steam distillation has the advantage of operating at or above atmospheric pressure, thus eliminating problems with possible air leaks, but would require large quantities of steam and could, in addition, present problems in water-fluid separation. Solvent extraction has been used to remove various additives, metals, and carbonaceous material from lubricating oils (usually followed by vacuum distillation) and to remove very high molecular weight compounds from Therminol 66 (Ref. 4-16). Catalytic hydrocracking would reclaim the polymeric material but would involve an extensive effort and a rather more complicated process.

Various vacuum distillation systems were considered, multistage strippers, thin-film evaporators, and mechanically aided thin film evaporators. Of these systems, it was decided that because of the probable high viscosity of the materials to be removed in concentrated form (perhaps as high as 50% by weight), the mechanically aided thin film evaporator will be used on the Pilot Plant. The evaporator-condenser can be rented on a monthly basis or purchased. Renting the unit has the advantage of allowing the user to determine whether it is indeed the type of device required for the task, if it has the proper evaporation capacity, and if it can be easily operated and controlled. If rented, the unit would still require three fluid pumps, a vacuum pump, and some minor instrumentation.

4.3.3.3 Design Description

The fluid maintenance unit for the Pilot Plant is shown schematically in Figure 4-26. It performs three functions to keep the fluid in good operating condition: filtration to remove suspended solids, distillation of a side stream to remove high boiling, polymeric compounds, and addition to fresh makeup fluid to replace the material removed by this unit and the ullage maintenance unit. The unit is designed to use existing commercial components.

Filters

Two 80 mesh filters are provided at the entrance to the pumps in the thermal storage main flow line. These filters will remove particles greater than 177 microns (0.0177 cm); this should include most of the rock dust, pipe scale and other foreign matter likely to be present in the system after system assembly. During the initial cycling of the system, most of these materials will be removed and the filters may have to be cleaned often. With each cycle the amount of material carried by the fluid will diminish since the TSU bed acts as an effective filtration medium, and only very

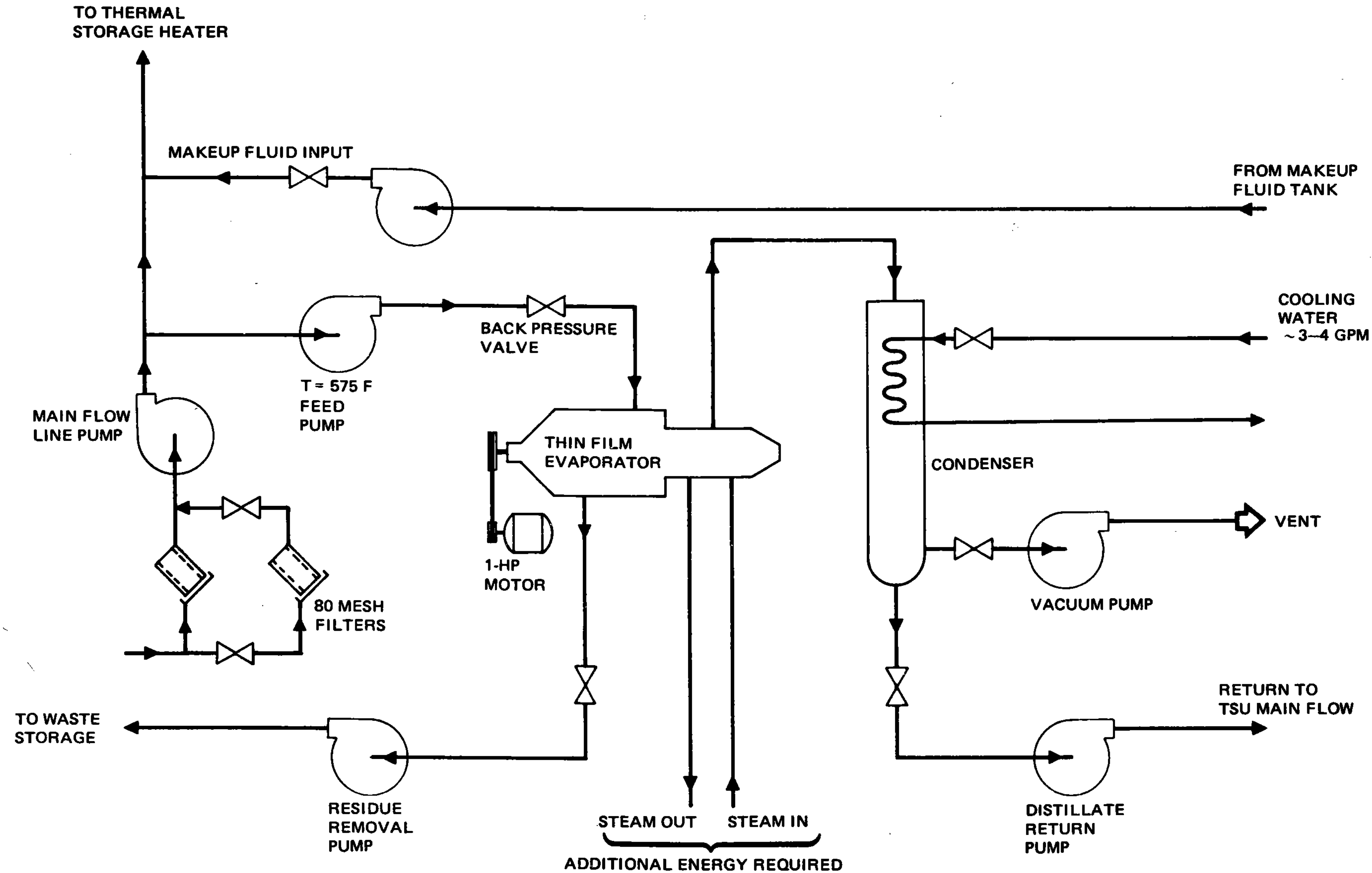


Figure 4-26. Fluid Maintenance Unit Design for 10-MWe Pilot Plant

small particles can be picked up by the slowly moving fluid (Ref: Para: 6.3.4.7). At typical fluid design velocities, particles of spherical shape and of the same density as rock will gradually settle back down to the bed if the diameter of the particle is greater than 16μ (0.0016 cm); only particles smaller than this could be carried along with the fluid. This particle size is so small, however, that as long as their concentration is low they will not harm the system.

The material collected by the filters will be examined to determine the extent and source of system contamination and will be of value in system design scaleup.

Low-density particulates of an organic nature may possibly appear in the fluid as a product of polymerization and carbonization as the fluid ages. Because of their low density these particles may not settle out by gravity in the TSU. This material, if it forms, will be removed in the vacuum distillation unit with other tar-like polymeric material.

Heat-Transfer Fluid Processing

The fluid processing unit will consist of a mechanically aided thin-film evaporator operated at a pressure of about 133 Pa (0.0193 psia) to 67 Pa (0.0096 psia). The unit will operate continuously over an 8-hr period. If polymerization is a greater problem than presently estimated, the unit can easily function continuously (24 hr/day).

The commercially available unit consists of a thin-film evaporator with a 0.74 KWe (1 hp) motor and a condenser, both constructed of 316 stainless steel, along with a variable-speed drive feed pump and a 255 l/min vacuum pump. The hot feed stream entering the vacuum evaporator is thrown by centrifugal force against a heated process wall to form a turbulent film. The vapors pass from the evaporator to a $0.7 m^2$ ($7.5 ft^2$) condenser where the distillate is collected and returned to the main circulating fluid loop by a small distillate return pump.

Any low boilers (volatile compounds) present in the incoming fluid are flashed immediately when the low pressure is first encountered as the fluid enters the vacuum distillation unit. Most of these highly volatile materials would pass on through the vacuum pump and normally would be exhausted to the atmosphere. If, during the operation of the Pilot Plant, the exhaust from the vacuum pump should become laden with low boilers, the operation of the ullage maintenance unit (which is the unit responsible for the removal of volatiles) will be checked and modified to return less of the liquid condensate from the ullage space to the heat-transfer fluid. If the venting continues, an activated carbon absorber can be attached to the vent line to remove the material.

Any heat required by the evaporator can be supplied by circulating steam or hot fluid through a heating jacket surrounding the evaporator.

The specifications for a unit which is about the maximum size anticipated for use in the 10-MWe Pilot Plant are given in Table 4-13.

Table 4-13

CHARACTERISTICS OF FLUID MAINTENANCE UNIT MECHANICALLY AIDED THIN FILM EVAPORATOR SYSTEM FOR 10-HWe PILOT PLANT

Fluid Feed Rate	7.5 l/hr (2.0 gal/hr)
Evaporator Motor	0.745 KWe (1 hp)
Fluid Pumping Power (3 Pumps)	1.30 KWe (1.75 hp)
Vacuum Pump	0.56 KWe (0.75 hp) 0.255 m^3/min (9 ft^3)
Cooling Water	15.1 l/min (4 gal/min)
Power Requirement for Evaporation and Heat Losses	3.39 KWe (8,600 Btu/hr)

Fluid Makeup

A 30-day supply of fresh Caloria HT43 will be kept on hand. Using Equation 4-1 and a temperature time cycle given in Figure 6-13, the weight-loss rate of Caloria HT43 was calculated to be 120 l/day and the polymerization rate to be 6 l/day. Assuming the polymer material is removed at a 50% concentration by the fluid maintenance unit (or 12 l/day), the fluid replacement rate is 132 l/day. The tank size for a 30-day supply is 4,050 liters (1070 gal). The fluid loss will be replaced once a day, or every several days, by pumping fresh fluid from the 4,050-liter storage tank into the TSU main line to the heat exchanger. The volume of makeup fluid actually required will be determined by recording the amount of low and high boilers lost.

Ordinarily, fresh heat-transfer fluid will contain a certain amount of dissolved oxygen from the air. When heated, this oxygen can cause some fluid degradation. However, fresh Caloria HT43 contains an antioxidant to react with dissolved oxygen. The antioxidant is destroyed at high temperatures and, thus, cannot afford protection to the fluid against air leakage once the fluid is heated. However, the tank pressure is maintained slightly above ambient pressure, thus eliminating the possibility of air leakage.

4.3.4 Desuperheater (DSH)

The DSH reduces the temperature of the receiver steam when it is being supplied to the thermal storage system at rated conditions. The effect of the desuperheater is to act as a buffer and to provide steam to the TSH at a temperature no higher than 343°C (650°F) to avoid shortening the life of the heat-transfer fluid.

4.3.4.1 Requirements

The basic purpose of the DSH is to lower the temperature of the steam directed to the thermal storage subsystem when it is incoming at rated conditions, 510°C (950°F). When steam is arriving at the thermal storage interface at 343°C (650°F) or below, the DSH is inactive. The requirements for the DSH are given in Table 4-14.

Table 4-14

PILOT PLANT DESUPERHEATER REQUIREMENTS

Superheated Steam (Inlet)	Pressure	10.1 MPa (1,465 psia)
	Temperature	510°C (950°F)
	Flow	16.5 Kg/sec (130,500 lb/hr)
Water (Inlet)	Pressure	14.5 MPa (2,100 psia)
	Temperature	211°C (412°F)
	Flow	3.35 Kg/sec (26,500 lb/hr)
Steam (Outlet)	Pressure	10.1 MPa (1,465 psia)
	Temperature	343°C (650°F)
	Flow	16.5 Kg/sec (130,500 lb/hr)
Turndown ratio		20.5
Maintenance: Minimum; must be accessible for replacing spray nozzles.		
Life: 30 yr		

4.3.4.2 Design Analysis

The desuperheater is basically a direct contact heat exchanger that provides cooling of the incoming superheated steam through the addition of and mixing with water. Since the product to be cooled and the coolant are identical, the most economical and basic type of unit simply sprays the water into the flowing steam. Analysis involves allowing for the heat and mass balance between the incoming water, incoming steam and leaving steam. The DSH length must provide an adequate mixing distance to ensure mixing and droplet evaporation before the water droplets hit the wall and thus condense on the pipe.

Water to the DSH is supplied from the tower feed water systems at 14.5 MPa (2,100 psia) and is throttled by the water control valve.

The DSH is primarily a mixing section of the main steam inlet line. The DSH will consist of an enlarged section of the main steam supply line with an atomizing probe injecting high-pressure feed water through three spray nozzles.

Water flow will be controlled to provide constant outlet steam temperature, 343°C (650°F) whenever the inlet temperature is above 343°C (650°F).

4.3.4.3 Design Description

The DSH is a commercial product and is made from a pipe section larger than the supply line to allow for adequate mixing and vaporization of the water droplets.

Figure 4-27 is a sketch of the DSH and lists basic characteristics. The DSH will be mounted in the main steam inlet line to the thermal storage heater and can be integrated into the system as a part of the steam feed line, immediately upstream of the thermal storage heater.

4.3.5 Thermal Storage Heater

4.3.5.1 Requirements

The TSH is the element of the thermal storage subsystem which transfers heat from incoming steam to the thermal storage heat transfer fluid. Table 4-15 lists the specific design requirements of the TSH for the Pilot Plant. Many preliminary designs and tradoff studies were made for the Pilot Plant TSH with earlier, slightly different requirements than those shown in Table 4-15.

The Pilot Plant TSH must demonstrate performance and efficiency under repeated daily thermal cycling over a wide range of fluid flowrates for 30 yr. The TSH is designed for an overall minimum cost of producing electricity. An optimum relationship exists between the initial capital cost of the TSH and the operational cost. This relationship depends on such factors as initial capital cost, amortization rate, power cost, and pumping efficiency of the heat-transfer fluid. The Pilot Plant TSH must operate with its own particular set of design conditions and also must be representative of the design for the Commercial Plant.

Reliability is an important criterion in the design of the TSH. It must be of high commercial quality. Metal surfaces must not overheat to prevent decomposition and degradation of the heat-transfer fluid. Deterioration of any part or parts of the TSH due to corrosion, erosion, or flow-induced vibration

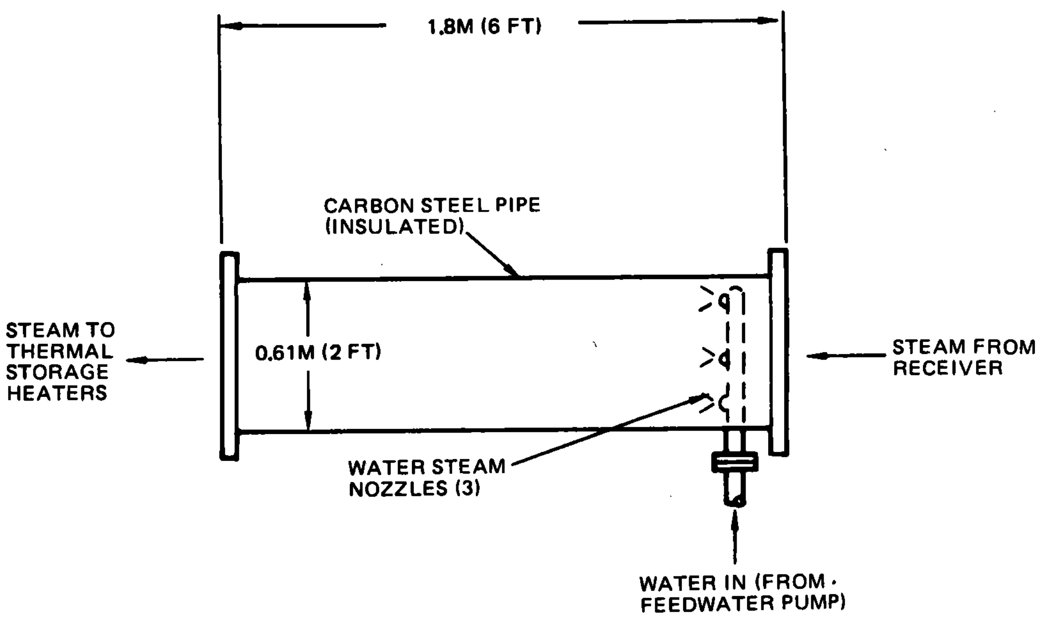


Figure 4-27. Desuperheater for 10-MWe Pilot Plant Thermal Storage Subsystem

Table 4-15

THERMAL STORAGE HEATER DESIGN REQUIREMENTS

No. of Exchangers Required	2
Following Information for Each of 2 Parallel Exchangers:	
Duty	15 MWt
Tube Side Conditions (Steam/Water):	
Inlet	343°C (650°F), 10.1 MPa 1,465 psia)
Outlet	246°C (475°F), 9.6 MPa (1,400 psia)
Minimum Flow	2,970 kg/hr (6,525 lb/hr)
Maximum Flow	29,597 kg/hr (65,250 lb/hr)
Shell Side Conditions (Caloria HT43):	
Inlet	219°C (425°F), 241 KPa (35 psig)
Outlet	304°C (580°F), 69 KPa (10 psig)
Minimum Flow	23,030 kg/hr (50,660 lb/hr)
Maximum Flow	230,230 kg/hr (506,500 lb/hr)

Service: The system operates on a daily cycle starting in the morning and ending in the evening. The duration of the daily operation varies between 8 and 10 hr per day.

Life Expectancy: 30 yr

Maintenance: Minimum maintenance or refurbishment.

Other: Must meet highway transportation limits.

must be avoided to ensure reliable operation of the system. Construction of the TSH from relatively standard commercially available materials is desirable.

Transportability within applicable federal and state regulations by highway and railroad carriers using standard transport vehicle and materials handling equipment must be considered in the design. Maximum highway limits are 18m (55 ft) in length and 33,000 kg (73,000 lb) in weight.

4.3.5.2 Design Analysis

Of the many type of heat exchangers available, a U-tube shell baffled heat exchanger was selected as the appropriate design. The U-tube is a compact heat exchanger which allows free expansion of the tubes and shell without special joint design, thus maximizing the life of the heat exchanger by minimizing the thermal stresses from the daily cycles. It is a conventional design and readily available from many heat exchanger manufacturers.

Design studies showed that it is more economical to use multiple exchangers, rather than just one. Having at least two exchangers in parallel gives greatly increased flexibility, controlability, and reliability during operations. Having more than one unit prevents complete shutdown of the subsystem during major repairs. Other advantages of multiple, smaller exchangers, rather than a single, large heat exchanger, are broader manufacturer availability, less difficulty in fabrication, and shorter leadtimes.

There are many variables involved when considering the design of a U-tube shell baffled heat exchanger (not all independent variables); some of the more prominent ones are:

Tube diameter
Tube length
Number of tubes
Shell diameter
Pitch/type (square, triangular)
Baffle spacing
Number of passes

Analysis at Rocketdyne sought to optimize these parameters with regard to total annual cost. A parametric study was performed on a countercurrent shell and tube heat exchanger. Design criteria assessed in the study were:

Material cost (tubes, headers, shells, etc.)
Performance (heat-transfer rates)
Operational cost (pumping and maintenance)

A computer aided parametric analysis was performed. Heat-transfer coefficients for both the oil and steam/water were computed automatically at several stations along the tube length. These coefficients were then used to perform energy balances on the heat exchanger to determine enthalpy change and wall temperatures. Fluid heat losses and gains were also computed. These calculations then fixed the fluid properties at the next station. This procedure continued in the program until the entire length of the heat exchanger was analyzed. The weight and cost of the heat exchanger were then estimated. Several runs were made with variations in the previously mentioned variables. From this anslysis a reasonable estimate of the final design was determined. The results are listed in Table 4-16. A more in-depth display of the parametric analysis is shown in the PDBR report (Ref. 4-2).

Table 4-16

RESULTS OF HEAT EXCHANGER OPTIMIZATION STUDIES*

No. of Exchangers	2
Length/Diameter Ratio	10
Tube OD	1.9 cm (0.75 in.)
Tube Pitch (square)	3.18 cm (1.25 in.)
Baffle Spacing	76 cm (30 in.)
Overall Coefficient	1,280 W/m^2-$^{\circ}$C (225 Btu/hr-ft, $^{\circ}$F)
Heat-Transfer Area (Based on Tube Mean Area)	377m^2 (4,100 ft^2)

* Guideline for evaluation purposes; final design in Table 4-18.

These numbers reflect the result of an economic optimization with respect to capital cost and fluid pumping power cost.

Further evaluation was carried out, however, considering structural life and reliability under operating conditions. These evaluations include the effect of fluid velocity on flow stability, tube vibration, and fatigue; the effect of temperature levels and temperature differences throughout the TSH on thermal expansion and thermal stresses; and the effect of any mixing of steam and fluid due to interstream leakage in either direction.

To answer these questions, services were obtained from an established heat transfer consultant firm (Reference 4-17) and, in addition, design specifications were obtained from heat exchanger manufacturers. The first and foremost criteria to consider is the ability of the designed heat exchanger to transfer heat as specified. Figure 4-28 shows a sketch of the enthalpy change versus steam and heat-transfer fluid temperature profiles as they go

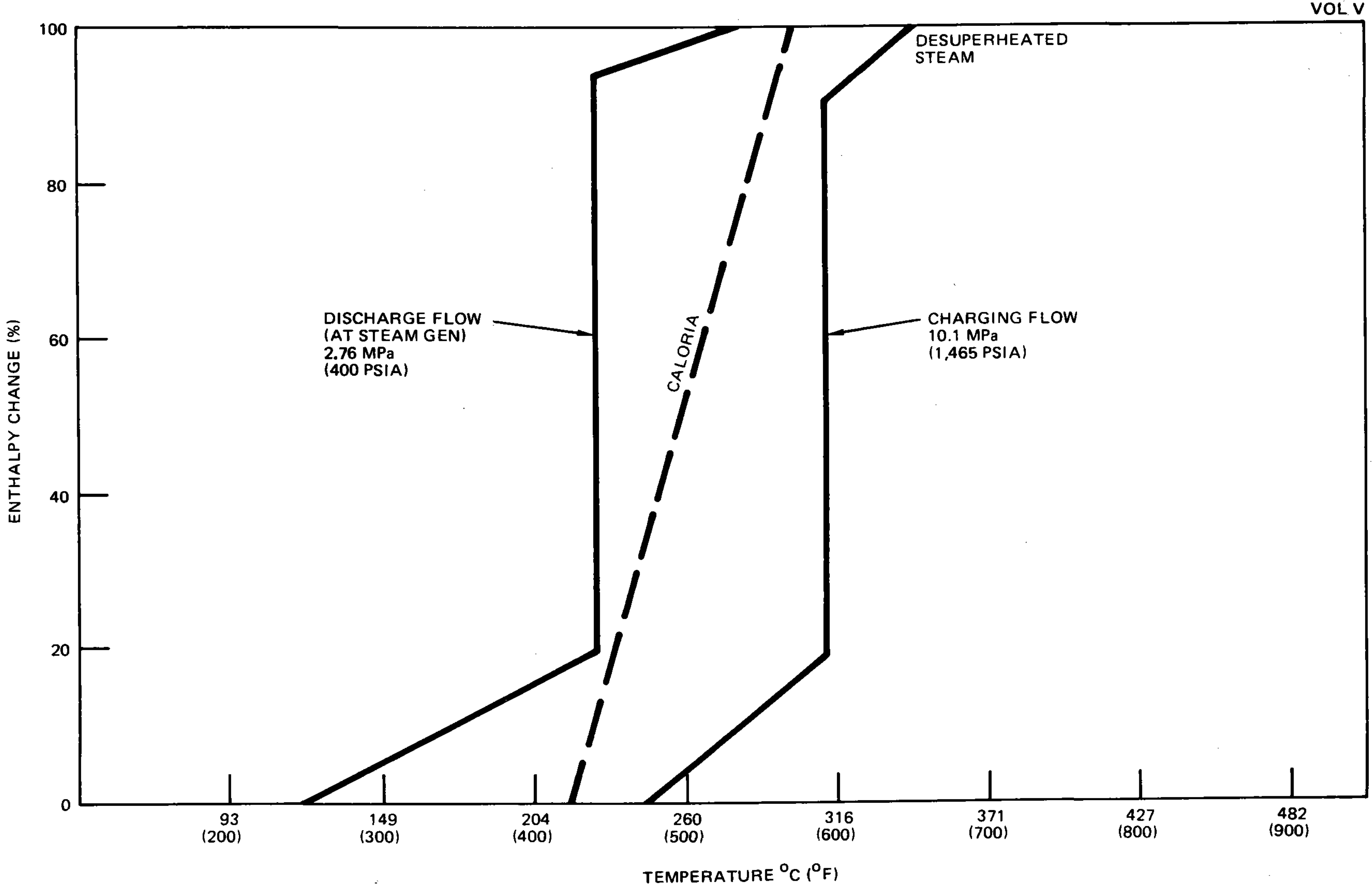


Figure 4-28. Thermal Storage Charging and Discharging Characteristics (Pilot Plant)

through the heat exchanger. The heat-transfer process on the steam/water side is divided into three regions: (1) desuperheating, (2) condensing, and (3) subcooling. Typical heat-transfer coefficients for each of these regions as well as coefficients for the fluid are listed in Table 4-17.

Table 4-17

TYPICAL HEAT TRANSFER COEFFICIENTS FOR VARIOUS CONDITIONS IN THERMAL STORAGE HEATER

Condition	Btu/hr-ft^2-$^\circ$F	W/m^2-$^\circ$C
Steam Desuperheating	5 to 20	30 to 110
Steam Condensing	1,000 to 3,000	5,700 to 17,000
Water Subcooling	50 to 3,000	280 to 17,000
Fluid Heating	10 to 300	60 to 1,700

In all cases, the fluid has the controlling resistance (lower coefficients). The heat exchanger was designed so that there would always exist an adequate margin between the fluid and steam temperatures at the pinch points (as illustrated by Figure 4-28).

Another aspect which was investigated was the possibility of interstream leakage. Any heat exchanger leaks will be normally from the high pressure steam to the fluid side. Due to the cyclic activity of the heat exchanger, however, there will be many instances where there will be a vacuum on the steam side as a result of condensation. The portion most susceptible to leakage is where there is a junction between the tubes and the tube sheet. It has been suggested that a double-tube sheet design be employed. The use of double-tube sheet designs are common in many shipboard condensors and coolers. The extra cost of the double-tube sheet is in the range of 5 to 15% depending on the size and number of tubes. If a leak occurs on either the steam or fluid tube sheet the fluid enters into a special ringed compartment around the space between the two tube sheets. The leak may be easily detected and the unit can be taken off in time. In addition, an Inconel overlay on the tube sheet can provide extra protection from both corrosion and erosion. The shell of the heat exchanger should be equipped with high-pressure relief valves in case of tube failure.

In most cases, U-tube heat exchangers will normally have baffles attached to the tube bundle in an effort to enhance the heat-transfer process by increasing fluid velocity through mixed cross and parallel flow paths. Close baffle spacing increases both the heat-transfer fluid film coefficient and the pressure drop, while it decreases the heat-transfer area. The velocity must be limited, however, due to vibration of the tubes occurring at high velocities. Vibration causes an unusually high stress on and increases erosion of the tubes. To increase the life of the heat exchanger, moderate velocities are generally recommended.

Carbon steel is a suitable material of construction which is compatible with both steam and Caloria HT43. The use of seamless tubes is normally prescribed at these conditions. The 30-yr service life may be somewhat stretching the capability of the material, but it is sufficiently inexpensive to make replacement bundles as a viable economic choice. There is a process now whereby fins can be rolled readily and economically onto the exterior of a tube. This increases the smooth tube area by a factor of three for the same tube length. The use of finned tubes has been recommended by some manufacturers, but the net effects on heat exchanger performance and size are not yet known.

The pitch of the tubes is another variable which was considered. Although triangular pitch is the most compact and reduces the shell diameter and provides better heat transfer, it permits no mechanical cleaning except for the exterior tubes. A square pitch arrangement allows free access to all the tubes but requires a somewhat larger shell. No coking of the fluid is anticipated, but if it should occur, mechanical cleaning would be a necessity. Therefore, a square pitch arrangement is the design choice. Drainage of both the water and fluid from the heat exchanger was considered. The tubes will be inclined for drainage when in a horizontal tube orientation. This really has no effect on heat exchanger performance as any accumulated water will be blown out by the steam. The only requirement for draining the fluid occurs when maintenance is required. Most heat exchangers have provisions for removal of the fluid as well.

The shell side of a U-tube heat exchanger must have two passes for there to be true countercurrent flow, which is the desirable condition to meet. This may be achieved in several possible ways, as depicted in Figure 4-29, 4-30, and 4-31. Figure 4-29 shows a potential configuration with a single heat exchanger with a longitudinal baffle affixed to the tube bundle. Since the bundle must be removable for maintenance and inspection, it cannot be attached permanently to the shell. Sealing of this baffle to the shell creates a new and difficult problem. Although initially sealing may not be a problem, sealing would become more difficult as time goes on due to bending, warping, etc., especially if the unit is long. The effects of leakage through this baffle become significant in reducing both the thermal efficiency of the heat exchanger and the heat-transfer coefficient (Heat Transfer Research Inc., Reference 4-18). Advantages of this configuration are that it is completely counterflow, which is the most efficient, and it is more compact, which reduces heat losses.

The second configuration, Figure 4-30 shows two single-pass U-tube heat exchangers in series. This can be thought of as being a single exchanger, rather than two heat exchangers. Although the first cost of this unit may be slightly higher, it has the advantage of relatively easy access of the tubes without any leakage of a longitudinal baffle. This method also has minimal thermal stresses both structurally and because of a lower mean temperature difference across each unit. The disadvantage is that there may be higher heat losses.

The third configuration investigated, Figure 4-31, has horizontal U-tubes with a longitudinal baffle which can be attached permanently to the shell. The flow of steam is baffled in the heat to reduce the mean temperature difference between the fluid and the steam. This concept is essentially the same as that in Figure 4-29, except that this exchanger is housed all in one unit. This design permits easy removal of the tube bundle with no sealing problem. A seal is still required at the tube sheath, but this presents no problem because of the relatively short distance involved. The pressure drops can be made higher in both this and the first type of heat exchanger because there is no worry of leakage. This increases the heat-transfer coefficient and makes the units more compact. The cost may be slightly higher for this third type than the second type of exchanger. Also, there will be more thermal stresses for this third type than for the second due to the higher difference in mean temperature across this single unit.

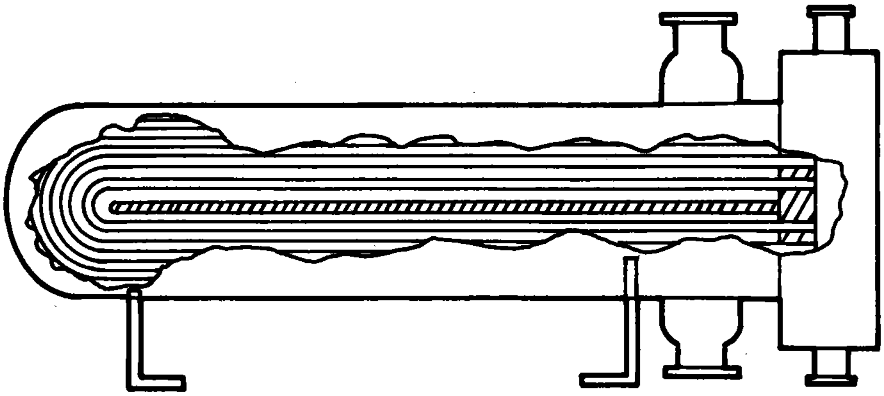


Figure 4-29. Typical Two-Shell Pass, U-Tube Heat Exchanger With Removable Tube Bundle

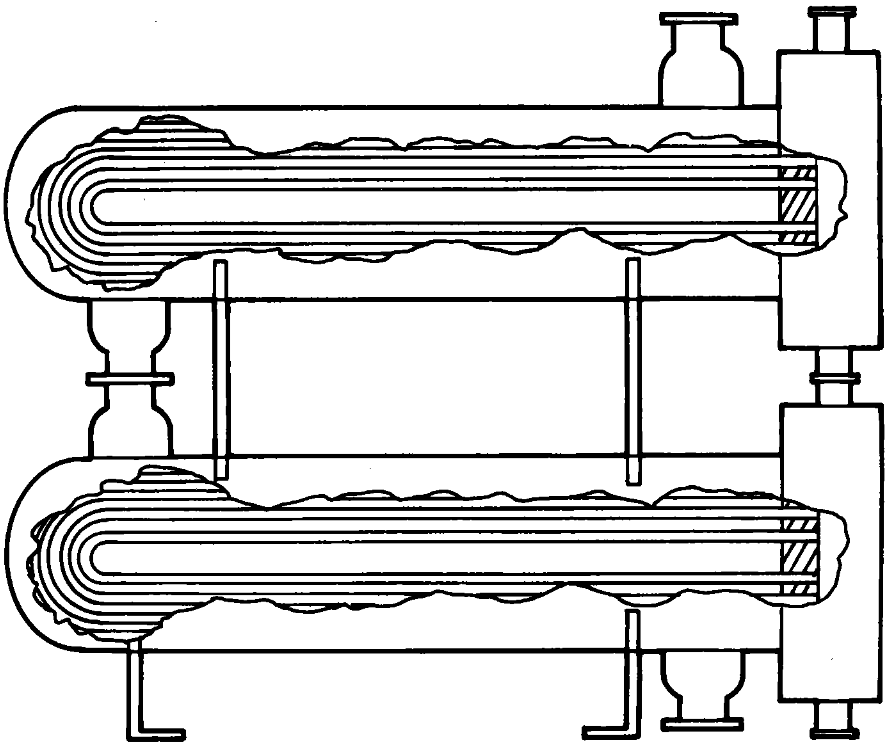


Figure 4-30. Two Single-Pass, U-Tube Heat Exchangers in Series, With Removable Tube Bundles

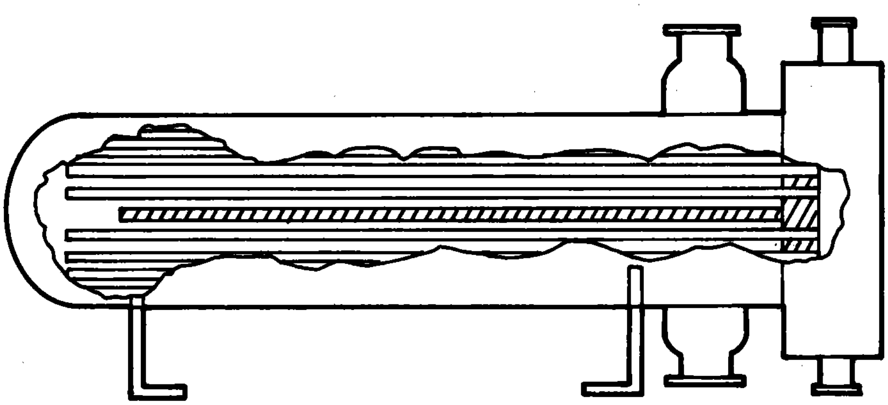


Figure 4-31. Typical Two-Shell Pass, U-Tube Heat exchanger With Tube Bends in Horizontal Plane, Permanently Attached Longitudinal Baffle, and Removable Tube Bundle

Table 4-18 summarizes key parameters of designs from four heat exchanger manufacturers who responded to RFQ's sent out by Rocketdyne to 20 companies. The preliminary design on which the bids were sought was a result of the analysis and optimization performed at Rocketdyne and discussed in the foregoing section. The maximum overall size of the heat exchanger — 12.8m (42 ft) length, 1.2m (4 ft) shell OD — the tube OD (1.9 cm, 0.75 in.) the pressure drop (172 KPa, 25 psi), and a TEMA classification of "R" were specified by Rocketdyne for a 15-MWt heat rate. The vendors were free to determine the remaining information to incorporate into their designs.

Table 4-18
SUMMARY OF THERMAL STORAGE HEATER DESIGNS FROM EXCHANGER MANUFACTURERS

Case	1	2	3	4
Shell Diameter (in.)*	31.2	32.4	36	42
Exchanger Length (ft)	42	42	31.5	30
Heat Transfer Area (ft^2)	12,141 (5,313)**	5375	5000	9024
Overall Heat Transfer Coefficient, Btu/hr-ft^2(°F)	68.5 (157)**	167.5	182.7	104.5
Weight (lb)	45,500	36,500	54,000	50,000
Pressure Drop (fluid side, psi)	20	10	25	—
Relative Cost	1.67	1.00	1.37	2.45

*Values are given for comparative purposes; therefore, are not translated to SI units

**Basic design has finned tubes; numbers in parenthesis are equivalent values based on smooth tube area

It should be noted that the overall heat-transfer coefficient, U, for heat exchanger Case 1, reflects an adjustment to account for finned tubes (the other designs use smooth tubes). The value of U for Case 1, based on the equivalent smooth tube surface area, gives a value (in parenthesis) which agrees well with Cases 2 and 3. The U value for Case 4 still deviates somewhat from that of the other three. This accounts for the area of Case 4 being larger than the others, which are in close agreement. It should be noted that heat exchanger companies usually provide extra tubes so that a major overhaul would not be required in the event of a tube failure. These areas listed here may include provisions for extra tubes. Also, some companies include the U-bend portion of the tube when computing area, while others do not.

The overall external dimensions of the TSH's varied somewhat, although the first two cases are almost identical. Optimum conditions usually tend towards high L/D's due to the expense involved in the fabrication of the tube sheet, especially when high pressures are involved.

Weights of the exchangers, with the exception of Case 2, are within a close range. The overall dimensions and area of Cases 1 and 2 are almost identical yet the weight of Case 2 is much lower. Items included in the weight analysis could vary. Other factors such as baffle spacing and the pitch of the tubes can influence the weight. For instance, the pressure drop of Case 2 is half that of Case 1, indicating a smaller number of baffles, which may account for the weight difference.

Cost is another important criterion. Again, there is a wide range with the last heat exchanger (Case 4) substantially higher than the others. The first three, however, are in a somewhat close proximity. The first exchanger (Case 1) is probably more expensive because of the finned tubes. Case 2, which has lowest weight, also happens to have the lowest cost. The cost of Exchanger 3 happens to fall between Cases 1 and 2, which seems reasonable from the standpoint of size, since it is 25% shorter than Case 1 and has a slightly larger shell. The cost also seems reasonable with regard to weight and area (with the absence of fins) when compared to Case 1.

The effectiveness of the finned tubing was also considered with regard to overall cost and performance. Estimates were made which indicate that Exchanger 1, which has the finned tubing, could be just as well off with smooth tubes when compared to Case 2 and 3. The fins appear to have increased the cost of the heat exchanger with no real benefit.

These design specifications submitted by manufacturers covered a considerable range of sizes, costs, weights, etc.; this is to be expected when bids are sought from a competing marketplace. As previously mentioned, each manufacturer has his own technique for determining heat-exchanger designs and cost, and they will vary. Considerable analysis was made, both before and after receiving the manufacturer's design, to evaluate the validity of these designs and costs.

The final heat exchanger design, described in Section 4.3.5.3, was developed from consideration of (1) Rocketdyne designs and design calculations made for this application during the past two years, (2) the designs submitted by exchanger manufacturers, and (3) calculations and design inputs of the heat-transfer consultant firm retained by Rocketdyne.

4.3.5.3 Design Description

The design for the Pilot Plant thermal storage heater is illustrated and summarized in Figure 4-32 and Table 4-19. Two of these exchangers in parallel will be required for the Pilot Plant. Each is a TEMA type "ΔFU", removable bundle, two-shell pass, U-tube heat exchanger. The thermal storage fluid, Exxon's Caloria HT43, is on the shell side; steam/water is in the tubes.

The U-tubes are in the horizontal plane, and there is a longitudinal baffle permanently attached to the shell. In addition, there is a double tube sheet and an Inconel overlay on the tube sheet to prevent tube sheet erosion if leakage should occur. There are six (unequal) tube passes, with a water level maintained in the subcooling region. A rotated square pitch will also be used to facilitate any required mechanical cleaning.

Conventional carbon steel will be used for all materials of construction. The heat exchangers will be thoroughly insulated.

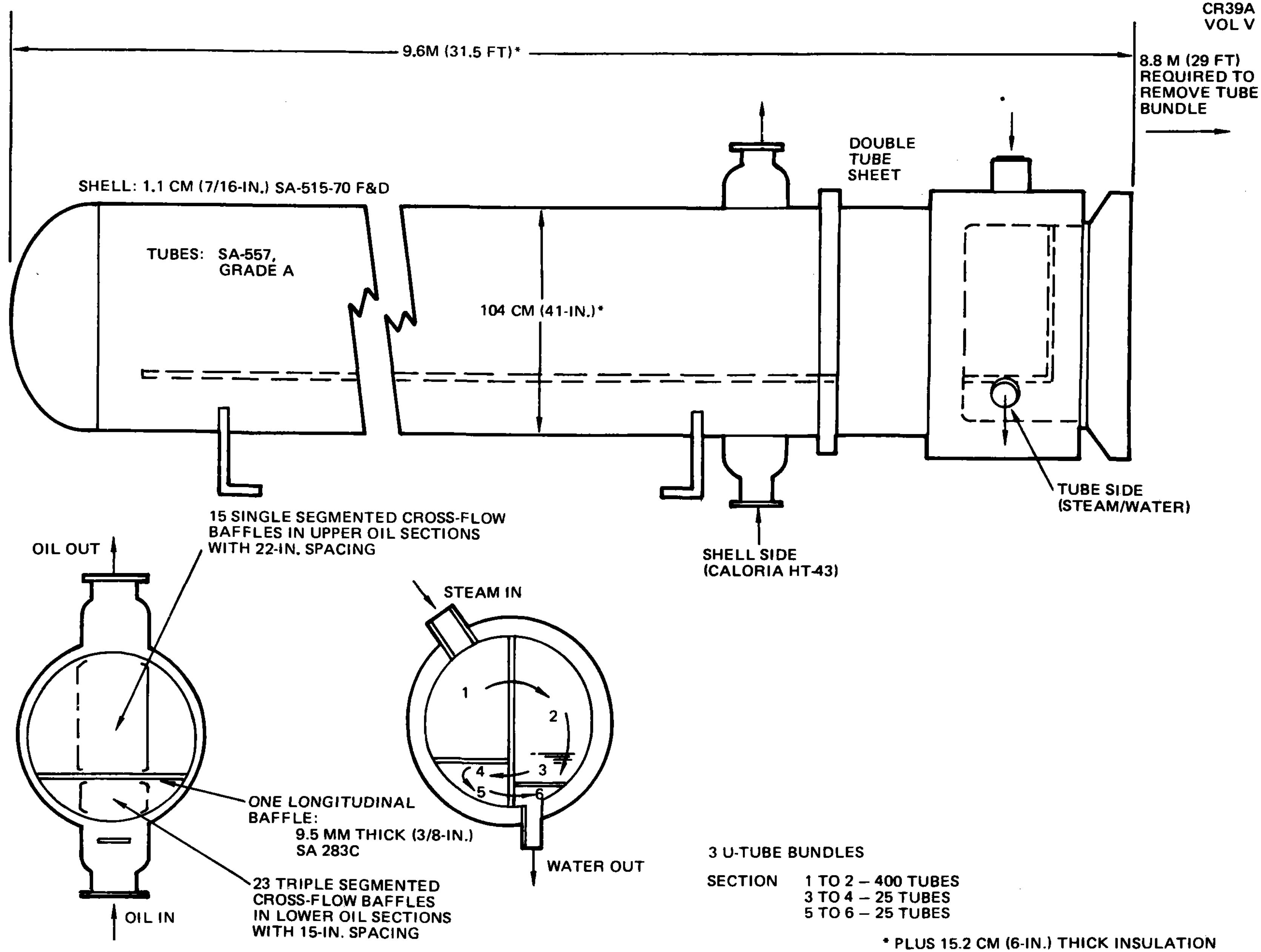


Figure 4-32. Thermal Storage Heater for 10-MWe Pilot Plant

Table 4-19

DESIGN DESCRIPTION OF THERMAL STORAGE HEATER FOR 10-MWe PILOT PLANT

No. of Units in Parallel	2
Capacity Per Unit	15 MWt
Flow Configuration	Caloria HT43 Shell Side, Steam Tube Side
Fluid Inlet/Outlet Temperature	218°/304°C (425°/580°F)
Fluid Flowrate/Unit	64 kg/s (141 lb/s)
Fluid Pressure Loss	0.17 MPa (25 psi)
Water Inlet/Exit Temperature	343°/246°C (650/475°F)
Water Flowrate/Unit	8.2 kg/s (18.1 lb/s)
Water Pressure Loss	0.1 MPa (15 psi)
Number of Tubes Per Unit	450
Tube I.D.	1.5 cm (0.6 in.)
Tube O.D.	1.9 cm (0.75 in.)
Tube Length (Average)	17.2m (56.6 ft)
Heat-Transfer Area Per Unit	464.5m^2 (5,000 ft^2)
Shell Diameter	104 cm (41 in.)
Exchanger Length	9.6m (31.5 ft)
Pitch Type	Rotated Square Pitch
Pitch/Diameter Ratio	1.25
Weight Per Unit	24,970 kg (55,000 lb)
Overall Heat-Transfer Coefficient Per Unit	1,033 W/m^2-°C
Longitudinal HT-43 Baffle	1
Triple Segmented HT-43 Cross-Flow Baffles in Lower Pass with 15-in. Spacing	23 (Exceeds Thermal Exchangers Mfg Assoc (TEMA) Standards)
Single Segmented HT-43 Cross-Flow Baffles in Upper Pass with 22-in. Spacing	15 (Exceeds Thermal Exchangers Mfg Assoc (TEMA) Standards)

4.3.6 Charging Loop Fluid Pumps and Piping

4.3.6.1 Requirements

The objective of this Pilot Plant design is to most effectively prove the 100-MWe Commercial Plant thermal storage concept. However, this imposed no special restrictions on the 10-MWe Pilot Plant design because pumps and piping systems are state-of-the-art. The flow schematics and fluid conditions are similar for the two plant sizes, with the pipe sizes and lengths and overall plant duty cycles differing.

The criteria used for the conceptual design of the Pilot Plant emphasized the following points:

- Capable of operating over a range of heat input
- Minimize parasitic energy consumption
- Lower possible capital cost
- High reliability

Obviously not all conditions can be satisfied and some trades were required. The plant duty cycle determines the operating range and indirectly affects the energy consumption. Multiple pumps and lower velocity are ways of minimizing parasitic energy consumption if the predominant operating range deviates from maximum conditions. This, however affects the capital cost adversely and the necessary trades must be made to achieve the optimum design.

All of the equipment, as well as the entire flow loop, must be reliable to maintain the integrity of the subsystem. Use of multiple units increases reliability and in some cases may decrease total operating cost as will be shown in the following sections.

4.3.6.2 Design Analysis

The first step in the analysis was to estimate the piping requirements of the 10-MWe Pilot Plan charging loop. These are shown schematically in Figure 4-33. It is essentially the same as that of the 100-MWe Commercial Plant with the valves, fittings, pumps, etc., in the same relative positions. For simplicity, it was assumed that there would be a corresponding thermal

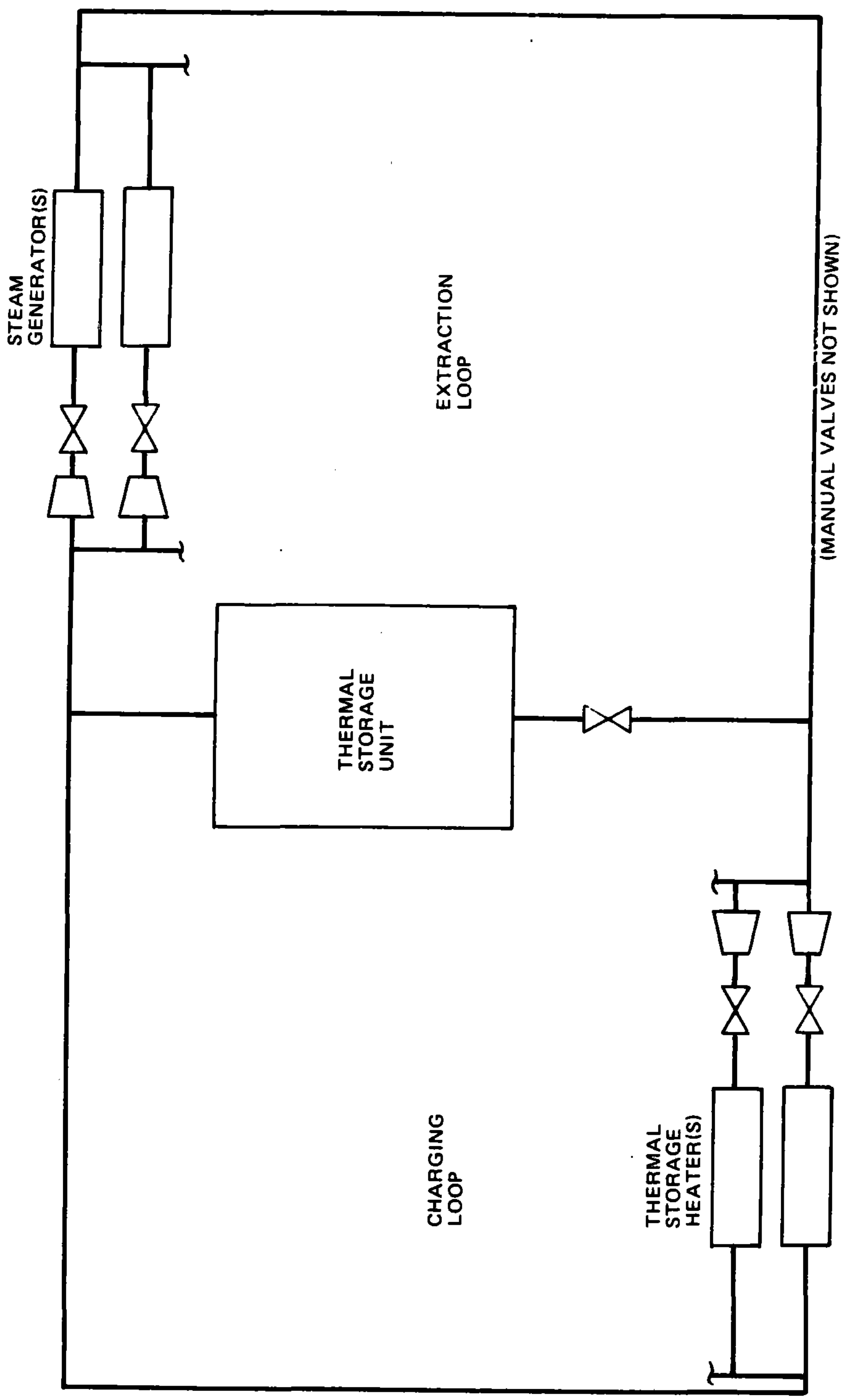


Figure 4-33. Diagram of Thermal Storage Subsystem Piping Components

storage heater for each pump, although not an absolute necessity. The quantities, material, and miscellaneous cost of these components are shown in Table 4-20. These correlations are based primarily on trends from Reference 4-19 (adjusted upward for inflation, based on data from Ref. 4-20) and on trends and absolute values from Ref. 4-21. For the valves, pumps, and motors, additional data from vendor quotations and catalogs were used in establishing the correlations. It is of interest to note that the installation costs of elbows, tees, and valves increase at a much faster rate if the diameters are greater than 24 in. This is because large components require special equipment and handling during installation.

In addition to the previously discussed piping component costs, the piping pressure drops were estimated as a function of fluid velocity from the well-known Darcy's formula. This determined the pump and motor horsepower requirements as well as the pump power and cost.

Before an accurate account of the power cost can be made, a duty cycle for the flow loop is required. The duty cycle is merely fluid flow as a function of time and is predicated on the basis of how much heat is available to storage. A typical daily cycle is illustrated in Figure 4-34. The annual charging duty cycle used for this analysis is given in Figure 4-35 and is in the form of total hours per year as a function of flow. As can be seen, the system operates over a wide range of flows which encourages the use of multiple pumps to curtail power consumption.

When the flow requirement is less than the particular pump output, the pump must be throttled to reduce the flow. Throttling reduces the flow but does not proportionately reduce power consumption because it forces the pump to operate inefficiently. The use of multiple pumps reduces the amount of throttling, but does not eliminate it. The power consumption versus flow can be determined readily for a particular pump from the corresponding flow-head curve; however, there exists no relationship to predict this for any size pump chosen at random. To circumvent this problem, turndown ratios versus percent maximum power were plotted for pumps over a multitude of ranges and characeristics. From these curves a correlation was developed

Table 4-20

PIPING COMPONENT SUMMARY FOR CAPITAL COST EVALUATION OF 10-MWe PILOT PLANT

Quantity*	Component		Material	Cost Correlation $/Unit = AX^B		
				X*	A	B
375 ft	Pipe		Carbon Steel	Dia (In.)	0.996	1.001
	Installation				2.44	0.772
	Insulation				3.38	0.857
	Painting				0.087	1.337
8	Valves			Dia (In.)	51.1	1.667
	Installation	< 24 In.			29.1	1.00
		> 24 In.			0.147	2.66
4	Tees			Dia (In.)	1.96	2.04
	Installation	< 24 In.			66.9	0.938
		> 24 In.			0.022	3.47
10	90° Elbows			Dia (In.)	0.568	2.42
	Installation	< 24 In.			20.1	1.00
		> 24 In.			0.147	2.66
1-12	Pumps					
	Installation			Hp	148.1	0.85
	Dual-Speed Motors				8.99	1.306

*English units only are given to fit standard USA piping conventions

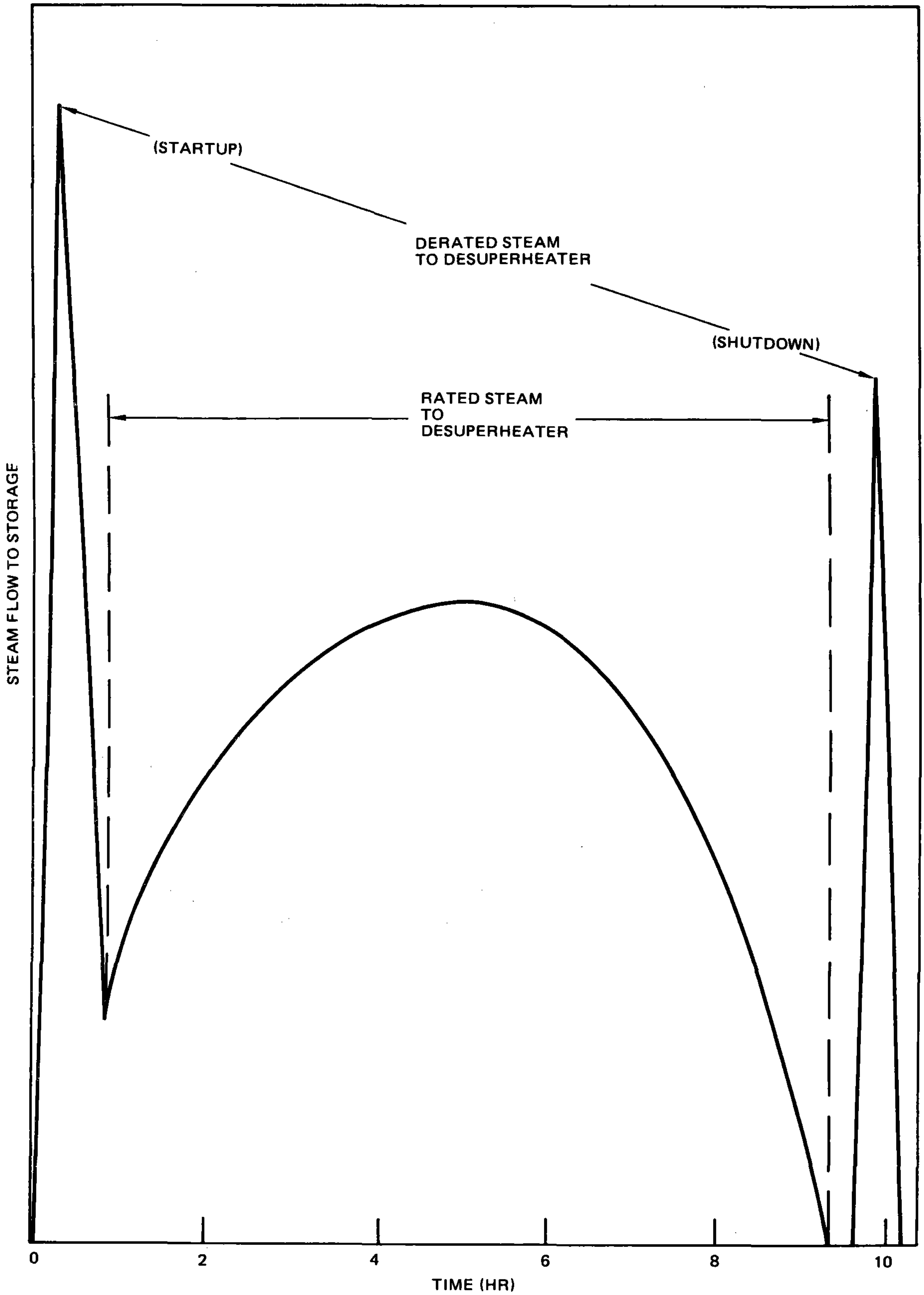


Figure 4-34. Typical Charging Duty Cycle at Equinox

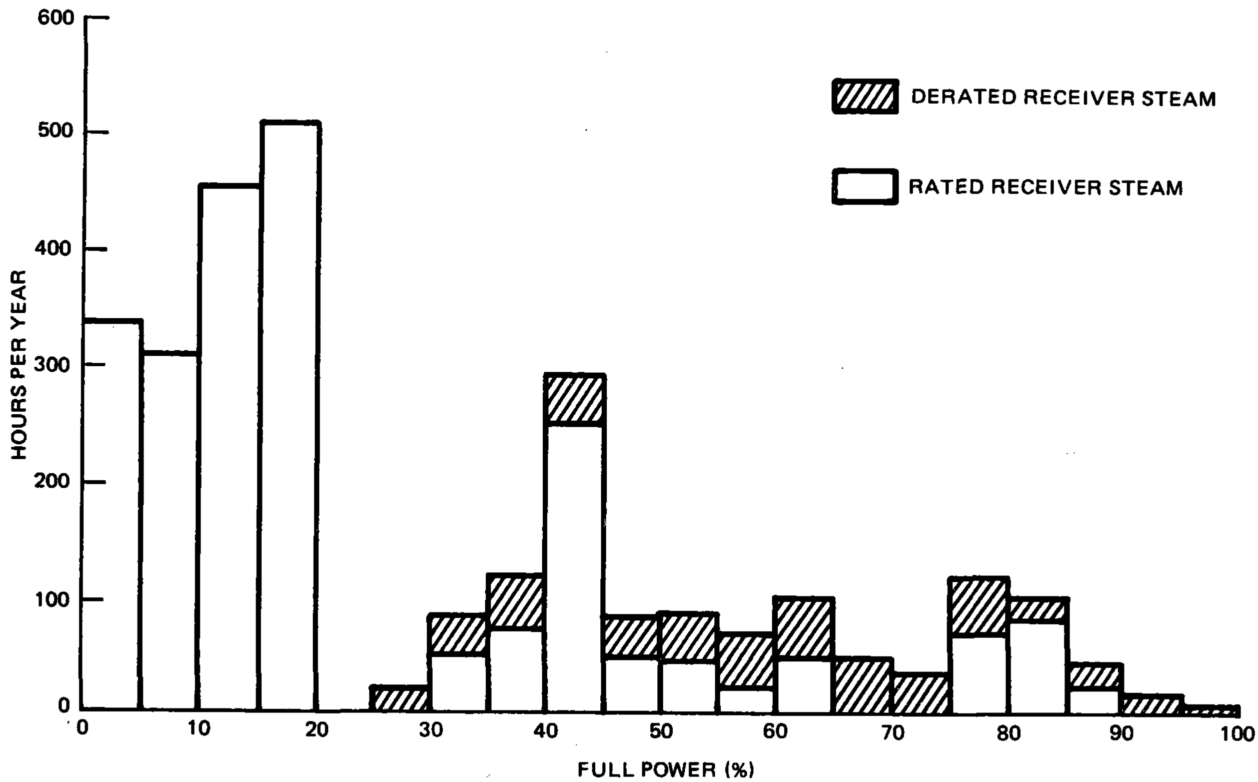


Figure 4-35. Pilot Plant Thermal Storage Charging Loop Annual Duty Cycle

to predict throttling power consumption for any pump. This correlation and the pump curves are shown in Figure 4-36.

These data and correlations were combined in the same cost-reduction computer program as used for the Commercial Plant fluid loop, but now using the Pilot Plant conditions. The final result of the computation, the total annual cost, is influenced by several parameters; namely, fluid velocity, number of pumps, and electricity cost. Fortunately, these parameters are independent so that a parametric analysis was easily performed by holding all but one of the variables constant. The program attempted to simulate the plant operation with regard to the cyclic flow requirements. Fluid conditions for each segment of the flow loop were specific. The maximum velocity was then chosen and the pipes were appropriately sized. A uniform pipe size was used throughout to facilitate the trade studies. The effect of this uniformity was a slightly lower velocity in the warmer sections, with little effect on the final results. The number of pumps was then chosen and cost of the pumps and associated valves, fittings, etc., was determined. The annual operation was then simulated by the appropriate number of pumps operating for a particular flow and the cost for this period determined for various electricity costs. The number of pumps was iterated several times, a new velocity was chosen, and the entire sequence was repeated. The annual operating costs were defined as the amortization of the initial capital cost and the annual power cost, using an amortization rate of 18%. The power costs were estimated by first determining the power required to run the pumps at the design point, i.e., maximum flow per pump at 75% and 98% pump and motor efficiencies, respectively, which was used in conjunction with the previously mentioned correlation (Figure 4-36) to compute actual power requirements. Secondly, the sum of the power cost was computed for each power level as time spent there.

The pumping costs are based primarily on the pressure drops (flow resistance) in the piping network. The sources of these are friction inside pipes, restrictions of flow as a result of fittings, valves, etc. and the heat exchanger(s). The components in the thermal charging loop and the respective loss coefficients are shown in Table 4-21.

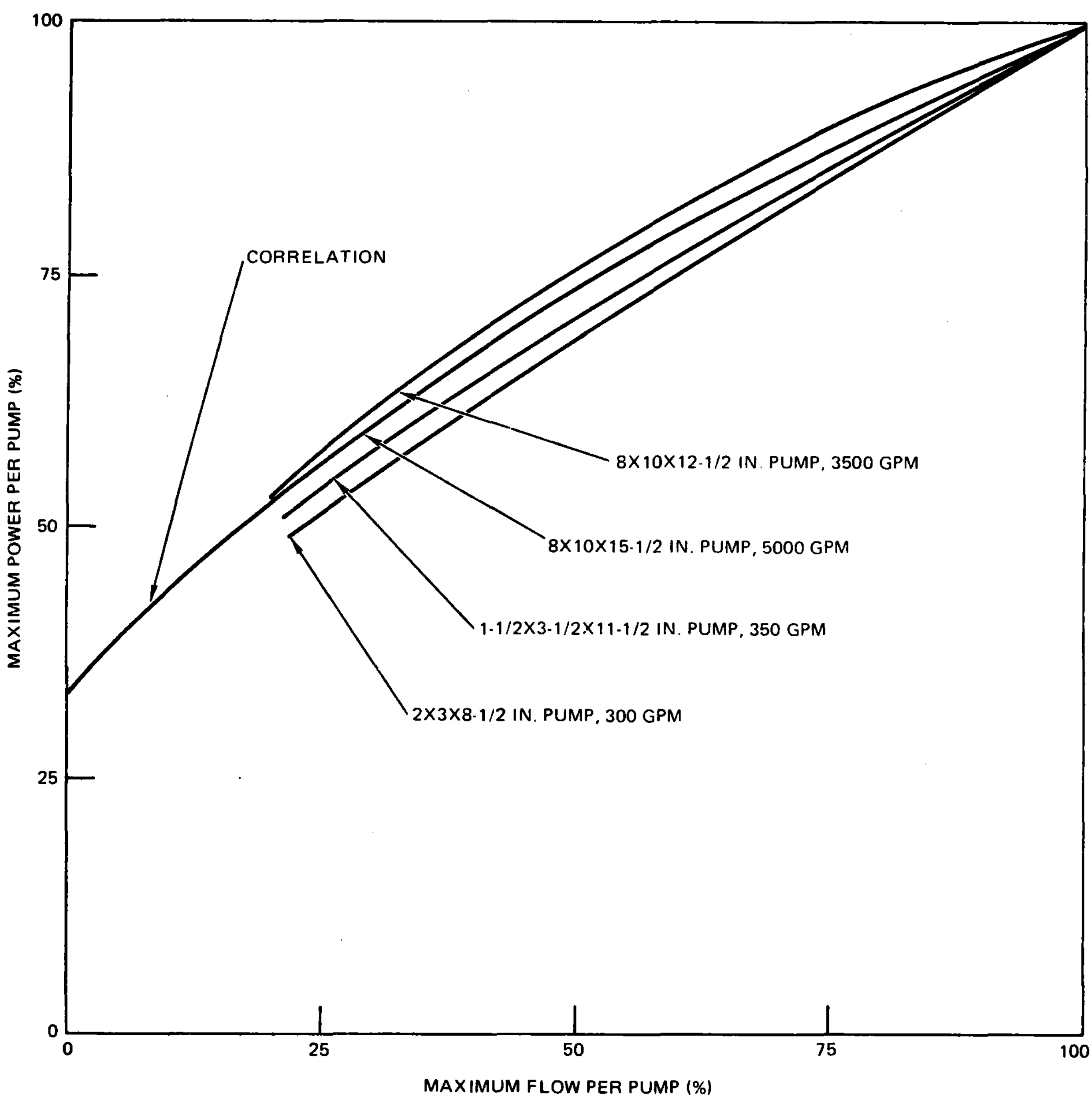


Figure 4-36. Effect of Reduced Flow (Turndown Ratio) on Power for Fluid Pumps

Table 4-21

PIPING COMPONENT DATA USED FOR PRESSURE DROP CALCULATIONS

Component	Loss Coefficient (Per Unit), Velocity Heads
Pipe	
Valve, Open	0.1
Throttle Valve	1.0
Elbow	0.5
Tee, Straight	0.3
Side Outlet	1.2

The most significant contribution, however, is due to the thermal storage heaters (TH-1 and TH-2) each of which requires a high pressure drop to provide effective heat transfer. A constant allowable pressure drop of 0.172 MPa, (25 psi), based on preliminary design estimates, was used in the analysis. Thermal storage heater costs were not included, as trades with heater costs are discussed in Section 4.3.5.

To better illustrate the impact of each variable the results are presented such that the number of pumps and fluid velocity are both shown as independent variables in Figure 4-37 and Figure 4-38, respectively. The series of curves represent the annual capital and total cost for electricity values of \$0.02 and \$0.04 per KWHe as functions of velocity and the number of pumps. The distance between the curves corresponds to the annual pumping cost. The curves in Figure 4-37 show that the capital cost drops substantially as the velocity is increased until the decrease in pumping cost is offset by the increase in pump cost, which occurs at approximately 6.0 m/s (20 fps). In addition to the increased pump size and associated fittings, more power is also required by higher velocities as evidenced by the divergence of the curves.

It is interesting to note that the curves become less divergent as the number of pumps increases, illustrating again the savings in parasitic power by the use of multiple pumps. The minimum annual cost occurs around a velocity

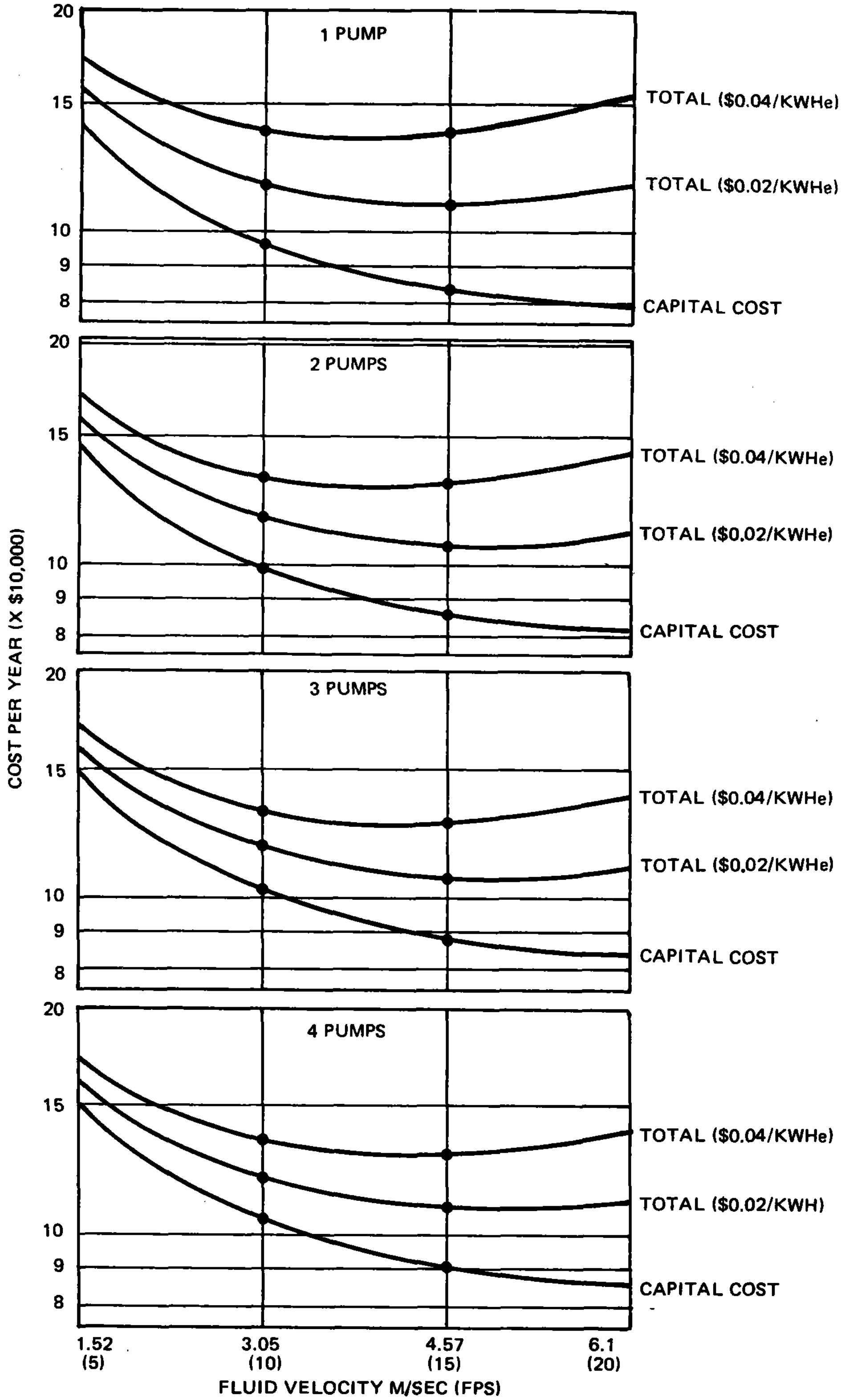


Figure 4-37. Annual Cost Optimization with Fluid Velocity, Number of Pumps, and Electricity Cost

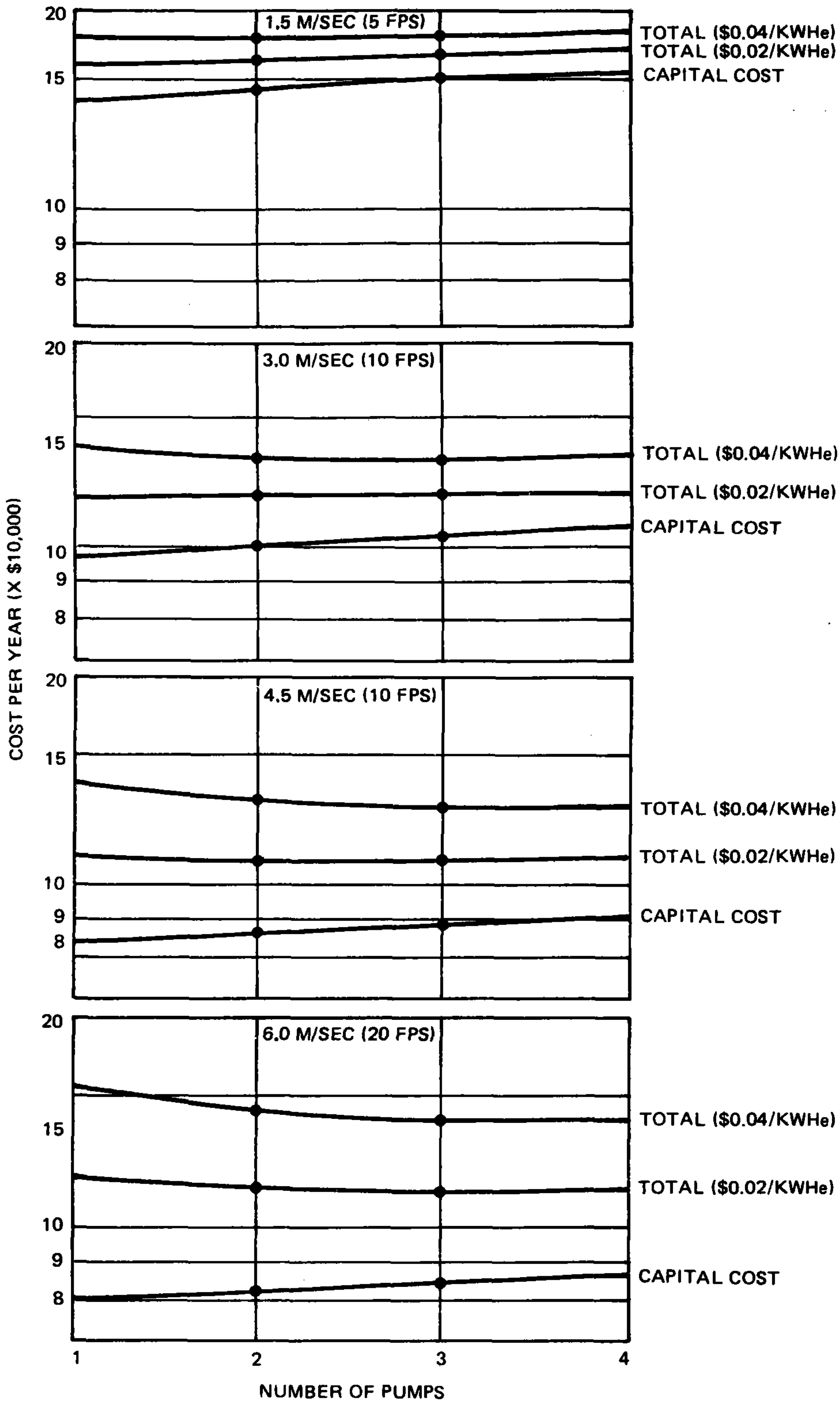


Figure 4-38. Annual Cost Optimization with Fluid Velocity, Number of Pumps, and Electricity Cost

of 4.5 m/s (15 fps) for an electricity value at $0.02/KWHe. Note how the minimum cost moves toward the higher velocities (and becomes less defined) as the number of pumps is increased. This effect is due to the pump predominately operating at less than maximum conditions so that the pumps are throttled less as the number is increased. As the electricity value increases, however, the minimum cost occurs at lower velocities, because the power cost became more significant.

The curves in Figure 4-38 show that the total annual cost is not significantly affected by the number of pumps (at an electricity value of $0.02/KWHe); in fact, at velocities less than 4.5 m/s (15 fps) the annual cost increases. The only drop in annual cost occurs at 6.0 m/s (20 fps) and only from one to two pumps. The power cost, however, does drop as the number of pumps increases, but this is, in most cases, offset by the increased capital required for the additional pumps and miscellaneous equipment. As the velocity and electricity values are increased, the number of pumps has a greater impact.

A first choice in this system design would be to choose the least cost pumping and piping scheme. Although the parasitic cost shown here is minimal, the value of electricity is certain to increase over the life of the plant whereas the capital cost will remain the same. The undefined minimum cost permits a choice from a wide range while still minimizing cost.

Due to the effects on operating cost and in an effort to minimize parasitic power consumption, a brief analysis on throttling methods was conducted. Several methods were analyzed: (1) a single pump, (2) multiple pumps, and (3) multiple pumps with variable-speed motors. The first two have already been investigated in terms of annual cost in the preceding analysis. The third method is a relatively new field with regard to pumping. The different types of variable-speed pumps available are:

1. Two-speed motor.
2. Eddy current clutch or hydraulic coupling.
3. Belt drive (limited to 60 hp).

Due to size limitations and inefficiencies, all but the first have been eliminated from this analysis.

To better illustrate the influence of each method, a plot of the pump power and system power requirements as functions of flow was developed, Figure 4-39. The solid curve to the right of the plot represents the theoretical power required to pump the fluid through the piping system. The more closely the pump power follows this curve, the more efficiently it operates. The distance between this curve and the pump curves represent the power being dissipated across the control valve. The single pump is clearly the worst case. Two pumps improve it somewhat and additional pumps would improve it in both the upper and lower flow regimes. With the appropriate number of pumps, the system load could be closely approximated but the capital cost would be high.

The most efficient method, however, is the two pumps with two-speed motors (1,750/1,150 rpm), as shown by the dotted line. Reduction of the speed also reduces the pressure rise of the pump while still maintaining the same flow, thus the excess pressure can be eliminated when not required.

The actual savings in power cost are still dictated by the duty cycle. No benefit from the dual speed pump can be realized if flows are greater than 66%. The duty cycle, Figure 4-35, shows that a considerable portion of the flow is below the 50% flow level, so that a dual speed pump merits investigation.

4.3.6.3 Design Description

On the basis of the preceding analysis and the criteria listed in Section 4.3.6.1, an optimum design was developed. A fluid velocity of 3.9 m/s (13 fps) was chosen, resulting in a pump diameter of 25.4 cm (10 in.). Two identical pumps in parallel with a total maximum capacity of 0.18 m^3/s, (2,950 gpm) were picked on the basis of both power consumption and reliability. The pump drivers are dual-speed (1,750/1,150 rpm) squirrel-cage induction motors. Since motor speed is inversely proportional to the number of poles in the stator windings, two speeds can easily be obtained by wiring the motor appropriately. The motor is a 480V 3-phase, two-winding type. A summary of the final design parameters and associated equipment with the corresponding size, model number, etc., is given in Table 4-22. The curves corresponding to the pumps selected are shown in Figure 4-40.

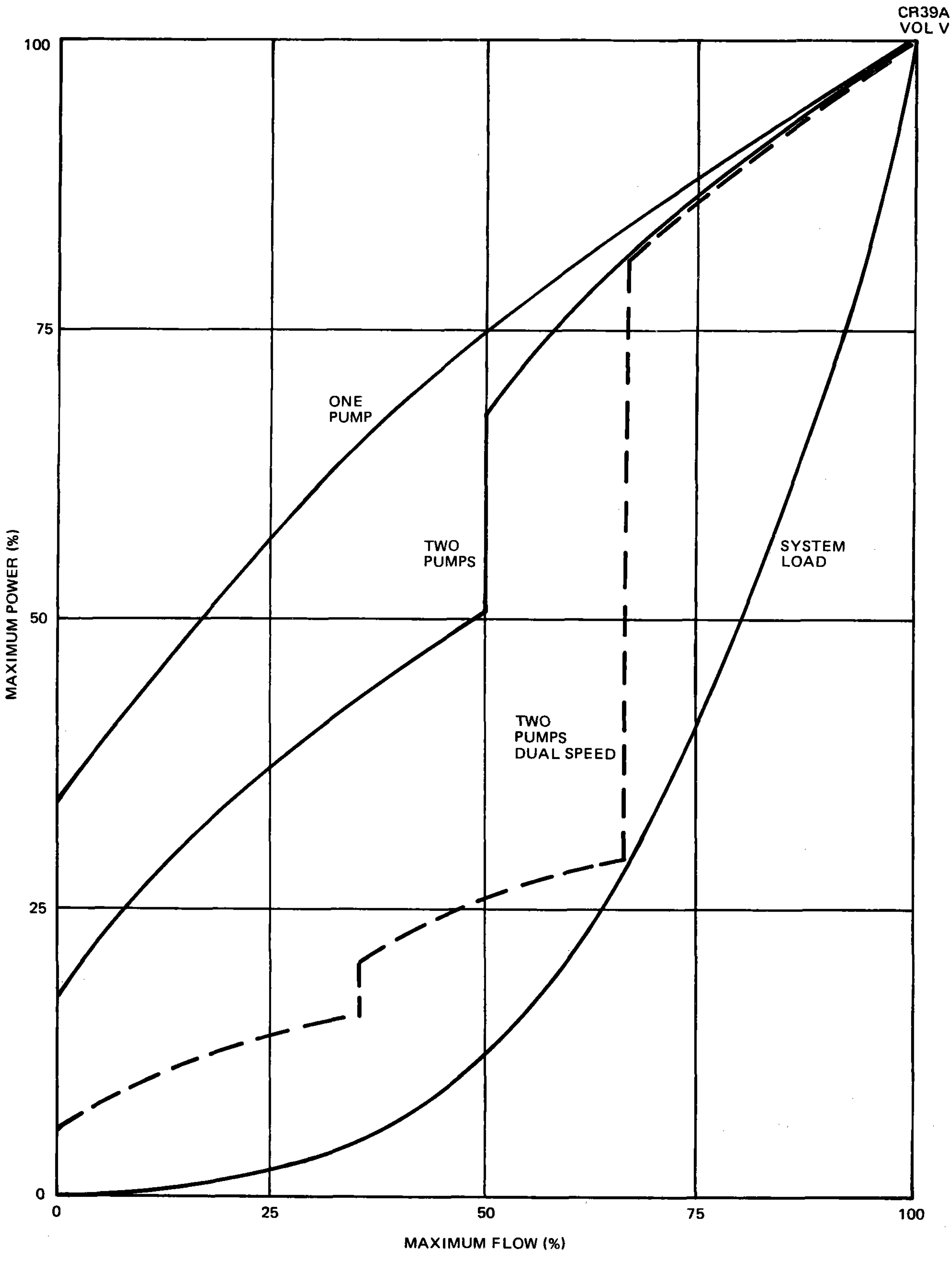


Figure 4-39. Effects of Turndown Ratio on Theoretical Power Required in Fluid Charging Loop

Table 4-22

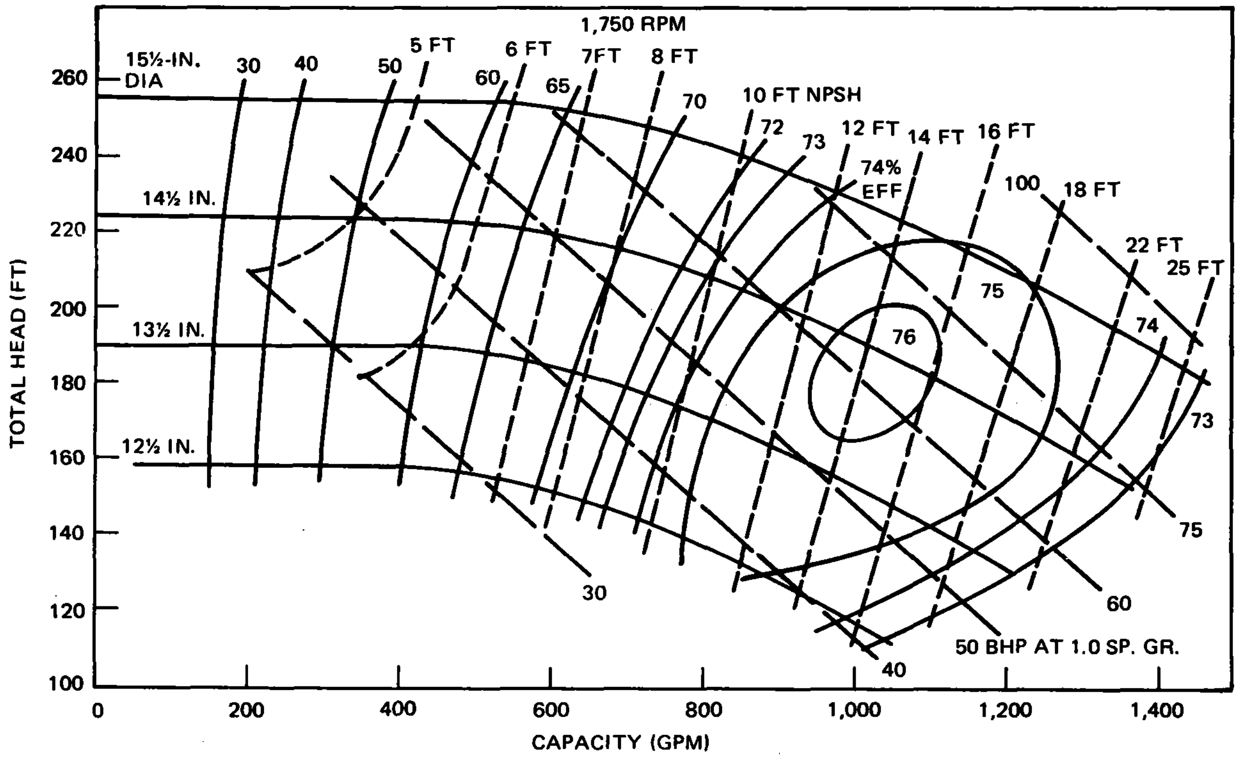
SUMMARY OF PILOT PLANT CHARGING LOOP FINAL DESIGN EQUIPMENT

Item	Quantity	Description
Pumps	2	Dean Brothers Model R454-4 in x 6 in. x 15-1/2
Motors	2	Dual Speed, Variable Torque, 80*** hp maximum, Two Windings, 480V, 3-Phase
Pipe	61m (200 ft) 21m (70 ft)	10 in. Schedule 40** 8 in. Schedule 40*
Valves		
Shutoff	5 6	8 in. 10 in.
Throttle	2	8 in.
3-Way	1	10 in.
Tees	2 4	8 in. 10 in.
90° Elbows	5 4	8 in. 10 in.

**US Standard pipe sizes given (no SI equivalent pipe sizes)
*8 in. Sch. 40 pipe was used for all segmented flow
***Required motor input power

4.3.7 Steam Generator

The thermal storage steam generator for the Pilot Plant is intended to provide all of the operational capabilities which would be necessary to permit a complete technical verification of the overall subsystem. These capabilities include providing admission steam for turbine startup and operation as well as providing a source of steam for blanket heating of heat exchangers and turbine seals during nighttime shutdown periods. As discussed in Section 3.2.6, the concept of exact verification of the ultimate commercial steam generator design has been compromised in favor of a more conservative approach to maximize steam generation reliability.



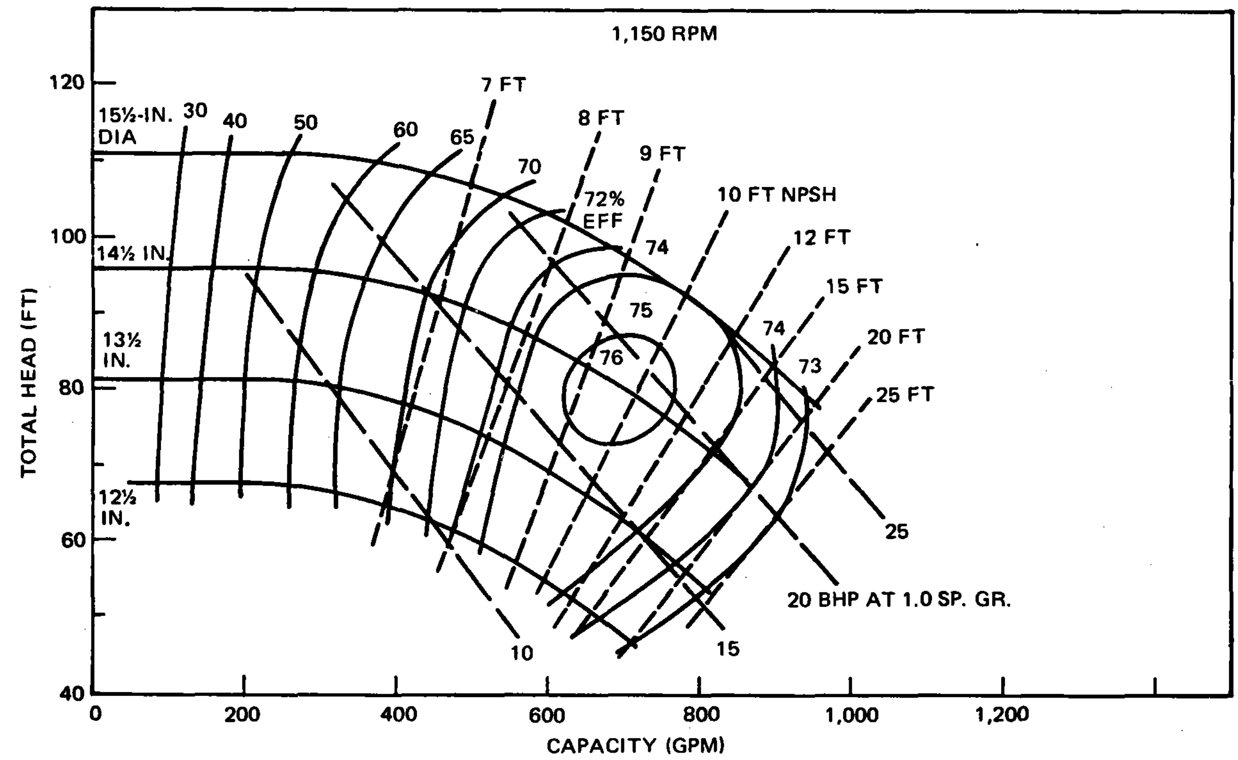


Figure 4-40. Thermal Storage Charging Loop Pump Characteristic Curves

As a result, the kettle boiler approach has been selected as the baseline for the commercial system (Section 3.2.6) instead of the lower cost single-pass-to-superheat design so that overall control complexity is minimized. The economic penalty involved in making this selection for the Pilot Plant is estimated to be $30,000 to $40,000 (excluding control hardware) which may be recovered in ease of operation.

4.3.7.1 Requirements

The principal requirements which must be satisfied by the Pilot Plant steam generator are listed in Table 4-23. The discharge and water/steam flow rate data shown represent the anticipated range of operation where control at the specified outlet temperature and pressure conditions must be maintained. The maximum values correspond to the requirement to produce 7-MWe net power from the turbine with thermal storage steam alone. The minimum values reflect the threshold of controlled flow when the turbine is being operated simultaneously from both the receiver and thermal storage steam sources or the minimum flow at which control would have to be maintained during turbine startup.

The feedwater inlet and steam outlet conditions (pressures and temperatures) are more or less limited by the envelope defined by the turbine admission steam requirement and the pinch point limitations between the Caloria and the water/steam which occurs in the steam generator equipment. This situation is illustrated in the thermal storage charging/discharging diagram shown in Figure 4-41. In contrast to the commercial system where the charging side pinch point was the most critical, the data shown for the Pilot Plant indicates that the discharge pinch point is by far the most critical point influencing the design. The difference is due to the fact that the Caloria is operated at a 14°C (25°F) lower average temperature for the Pilot Plant than for the commercial system for reasons of design conservatism. In addition, the Pilot Plant turbine is designed to operate with less superheat than the corresponding commercial turbine. This translates into a higher turbine admission steam pressure requirement for the Pilot Plant and imposes a severe restriction on the steam pressure drop permitted in the superheater.

Table 4-23

PILOT PLANT STEAM GENERATOR REQUIREMENTS

Discharge Rate	
Maximum	32.1 MWt (109.6 x 10^6 Btu/Hr)
Minimum	3.1 MWt (10.6 x 10^6 Btu/Hr)
Feedwater/Steam Flow Rate	
Maximum	13.21 Kg/Sec (104,700 Lb/Hr)
Minimum	1.27 Kg/Sec (10,100 Lb/Hr)
Feedwater Inlet Conditions	
Temperature	121°C (250°F)
Pressure	2.90 MN/m^2 (420 psia)
Outlet Steam Conditions	
Temperature	277°C (530°F)
Pressure	2.76 MN/m^2 (400 psia)
Hot Standby Demand	
(Maintain Preheated Equipment)	0.02 MWt (68,260 Btu/Hr)
Nighttime Seal Steam	
219°-149°C (425-300°F) Caloria Temp	0.326 MWt (1.11 x 10^6 Btu/Hr) for 16 hr with steam at 135°C (275°F)

As in the case of the commercial steam generation equipment, the Pilot Plant version will be designed to be maintained in a hot standby condition during periods of turbine operation in the event rapid demands are imposed. In addition, the steam generator will be designed to provide turbine seal and heat exchanger blanket steam during turbine shutdown periods in the quantity indicated in Table 4-23, using low-temperature Caloria.

4.3.7.2 Design Analyses

Since the overall steam generating concept involving a kettle boiler with separate preheater and superheater elements has been selected as the preferred approach, the specific design analyses focus on defining the detailed design information for the Pilot Plant equipment. Design parameters involved in

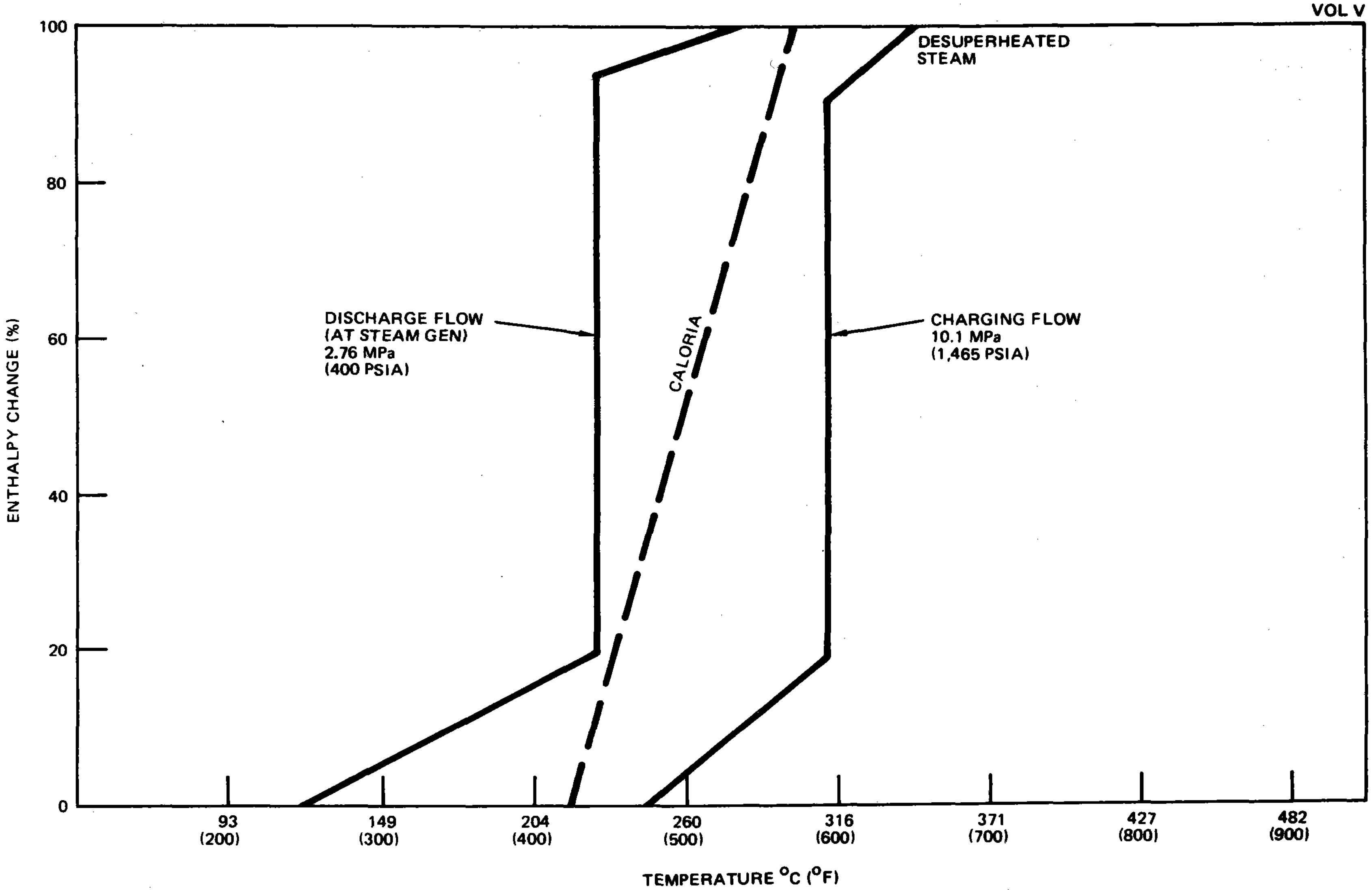


Figure 4-41. Thermal Storage Charging and Discharging Characteristics (Pilot Plant)

these analyses include number of steam generator trains, number of tubes, tube size and diameter, shell size, and flow control equipment. As in the case with the commercial steam generator design, an effort was made to restrict parameters to those values commonly used in the heat exchanger manufacturing industry.

Number of Steam Generator Trains

The number of steam generator trains appropriate for the Pilot Plant was determined based on operational, economic, and reliability considerations. From an operational standpoint, the steam throttling range requirement has a direct impact on the Reynolds number which must be maintained for the tube side fluid (Caloria). If a single unit were employed, the design point Reynolds numbers would have to be in excess of 30,000 to maintain some flow turbulence at the minimum power (~10%) level. This fairly high Reynolds number has a direct impact on the Caloria pressure drop and resulting pumping power. On the other hand, the use of parallel trains reduces the required throttling range of any one of the trains by the inverse of the total number of trains available. Thus, the design point Reynolds number can also be reduced by an equivalent factor, with a significant reduction in pressure drop across the heat-exchanger elements. An estimate of equivalent annual cost, including both hardware and pumping power costs, for one, two, and three steam generator trains is shown in Figure 4-42. It is seen from this data that three parallel trains are prohibitively expensive while a single train is only slightly superior to two parallel units, particularly when considering the combined effects of hardware and pumping power costs. When reliability factors are introduced into the evaluation, the redundancy aspects available when two parallel half capacity trains are involved offers an obvious superiority over a single full-capacity train. Bearing in mind also the experimental nature of the Pilot Plant where component flexibility is desirable, the two parallel train configuration was selected for the Pilot Plant.

Expanded Steam Generator Analysis

To precisely define the steam generator equipment desired, it is necessary to expand the overall performance factors into an optimized design. The basic performance data, along with a schematic of one of the two parallel

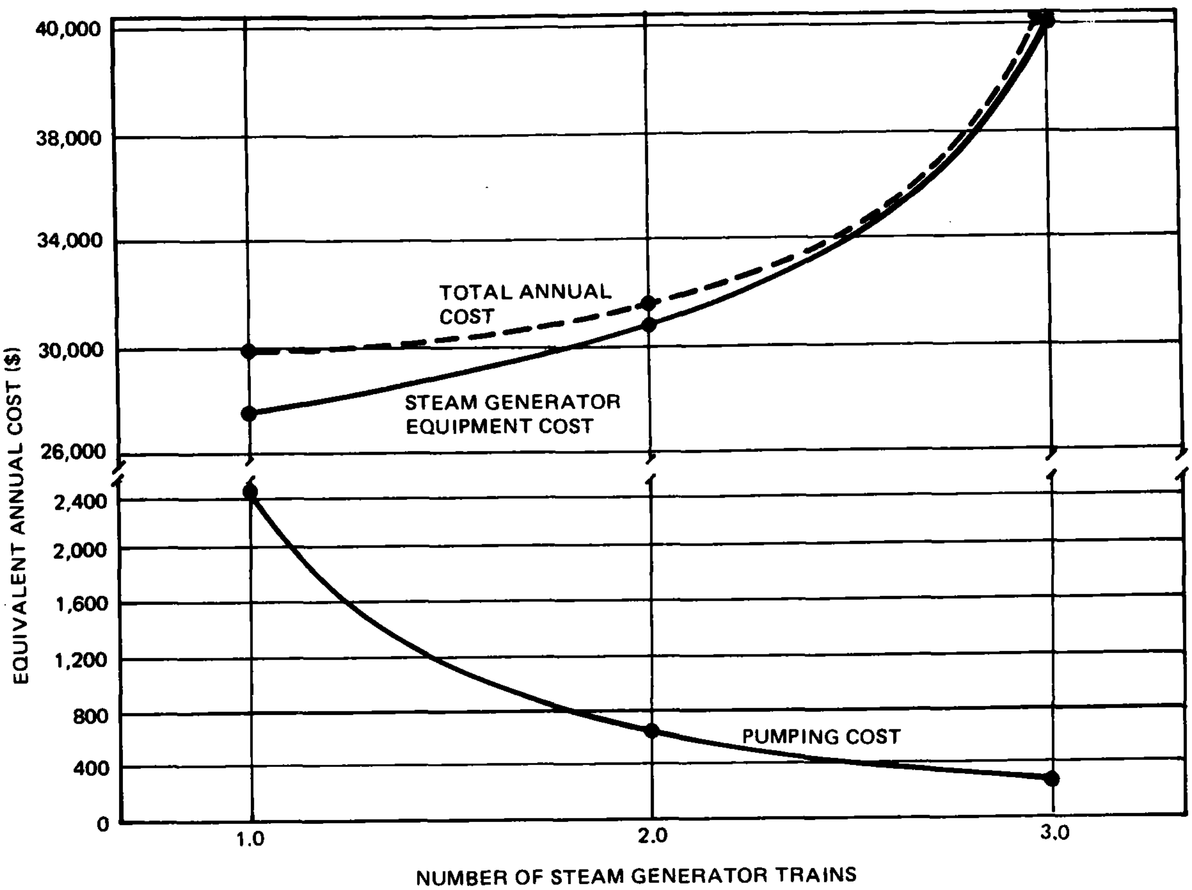


Figure 4-42. Impact of Number of Steam Generator Trains on Estimated Annual Cost

trains, is shown in Figure 4-43. The specific heat exchanger design for each of the three stages must satisfy the relationship.

$$(UA) = \frac{1}{\left(\frac{1}{h_c A_c}\right) + \left(\frac{R_{foul_c}}{A_c}\right) + \left[\frac{R_{wall}}{\left(\frac{A_c + A_f}{2}\right)}\right]\left(\frac{R_{foul_f}}{A_f}\right) + \left(\frac{1}{h_f A_f}\right)}$$

where

h = metal to fluid heat transfer coefficient

A = surface area

R = thermal resistance

and the subscripts are defined as

c = Caloria side

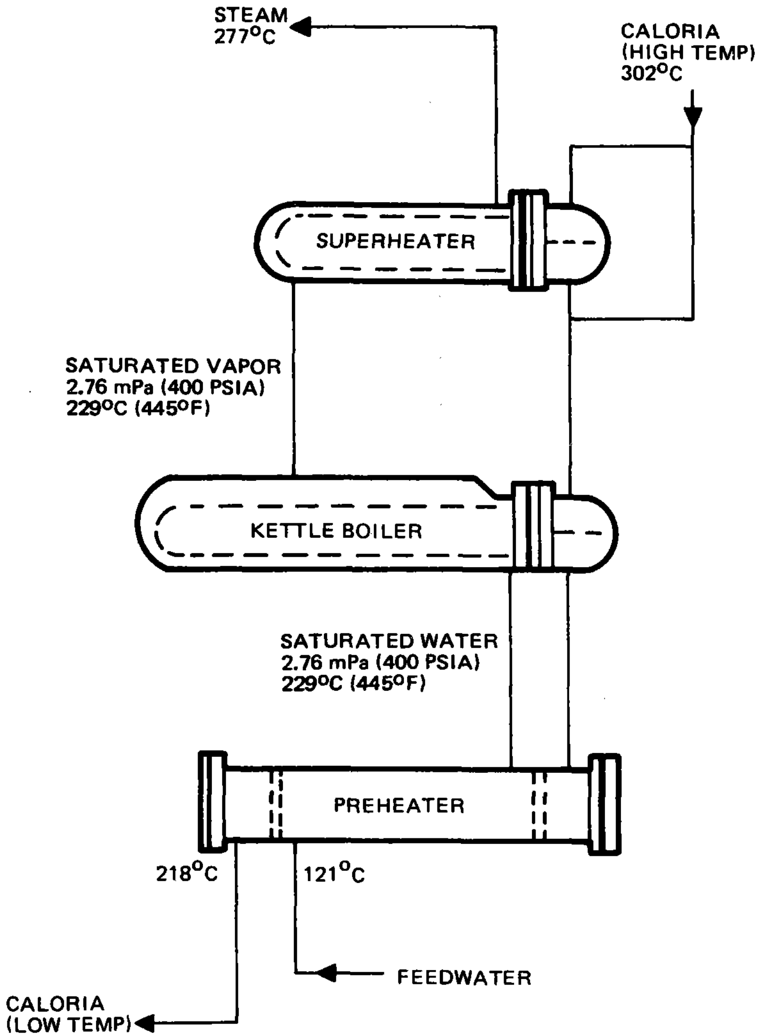
f = fluid side (water or steam as appropriate)

The basic design parameters which affect this relationship include:

- Tube diameter, length, and wall thickness
- Number of tubes
- Number of shell passes
- Tube spacing and pattern

In turn, these parameters, along with the shell size and resulting pressure drops, have a direct impact on the overall cost.

A computer program designed to identify possible combinations of the various independent design parameters which are capable of satisfying the preceding performance relationship was exercised to identify candidate designs. Subsequent economic analysis considering the equivalent annual cost of the heat exchangers, along with the cost of pumping power were carried out to aid in the design selection. Typical results of the economic analysis are shown in Figure 4-44 for the kettle boiler and Figure 4-45 for the superheater, along



EFFECTIVENESS	C_{RAT}*	N_{TU}	(UA)
0.657	0.390	1.23	21,290 W/°C) (40,355 BTU/HR-°F)
0.907	0	2.37	460, 450 W/°C (872,760 BTU/HR-°F)
0.944	0.160	3.32	96,438 W/°C (182,794 BTU/HR-°F)

$$^*C_{RAT} = \frac{(\dot{W}\,C_p)\ MIN}{(W\,C_p)\ MAX}$$

Figure 4-43. Steam Generator Schematic and Performance Data

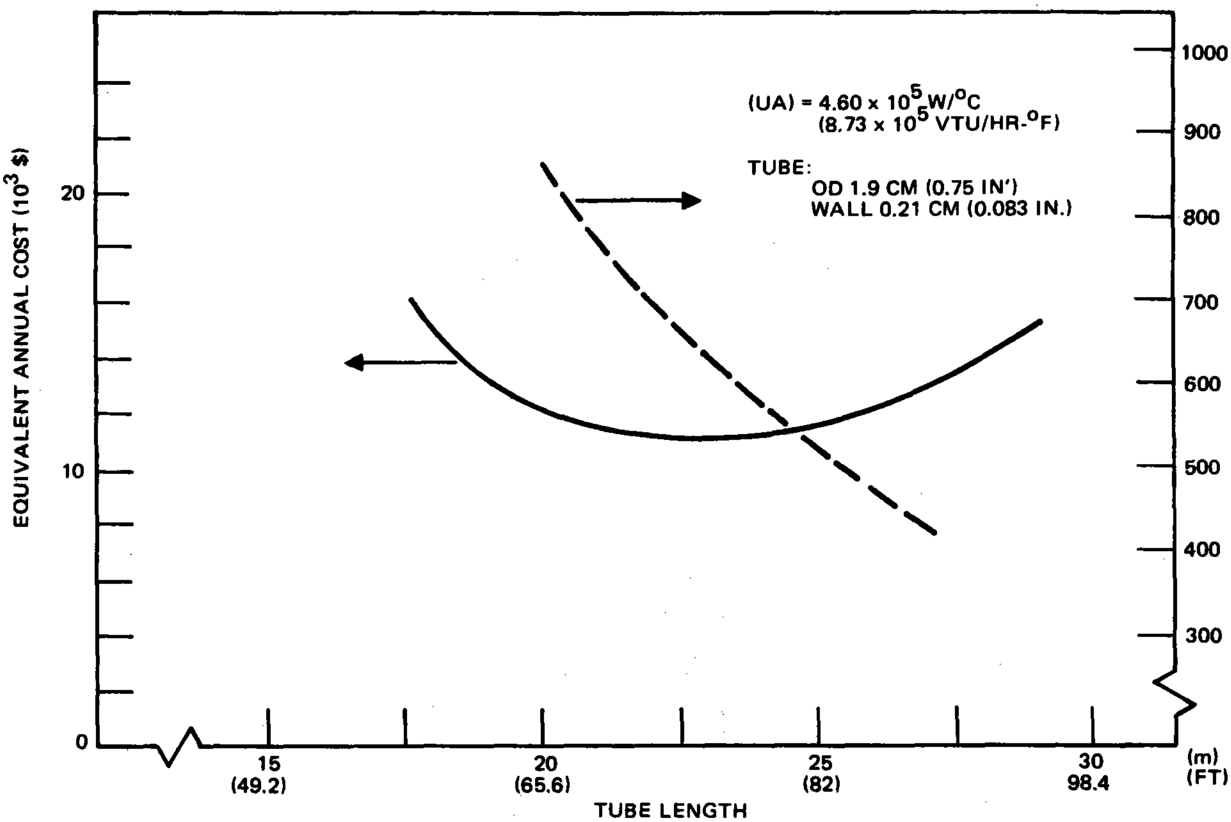


Figure 4-44. Optimum Kettle Boiler Desigr

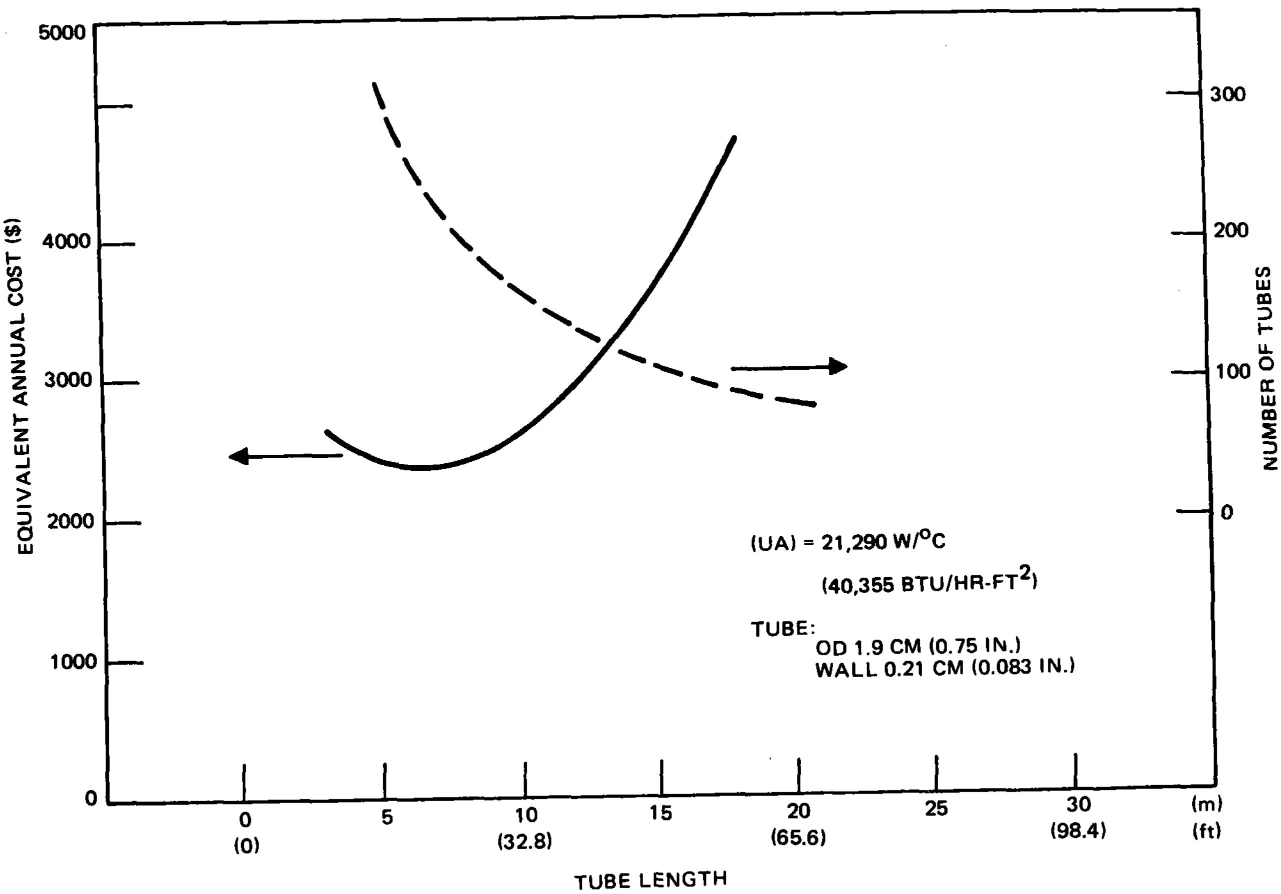


Figure 4-45. Optimized Superheater Design

with the number of tubes necessary to satisfy the overall heat transfer (UA) requirement. For the case of the kettle boiler, the initial cost decrease is due to a reduction in number of tubes and shell diameter with a corresponding reduction in shell thickness. As the tube length continues to grow, shell length and Caloria pressure drop effects begin to dominate resulting in the upward turn in cost toward the right side of the chart. A similar trend is noted in Figure 4-45, although the shell diameter and thickness effects are much less significant since the shells for the various cases considered are at or near the minimum gage limit. From these figures, it is possible to read directly the desired tube lengths and number of tubes required.

4.3.7.3 Design Description

The principal design parameters for the selected steam generation equipment are listed in Tables 4-24 through 4-26 for the superheater, kettle boiler, and preheater, respectively. The material in all cases is assumed to be carbon steel. Because of the undesirable effects which may result from leakage of Caloria into the feedwater circuit, tube to tube sheet joints will first be rolled to give a primary seal and then welded to provide backup protection. In addition, provisions will be made to provide GN_2 blanket pressure on the shell (water/steam) side of all elements during shutdown periods to ensure that the tendency for leakage would be toward the Caloria.

Figure 4-46 schematically illustrates a single heat exchanger train as it would interconnect with the rest of the subsystem. Included in this figure is data pertaining to the piping, valves, and sensors which would be included in a typical train.

4.3.8 Extraction Loop Fluid Pumps and Piping

4.3.8.1 Requirements

The function of the Pilot Plant thermal storage extraction loop, as the name implies, is to remove heat from the thermal storage unit to produce steam. Although similar to the charging loop with regard to the basic criteria (i.e., wide operating range, minimum capital and power cost, reliability, etc.), differences are shown by the design analysis.

Table 4-24

SUPERHEATER DESIGN SUMMARY

No. of Units	2
Configuration	Horizontal U-Tube, Crossflow
Tube-Side Fluid	Caloria HT-43
Shell-Side Fluid	Steam
No. of Tubes	225
Tube Length	7.0m (23 ft)
Tube OD	1.9 cm (0.75 in.)
Tube Wall Thickness	0.21 cm (0.083 in.) (BWG-14)
Pitch (Staggered)	2.38 cm (15/16 in)
Shell Length	3.65m (12 ft)
Shell Diameter	0.64m (2.1 ft)
Shell Wall Thickness	1.11 cm (0.438 in)
Mean Heat-Transfer Area	$84 m^2$ ($904 ft^2$)
No. of Passes	2
Insulation	15.2 cm (6 in.) thick

Table 4-25

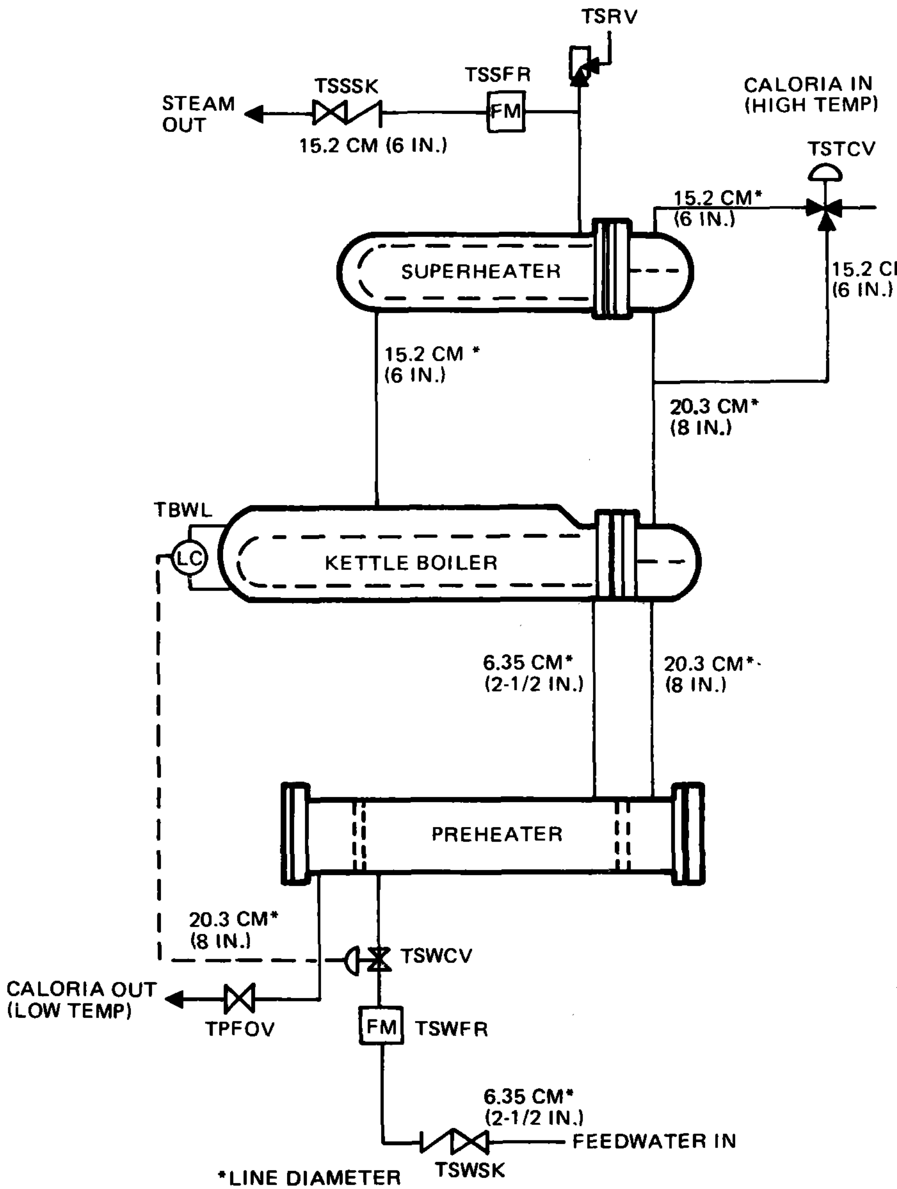
KETTLE BOILER DESIGN SUMMARY

No. of Units	2
Configuration	Horizontal U-Tube
Tube-Side Fluid	Caloria HT-43
Shell-Side Fluid	Water/Steam
No. of Tubes	650
Tube Length	22.9m (75 ft)
Tube OD	1.9 cm (0.75 in)
Tube Wall Thickness	0.21 cm (0.083 in) (BWG-14)
Pitch (In line)	2.38 cm (15/16 in)
Shell Length	10.82m (35.5 ft)
Shell Diameter	1.1/1.67m (3.6/5.5 ft)
Shell Wall Thickness	2.54 cm (1 in)
Mean Heat-Transfer Area	791m^2 (8,513 ft^2)
Insulation	15.2 cm (6 in.) thick

Table 4-26

PREHEATER DESIGN SUMMARY

No. of Units	2
Configuration	Straight Tube Floating Head Counterflow
Tube-Side Fluid	Caloria HT-43
Shell-Side Fluid	Water
No. of Tubes	431
Tube Length	8.54m (28 ft)
Tube OD	1.9 cm (0.75 in.)
Tube Wall Thickness	0.21 cm (0.083 in.) (BWG-14)
Pitch (Staggered)	2.38 cm (15/16 in)
Shell Length	9.15m (30 ft)
Shell Diameter	0.64m (2.1 ft)
Shell Wall Thickness	1.11 cm (0.438 in)
Mean Heat-Transfer Area	$196 m^2$ ($2,106 ft^2$)
No. of Passes	4
Insulation	15.2 cm (6 in.) thick



DESIGN DATA

(ALL CARBON STEEL)

TSWSK – 2-1/2 IN., 600 LB, STOP CHECK VALVE
TSWCV – 2-1/2 IN., 600 LB, PNEUMATIC CONTROL VALVE
TSRV – 4 IN., 600 LB, SAFETY RELIEF VALVE
TSSSK – 6 IN., 600 LB, STOP-CHECK VALVE
TSTCV – 8 IN., 150 LB, 3-WAY DIVERTING VALVE
TPFOV – 8 IN., 150 LB, GATE VALVE

LINES: (ASTM 106 GRADE B SCH 40)

INSULATION: CLOSED CELL GLASS
15.2 CM (6 IN.)
(HEAT EXCHANGERS)

SENSORS:

TSWFR – 2-1/2 INC., 600 LB, VENTURI FLOW METER
TBWL – EXTERNAL, CAGE-TYPE LEVEL SENSOR
TSSFR – FLOW TO CURRENT TRANSMITTER

Figure 4-46. Steam Generator Installation Data

The most distinct difference between the charging and extraction loops is in duty cycles. Unlike the charging cycle, which varies throughout the day, the extraction cycle operates at or near maximum conditions until all the heat is removed. The exception is during days of intermittent cloud cover where the flows will fluctuate. A typical daily extraction duty cycle is shown in Figure 4-47. The extraction pumps also will be required to operate at higher temperatures (typically 302°C as opposed to 218°C).

4.3.8.2 Design Analysis

The analysis of the thermal storage extraction loop proceeded in the same manner as that of the charging loop. The quantities of the various items of equipment assumed to be used are shown in Table 4-27. A schematic of the extraction loop is on the right half of Figure 4-33.

Important results from the design analysis are illustrated again with the velocities and number of pumps as independent variables in Figure 4-48 and Figure 4-49, respectively. The power cost is represented by the distance between the curves. Figure 4-48 shows the effect of velocity on total cost for one and four pumps. Unlike the charging loop, the power cost is not significantly affected by the number of pumps, but, as Figure 4-49 illustrates, the capital cost does increase. The minimum total operating cost for the extraction loop tends toward the higher velocities, and, as the electricity value increases, the optimum velocity decreases for the extraction loop. Although the extraction loop operated primarily at maximum conditions, the total annual operating time is low. The power costs are insignificant with respect to equipment cost which explains the higher optimum velocities. Figure 4-49 shows the effect of the number of pumps on total cost at different velocities. Although the capital cost increases, the power cost (distance between the curves) has little change. The power costs are not significantly affected, due to the extraction loop operating primarily at maximum conditions. Figure 4-49 illustrates that multiple pumps or dual-speed pumps provide no real advantage. The relative flatness of the curve permits a wide choice of the parameters while still maintaining a minimum cost.

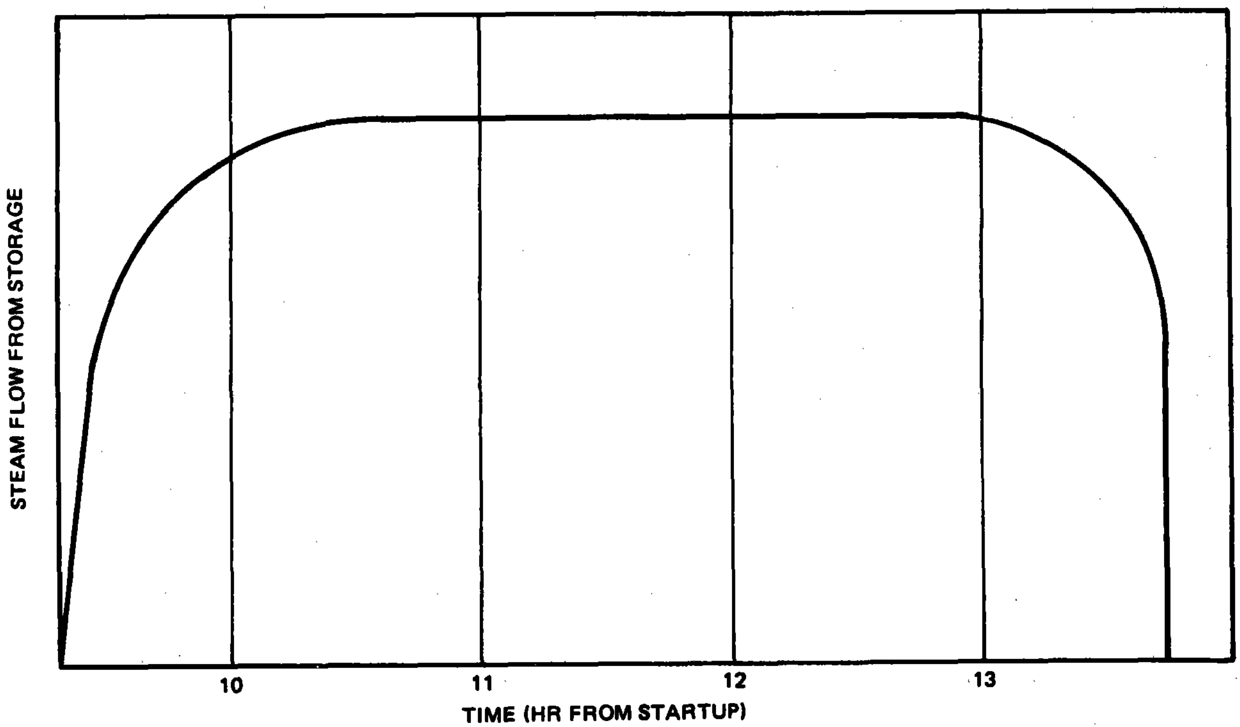


Figure 4-47. Typical Extraction Duty Cycle at Equinox

Table 4-27

PIPING COMPONENT SUMMARY EXTRACTION LOOP ANALYSIS

COMPONENT	QUANTITY
Pipe	69m (225 ft)
Valves	6
Tees	4
90° Elbows	8
Pumps	1-10

4.3.8.3 Design Description

An optimum design was developed based on a fluid velocity of approximately 5.2 m/s (17 fps), which corresponds to a pipe diameter of 22 cm (8.6 in.). Standard pipe sizes were chosen which yielded the closest velocity to that chosen, i.e., a 10-in., schedule 40 pipe size was selected, giving fluid actual velocities at maximum conditions of 4.7 m/s (15.3 fps). Two identical pumps in parallel with a total maximum capacity of 0.25 m^3/s (4,000 gpm) were decided upon, not on the basis of power cost, but because of reliability. These pump drivers are single speed (1,750 rpm) induction motors, 480V 3-phase. A summary of the final design parameters and equipment is given in Table 4-28. The flowrate head curves for the pumps are shown in Figure 4-50.

4.3.9 Controls and Instrumentation

4.3.9.1 Requirements

The control system will be required to respond automatically to commands from operating personnel and/or from the central controller. It will be required also to change smoothly and rapidly from one operating mode to another and, subsequently, to respond smoothly and rapidly to changes in thermal charging rate or extraction rate or both.

The basic requirement during the TSS charging mode will be to maintain the fluid output temperature from the charging heat exchangers at the command set

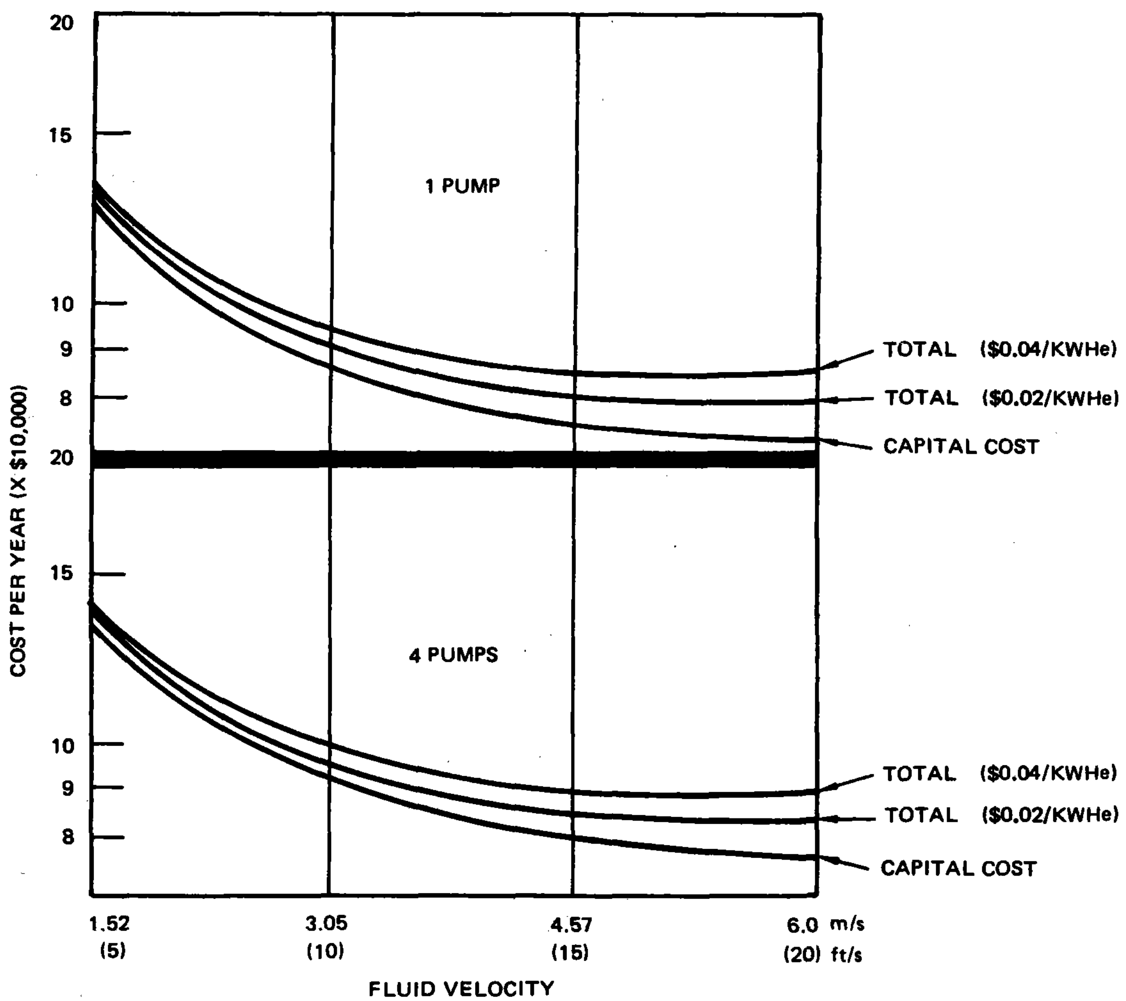


Figure 4-48. Annual Cost Optimization with Fluid Velocity, Number of Pumps, and Electricity Cost

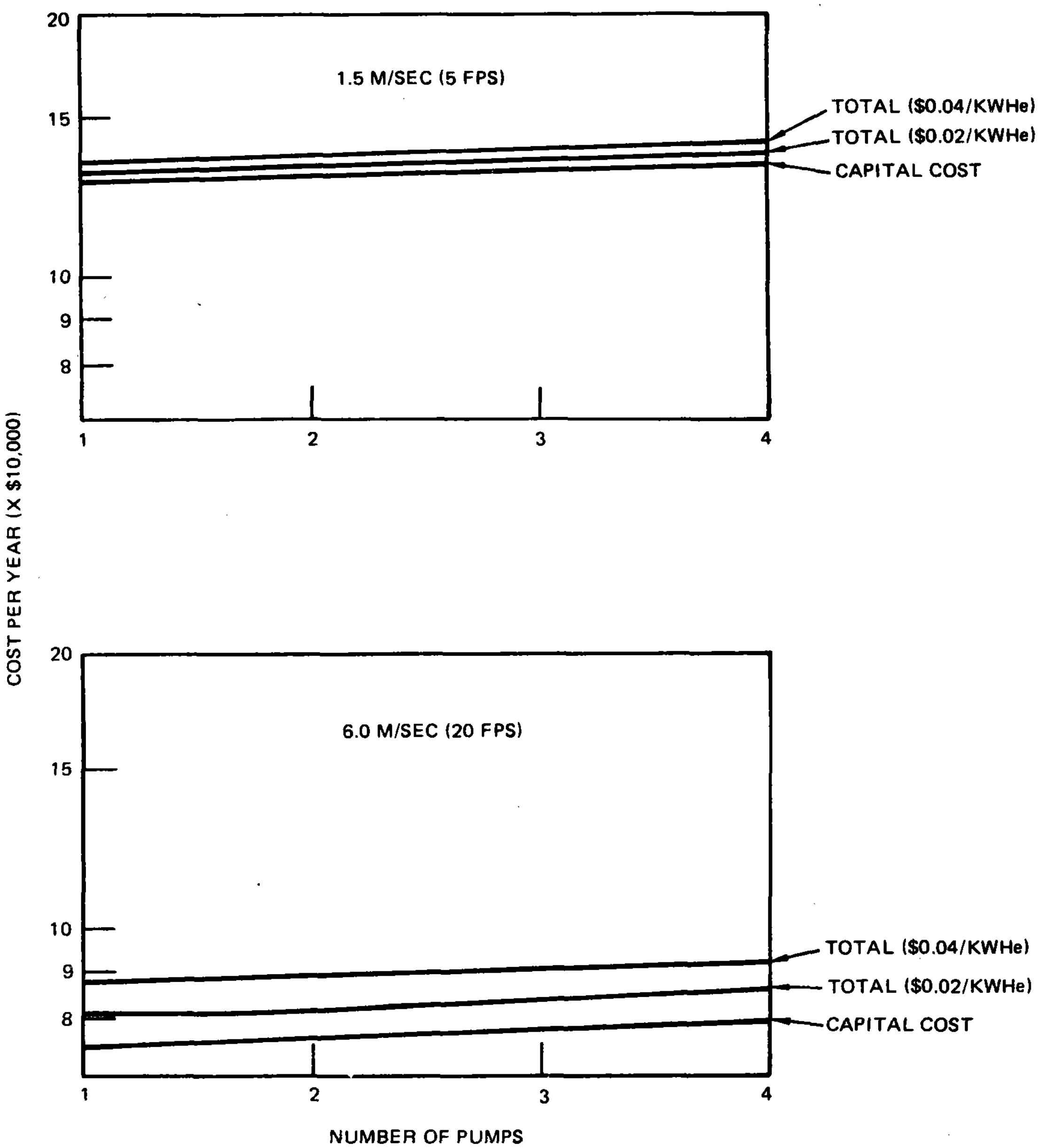


Figure 4-49. Annual Cost Optimization with Fluid Velocity, Number of Pumps, and Electricity Cost

Table 4-28

PILOT PLANT EXTRACTION LOOP FINAL DESIGN EQUIPMENT SUMMARY

Item*	Quantity	Description
Pump	2	Dean Brothers Model R454-6 in. x 8 in. x 12-1/2 in.
Motors	2	Single Speed, induction motor, 70 HP-maximum, 480V, 3-phase**
Pipe	52m (160 ft)	10 in. Schedule 40***
	21m (65 ft)	8 in. Schedule 40****
Valves		
Shutoff	5	8 in.
	1	10 in.
Throttle	2	8 in.
3-way	2	8 in.
90° Elbows	6	8 in.
Tees	7	8 in.
	2	10 in.

*Some items are common with the charging loop and have been omitted.
**Required motor input power.
***US Standard pipe sizes given (no SI equivalent pipe sizes).
****8-in. schedule 40 pipe was used for all segmented flow.

point, nominally 304°C (580°F). This temperature will be maintained within $\pm$1°C of the nominal setting as a steady-state error band. Transient overshoot can be allowed to be about $\pm$4°C from the nominal final value.

In general, the penalty for operating above the nominal maximum steady-state temperature is that the degradation rate of the fluid increases exponentially with temperature. Operation below the nominal reduces the total heat stored and reduces turbine efficiency. Transient errors in temperature can be allowed to be relatively large.

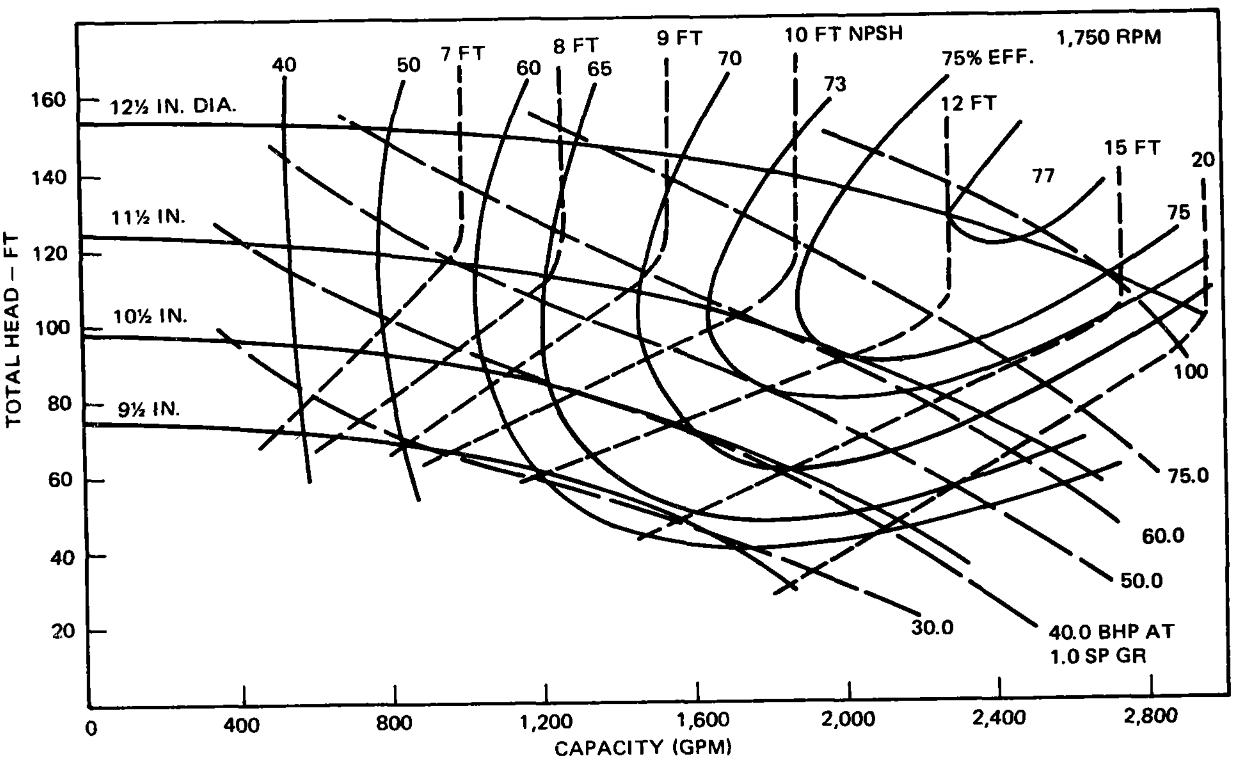


Figure 4-50. Pilot Plant Thermal Storage Extraction Loop Pump Characteristic Curves

In the extraction mode, the TSS must deliver steam at the nominal maximum temperature and pressure of 276±4°C (530 ± 7.2°F) and 2.76 ± 0.069 MPa (400 ± 10 psia) respectively, with a rather broad tolerance on the temperature and pressure, since tighter requirements by the turbine are not necessary.

The desuperheater control loop will be required to hold the desuperheated steam output temperature at 343 ± 1.1°C (650 F ± 2°F). Transient response can allow ± 5.6°C (10°F) overshoot from the final value.

In general, response times are not required to be rapid since the large thermal inertias involved are expected to provide the longest controlling lag time in all these control loops. In addition, TSS damage will not result from control errors or overshoot, since the subsystem is inherently a passive one. Only reduced efficiency will result except for the following possibilities:

A. Thermal stress fatigue in the heat exchangers.
B. Increased fluid degradation rate during excessively high temperature excursions.

Instrumentation capabilities are required for the following:

A. Direct monitoring of temperatures, pressures, flow rates and liquid levels with display and recording capabilities at a remote control panel.
B. The above analog signals are to be used by data loggers and/or computers to store data and calculate several parameters such as:
    1. State of charge of the TSU, i.e., KWHt available heat.
    2. Quantity of fluid in the TSS based on TSU temperature distribution vertically in the tank.
    3. Average fluid temperature in the TSU and a warning signal when this average falls below 169°C (350°F).
    4. Warning signals to operating personnel regarding violation of limits on critical parameters such as TSU tank pressure.
C. Provide recording, indicating and enunciating displays on the TSS section of the master control panel for use by operating personnel.

A combination of instrumentation, control, and logic functions will be required in the charging loop to aid personnel in automatically switching from one combination of pumps and heat exchangers to another as the demands on the system are changed.

4.3.9.2 Design Analyses

Instrumentation

The charge condition of the TSU will be determined by measuring the temperature distribution vertically through the bed. An array of 11 thermocouples spaced evenly 1.2m (4 ft) apart along the tank centerline between the upper and lower manifold will be used as the basic temperature measuring system for determining the thermal storage charge condition.

A second set of thermocouples will be installed in two groups, one just below the top manifold and the other group above the bottom manifold. Each group consists of 5 extra thermocouples spaced at 0.2m (1 ft) intervals vertically between the thermocouples already provided in the previous set. These measurements will furnish temperature profiles near the top and bottom manifolds to determine more accurately the steepness of the thermocline as it approaches either a full-charge condition or a fully discharged condition. These measurements will aid greatly the control personnel in predicting when to stop a charging or discharging operation to remain within control limits on the TSS system.

Another set of 20 thermocouples will be arranged in various places in the thermal storage unit. For example, about half way between the upper and lower manifolds there will be placed a horizontal array going from the outside wall to the center of the TSU. Similarly, thermocouples will be placed at strategic points near the upper manifold to permit study of the effectiveness of flow distribution accomplished by the upper manifold.

There will be other temperature measurements throughout the TSS system. Of the 150 temperature measurements, 130 will be thermocouples and 20 will be temperature bulbs. About 100 of the above temperature measurements will be associated with the TSU; of these 50 will be used for control purposes, and the others will provide additional data for Pilot Plant evaluation purposes.

The thermocouples used for control input will have a special redundant system. Two thermocouples are used for each measurement with both measurements brought to indicating instruments on the control panel. Control personnel can choose between the two thermocouples by throwing a switch; the switch will connect either thermocouple into the associated control loop, while the remaining thermocouple will give a temperature indication on the control panel only. The operator can compare these two readings with other readings in the system and thereby evaluate very quickly which one of the two is the more desirable to use for control purposes, and to evaluate whether a measurement element is faulty or not.

In general, pressure transducers are considered to be more reliable than thermocouples; therefore, a redundance has not been provided, except that virtually all the pressures that have been listed will be duplicated in the form of ordinary sight pressure gages mounted on the same pressure tap with the pressure transducers. In addition, pressure does not enter as directly into the control loops as temperature does. During the detailed study phase, a re-evaluation of whether or not to duplicate certain critical control loop pressure measurements will be made.

System Dynamics

The general equations describing the system dynamics of the thermal storage system (TSS) were written and were included as part of the entire power plant dynamic study which is reported elsewhere (see Volume II, Section 4.7).

However, certain predictions can be made regarding the TSS dynamic response based on practical experience and observations made during the SRE experiment. From this, it is concluded that the controlling time lag or time response of sensing elements and signal-conditioning equipment is on the order of a few seconds or shorter, the time response of valves are on the order of tens of seconds while the time response of the heat exchangers and associated fluid flow lines are on the order of minutes. The limits on time response of the entire system to changes in demand are, therefore, limited only by the thermal inertia of the passive elements such as heat exchangers and not by the sensing and active control elements such as valves.

Adequate control system stability is predicted to be easily achieved, based on this simplified time constant analysis. However, the system will display transport lag which may be adequately dealt with at the nominal design maximum throughput, but which may change drastically as the turndown ratio is increased, resulting in large increases in transport lags.

The feed-forward type automatic controls used are designed to deal with the above problem as described in Section 4.3.9.3 under the subheading of the control loop under consideration, e.g., Charging Loop Controls, etc.

The time lags, t_1, t_2, t_3, can best be determined once detailed design specification for the heat exchangers and associated piping have been established during Phase 2. Meanwhile, results forming part of the overall power plant dynamics are reported in Volume II, Section 4.7.

TSS Remote Control

The Thermal Storage Subsystem Panel (TSSP) provides for operator control of thermal storage operations. All remote control valves, on-off, and modulating can be controlled with manual inputs. The panel will include display equipment to the extent required for operator supervision and manual control, and will include recording equipment to the extent required for Power Plant record keeping and for operational analysis.

In addition to the provisions for operator manual control, the TSSP has provisions for semiautomatic control with operator supervision. In this mode of operation the modulating valves are under closed-loop control to maintain temperature settings which are operator adjustable. Additionally, the operator can transfer operation to a fully automatic mode with limited control by the programmable master controller. In the automatic mode of control, the TSSP continues to provide the operator with supervisory data display and recording (see Figure 4-51).

Signals from the TSS sensors are brought to a nearby, unmanned, weatherproof location where an electronic cold junction box is located for all the thermocouple wire terminations. Terminal strips, control relays, etc.,

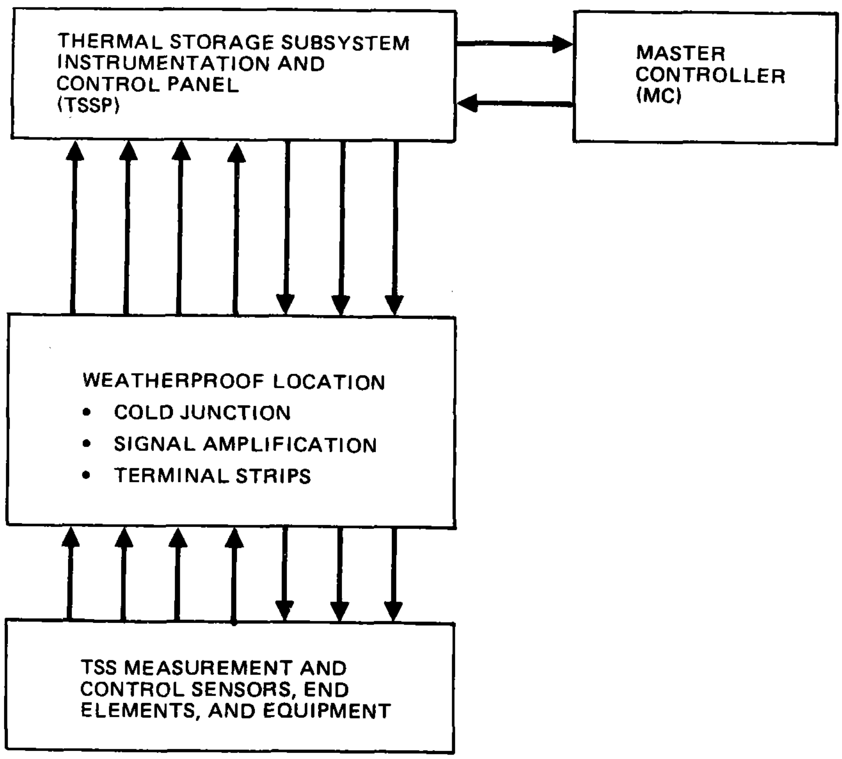


Figure 4-51. Block Diagram of Information Flow in the TSS

are also located here. Copper wire cables and relatively high-level signals are then carried (in conduits separate from those carrying power cables) to the control room and the control panel.

The TSS control loops are basically independent, electronically, of one another. However, a control logic system will be included having high flexibility so that some automatic interaction can be provided for, after experience is gained with the Pilot Plant system characteristics. This same provision will be made for the interface with the master controller (MC). The operator at the control panel will have the option of overriding these interconnections so he can achieve manual control of any component, as necessary.

The control panel includes the circuits for closed-loop analog control of the various heat-transfer fluid flowrates. Additionally, there are logic circuits for fluid pump starting, stopping, and speed changes, as flowrate demands vary. Logic circuits (to ensure correct sequences of events and to provide alarm and safety cutoff signals) are included as well as signal-conditioning circuits for all the parameter-sensing elements. Buffer circuits are provided which are designed to isolate the TSSP and the MC from each other, so that neither can affect the other.

Control Logic

The thermal storage system contains a number of areas in which decisions can be made regarding alternative operating modes, i.e., whether one or two heat exchangers should be used to respond to a given demand or whether to use some other combination of pump speeds or number of pumps. Although all these decisions can be designed to be automatic, the guiding strategy will be to provide the plant with these logic circuits (or special computer programs) but to defer connecting their outputs to the system, i.e., they will function to give information only as a guide to the operator as to what action should be taken. The final action will be taken by the operator. It is presently estimated that the time constants associated with the TSS will be long compared to operator response time. The rapidity afforded by automatic logic circuits is not required as regards response time. Repetitive decisions causing operator boredom can profitably be accomplished using logic circuits. Operational experience with the Pilot Plant can best provide input for evaluating such requirements. The following is given as an example of the type of analysis which will be made during the detailed design (Phase 2) regarding logic-type decisions, and the design to implement it.

The following sentences discuss the criteria used to design that part of the control logic which makes decisions automatically as to whether, at any given moment, pump speed should be high or low or whether one or two heat exchangers need be used when the TSS is in the charging mode. Precise switching points will be established during the detail design (Phase 2). The values given here are only approximate. The following discussion begins with steady operation at the maximum nominal charging rate.

Referring to Figure 4-52, and starting at Point C, as demand drops off from the maximum nominal value, the two fluid throttle valves, THFIV-1 and THFIV-2, will close until the flow decreases to a little less than 2/3 of the nominal maximum value. A logic circuit then will switch the two electric pump motor circuits so that the fluid pumps will run at 1, 150 rpm instead of 1, 750 rpm. This sudden change will return the two throttle valves almost to full open position thereby increasing the flow to the same level as before the change in pump speed.

If demand is reduced still further to a point just below one-half nominal maximum, then one pump will be shut down and its associated throttle valve will be thrown to the maximum closed position by the control logic circuit to render one heat exchanger idle. The system then operates with only one pump and one heat exchanger. From this point down to the minimum nominal flowrate (of 1/30 maximum), one of the throttle valves will throttle the flow through one heat exchanger, using only one pump running at 1, 150 rpm.

It is anticipated that the TSS will be in a charging turndown ratio of from 1/30 to 1/4 (3% to 25% flow) most of the time, a range for which only one pump and one heat exchanger will be required with the single pump running at its lower speed (1, 150 rpm). However, a different switching criterion will be used on increasing demand. If the flow required by one pump at the

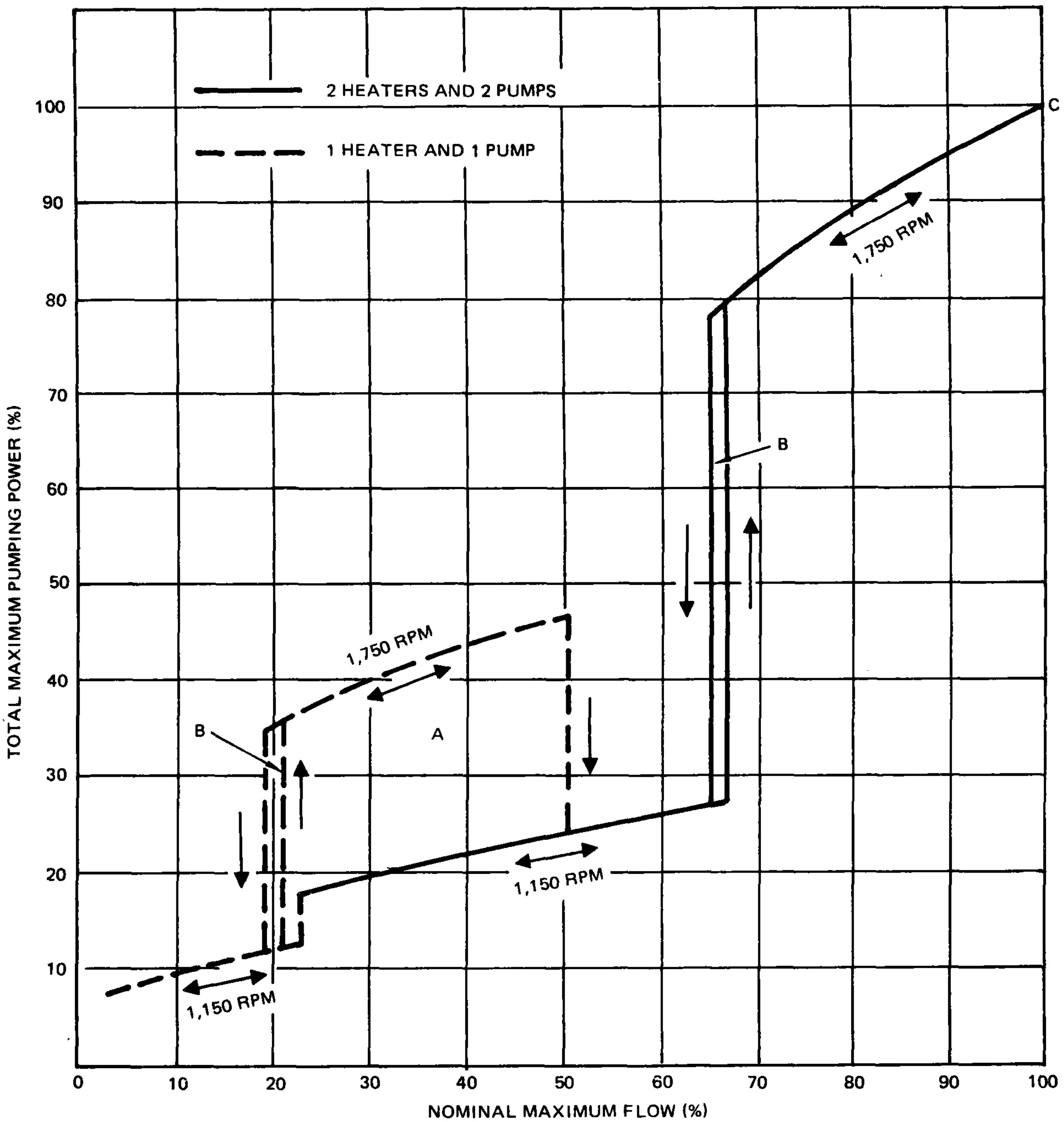


Figure 4-52. Control Logic Criteria for Determining Operating Modules in Charging Loop

lower speed is not sufficient, the pump will be switched to the higher value of 1,750 rpm (rather than performing the reverse of the switching scheme provided for in the case of decreasing demand, i.e., switching to two heat exchangers). The reason for the different criterion is that one heat exchanger and one pump should be used in the low demand regime (high turndown ratio) for as long a time as possible once the second heat exchanger has been shut down. This strategy minimizes thermal cycling of the second heat exchanger. Similarly, once increasing demand has required the system to use two heat exchangers, switching back to one heat exchanger on decreasing demand is not undertaken until a considerable time has been spent at a flowrate several percent below the nominal switchover point near 25% of maximum flow.

Manual override and control commands by the MC will be possible since it will have predictive ability regarding future turndown ratios and can provide switchover point decision supervision and override with the objective of minimizing pumping power requirements and maximizing heat exchanger life by reducing thermal cycling.

The control logic design also builds a stability through an inherent "toggle" action, in which the decision process at switchover points allows the system to proceed beyond the nominal switching point before a command to switch is executed. This prevents the system from cycling back and forth when demand requires continuous operation at or close to a nominal switching turndown ratio. This applies both to switching from one to two heat exchangers (large loop marked A in Figure 4-52) and to the speed switching points (small loops marked B in Figure 4-52). The extra pumping power required when time is spent on the dotted line above the letter A (compared to the solid line below the letter A) is the small price paid for reducing the total number of cycles which one of the two heat exchangers must withstand.

The other possible combinations of using one pump with two heat exchangers and two pumps with one heat exchanger, each at the two pump speeds, has similar logic and considerations. The final control logic design will generalize and simplify to limit the number of possible operating choices, thereby relieving the operator of having to make too many decisions.

4.3.9.3 Design Description

The description of controls is rather lengthy, to communicate details. The description is organized under five subheadings: Charging Loop Controls, Extraction Loop Controls, Desuperheater Controls, Ullage Maintenance and Fluid Maintenance Unit Controls, and Control Equipment.

Charging Loop Controls

Figure 4-53 is a schematic diagram of the piping system and control elements that are required for thermal storage heater fluid flow control. Heat-transfer fluid from the lower manifold of the thermal storage unit (TSU) is pumped through the thermal storage heaters, TH-1 and TH-2, in which the fluid temperature is increased by heat transfer from a supply of steam from the desuperheater.

Two centrifugal thermal storage charging pump units in parallel, TCP-1 and TCP-2, are powered by two-speed alternating current motors. Check valves in the pump discharge lines prevent backflow of fluid through a pump that is shut down while the other pump is in operation. Flowmeters THFFR-1 and THFFR-2 measure the flowrates to each pump.

The charging controller (CC) which physically is a portion of a larger assembly identified as the Thermal Storage Unit Controller, modulates thermal storage heater fluid inlet valves THFIV-1 and THFIV-2 for continuous adjustment of each pump flowrate to maintain constant heater fluid outlet temperatures THFOT-1 and THFOT-2, as heating steam flowrates vary. Electronic circuits in the charging controller process the signals from each set of two redundant temperature sensors. The two pumps have nominally identical pressure-flowrate characteristics when operating at the same speed level. When both pumps are operating at reduced speed, and charging-loop total flowrate increases to an extent that the pumping capacity is approached, logic circuitry in the charging controller then switches both pump speeds to the full-speed level. Both pumps again have nominally-identical pressure-flowrate characteristics, and control valves THFIV-1 and THFIV-2 continue to operate with nearly identical pressure drops, maintaining equally good throttling control characteristics in the parallel flow paths. Automatic

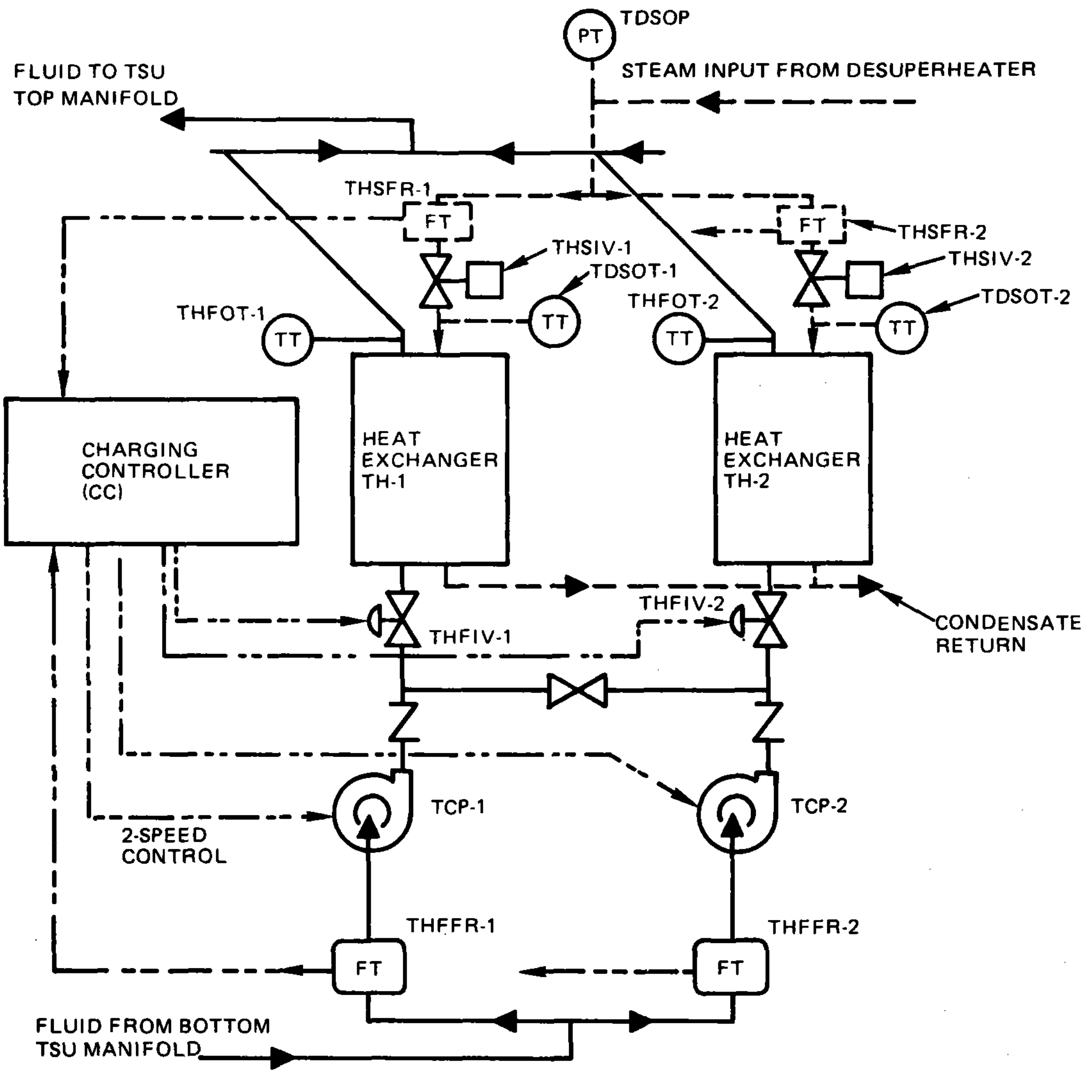


Figure 4-53. Charging Heat Exchanger Control Loop Components

control of pump speeds, in response to charging controller logic, is utilized to minimize pump electric motive power demand. An analysis of the switching logic is given in Section 4.3.9.2.

Modulating butterfly valves THFIV-1 and THFIV-2 were selected because of their simplicity and adequate throttling control characteristics with low pressure loss under maximum flowrate conditions. A type with metal-to-metal seating contact between the valving element and the housing will provide adequate shutoff capability for this application, with no need for a flexible seat seal with limited endurance capability.

Valves THFIV-1 and THFIV-2 are pneumatically actuated, with a nitrogen supply pressure of 100 to 150 psia. Closed-loop control of valve displacement from its closed position is provided by a positioner. A 3-15 psig pneumatic input signal to the positioner is related to a mechanical feedback measurement of actuator and valve position, through a servovalve mechanism, so that a 0 to 100% valve displacement corresponds to the positioner input signal range. An electropneumatic transducer coverts an electric current signal input, 10 to 50 milliamperes, to a 3 to 15 psig pneumatic output signal to the valve positioner. Control valve displacement is thereby controlled by electrical command signals from the charging controller portion of the Thermal Storage Unit Controller.

Solenoid pilot operated steam inlet valves THSIV-1 and THSIV-2 provide for on-off control of steam flow through each heater. Steam inflowrates to the heaters are measured by flowmeters THSFR-1 and THSFR-2. Steam pressure and temperature measurements obtained with transducer TDSOP and redundant sensors TDSOT-1 and TDSOR-2 provide data to the charging controller for relating steam weight flowrate to the measured volumetric flowrate, and for determing the incoming available heat energy. Redundancy is not provided for the pressure measurement because of the inherent greater reliability in pressure instrumentation relative to temperature instrumentation. However, pressure measurement redundancy can be provided easily if failure-mode-effect analyses, in support of a finalized design, indicate a need which warrants additional complexity.

During a time interval in which the number of pumps and the number of thermal storage heaters in operation are adequate for absorption of the available heat energy, heat-transfer fluid flowrate through each heater is controlled as indicated by the block diagram of Figure 4-54 using the control elements that are identified in Figure 4-53.

When both heaters are in operation, with shutoff valves THSIV-1 and THSIV-2 both open, the steam flow tends to divide evenly between the two heaters. If needed, a trimming orifice can be added to one of the heater steam lines to equalize the two heater steam flow resistances.

If the steam pressure and temperature are within their normal range, weight flowrate of steam is approximately proportional to volumetric flowrate. This is indicated in Figure 4-54, presuming a first-order time lag in the weight flowrate measurement, using conventional transfer function notation. With more complex transfer functions (not shown), the steam pressure and temperature can also be used for steam weight flowrate computations throughout the full range of variable operating conditions.

If all system controlled variables are at their nominal regulated values, the weight flowrate of heat-transfer fluid through the heaters will be throttled to be proportional to the heat energy to be extracted from the steam, i.e., heat-transfer fluid weight flowrate will be controlled to be proportional to steam weight flowrate. Therefore, as indicated in Figure 4-54, a fluid weight flowrate command signal, $\dot{W}_1$, is generated in proportion to the measured value of steam weight flowrate. As will be discussed later, close accuracy in generating this command signal is not required; errors related to variations in system variables, from scheduled nominal values, are acceptable.

Thermal storage heater fluid discharge temperature, as delivered to the TSU, is measured by redundant temperature transducers. As indicated in Figure 4-54, a first-order time lag in temperature measurement is presumed. The measured average temperature signal is compared with a reference temperature signal. As indicated by the block with the transfer function D/s,

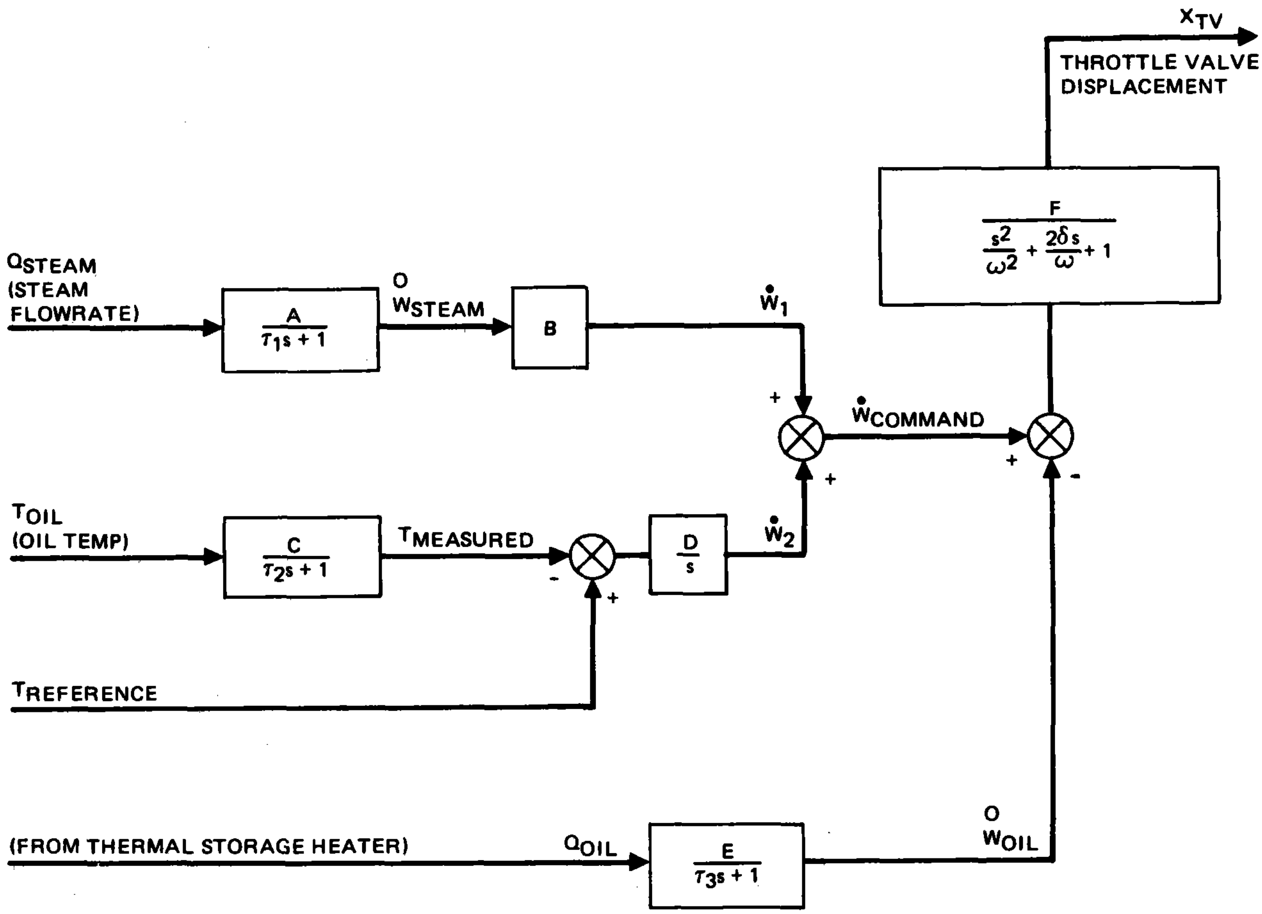


Figure 4-54. Block Diagram of Charging Fluid Controls

temperature error is continuously integrated, and an oil weight flowrate command signal, $\dot{W}_2$, is generated to eliminate steady-state error.

The weight flowrate command signal is shown in Figure 4-54 as the sum of $\dot{W}_1$ and $\dot{W}_2$. The flowrate command signal is compared with a flowrate measurement signal that is obtained through the use of a flowmeter. A difference between commanded flowrate and measured flowrate results in control valve (tag number THFIV-1 or THFIV-2) displacement and in consequent fluid flowrate control. A typical second-order response transfer function is presumed for modulating throttle valve closed-loop control.

With this temperature control concept, the commanded heat-transfer fluid flowrate signal is primarily a scheduled function of the system variable input steam flowrate with trimming corrections as required in eliminating steady-state error in the controlled output (fluid temperature).

Transient response time lags in measurements of flowrates typically are orders to magnitude less than the dominant time lags in this control loop. Changes in incoming steam flowrate will result in fast-response changes in fluid flowrate, with fluid flowrate scheduled to be approximately correct in maintaining the reference temperature set-point value. Changes in fluid flowrate, resulting from integration of temperature error, will occur at a relatively slow rate.

The dominant time constants in the temperature control subsystem are related to thermal response time lags in the heat exchanger tubes and in the temperature sensors. Typically, in a heat-exchanger continuous-temperature-control process, in which the control loop is closed on controlled temperature measurement and a reference signal only, output/input phase lag under transient operating conditions results in an inherent tendency toward oscillatory operation. A low-gain temperature-control loop, as required for dynamic stability, generally results in excessive temperature control bandwidth and in excessive transient error in response to input excursions. The addition of an integrator to minimize bandwidth and error, in a loop which is closed on temperature signals only, generally is not feasible, because the additional phase lag that an integrator introduces will usually result in a dynamically unstable control loop.

With the control concept illustrated in Figure 4-54, fluid flowrate is scheduled to track excursions in steam inflow, whether or not the outlet temperature is in error, e.g. corrective action is initiated by a change in input before an output error occurs. Integration of temperature error is introduced for trimming purposes only, in a manner that does not contribute phase lag to the basic scheduled control loop.

The thermal storage heater flow control concepts which have been described in this discussion are designed for "load following" capability. However, as is often the case in Power Plant systems, efficiency and capability may be improved under transient operating conditions if the Power Plant master controls are designed to provide "feed forward" signals to the storage subsystem to result in anticipatory response to significant impending changes in operating conditions.

Under conditions in which the incoming steam conditions are inadequate for heating the heat-transfer fluid to an outlet temperature of 302°C, a warning signal will be given to operating personnel so that the charging operation can be halted immediately. After experience is gained, this function can be automated easily if desired.

Extraction Loop Controls

Figure 4-55 is a schematic diagram of the plumbing system and control elements that are required for operation of the TSU extraction loop.

In the extraction mode of operation, heat-transfer fluid is pumped from the upper hot end of the TSU, through two parallel steam generators, to the lower cooler end of the TSU. Two centrifugal pumps, powered by alternating current electric motors, operate in parallel. A manually operated hand shutoff valve, with tag number TSFBV, interconnects the pump outlets so that either pump can be used to pump fluid through either of the two steam generators. Check valves prevent backflow of fluid through a pump that is shut down while the other pump is in operation.

Because it is anticipated that most of the steam generator operating time will occur with steam flowrates in a narrow range near maximum, single-speed

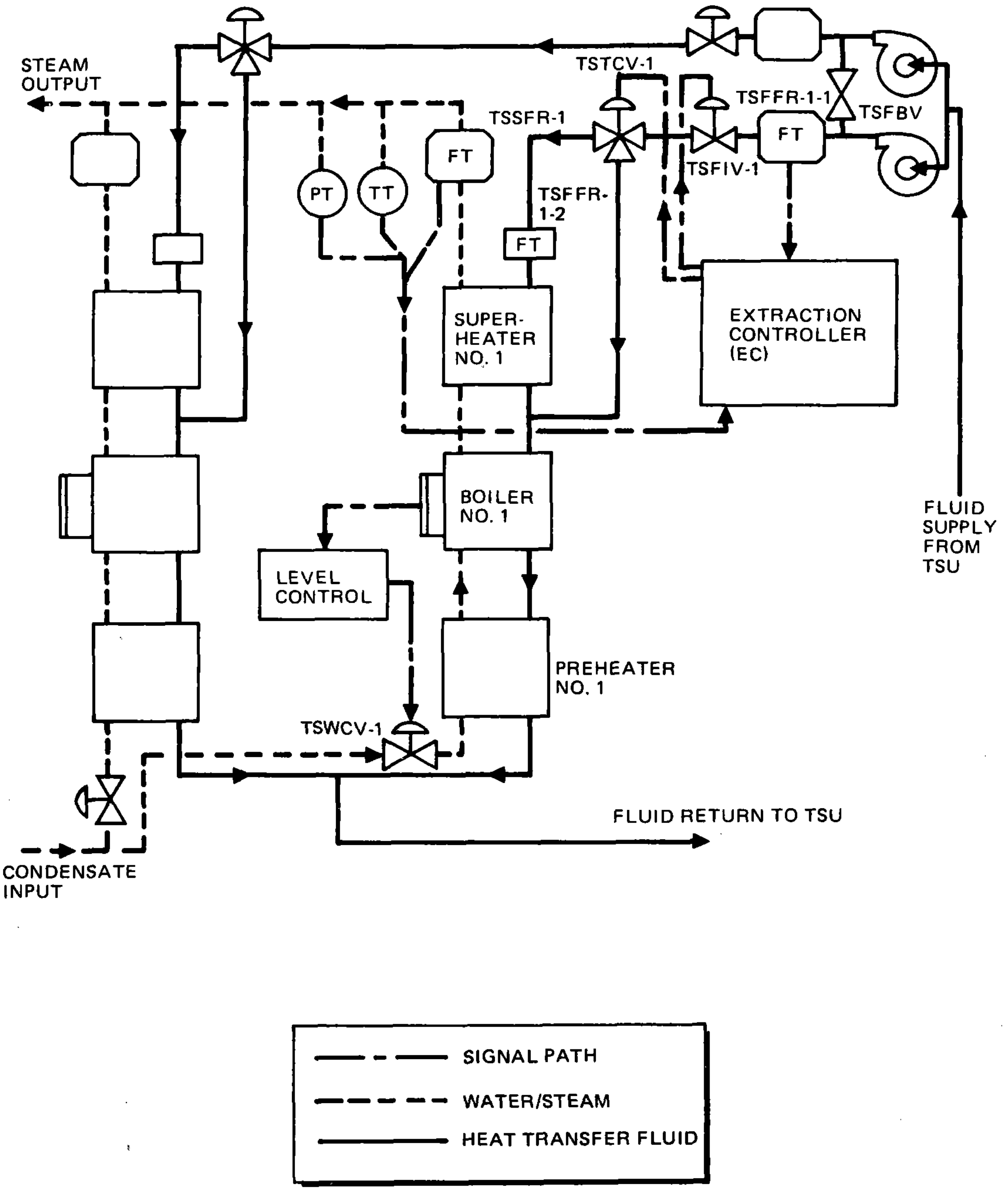


Figure 4-55. Block Diagram of Extraction Loop Controller

pump motors were selected. Under conditions of low steam flowrate demands, operation can be obtained with one steam generator and one pump.

The TSU heat energy extraction loop is controlled for automatic response to variations in demand for heat energy transfer. When the two steam generators are operating in parallel each is controlled independently with control circuits described in Figure 4-55 and 4-56. Steam generator control can best be understood by walking through a typical extraction cycle. For purposes of this example, we will assume that the steam generator is in a hot standby mode with only a small steam discharge flow to maintain system heat.

The superheater fluid inlet valve, TSFIV is positioned as required to maintain, primarily, superheater outlet steam flowrate as called for by the master controller, i.e., steam $\dot{W}$ command, and secondly, to maintain the superheater steam outlet pressure. Steam pressure is measured and compared with a reference steam pressure and any difference is added to the difference between the measured and demand steam mass flow rate. The resulting signal is amplified and compared to a signal proportional to the thermal storage fluid flowrate and the difference between these two signals alters the throttle valve (TSFIV) displacement in such a direction as to maintain the command steam mass flow rate at demand.

The influence of the pressure control is automatically reduced to a negligible value except when steam flow demand is reduced to near zero, at which time the pressure controller acts to maintain the pressure at the proper level even though the mass flow demand may be zero.

In general, upon receipt of a command from the power plant operator, or master control, to provide a specific steam output from thermal storage to the turbine, TSFIV is opened to the appropriate position to provide the necessary hot fluid flow to the steam generator. As steam flow increases the turbine admission steam valve will open to maintain system turbine inlet pressure. If all system controlled variables are at their nominal regulated values, the weight flowrate of heat-transfer fluid through the steam generator will be approximately proportional to the steam weight flowrate. Therefore, as indicated in Figure 4-56, a fluid flowrate command signal, is generated in proportion to the measured steam flowrate.

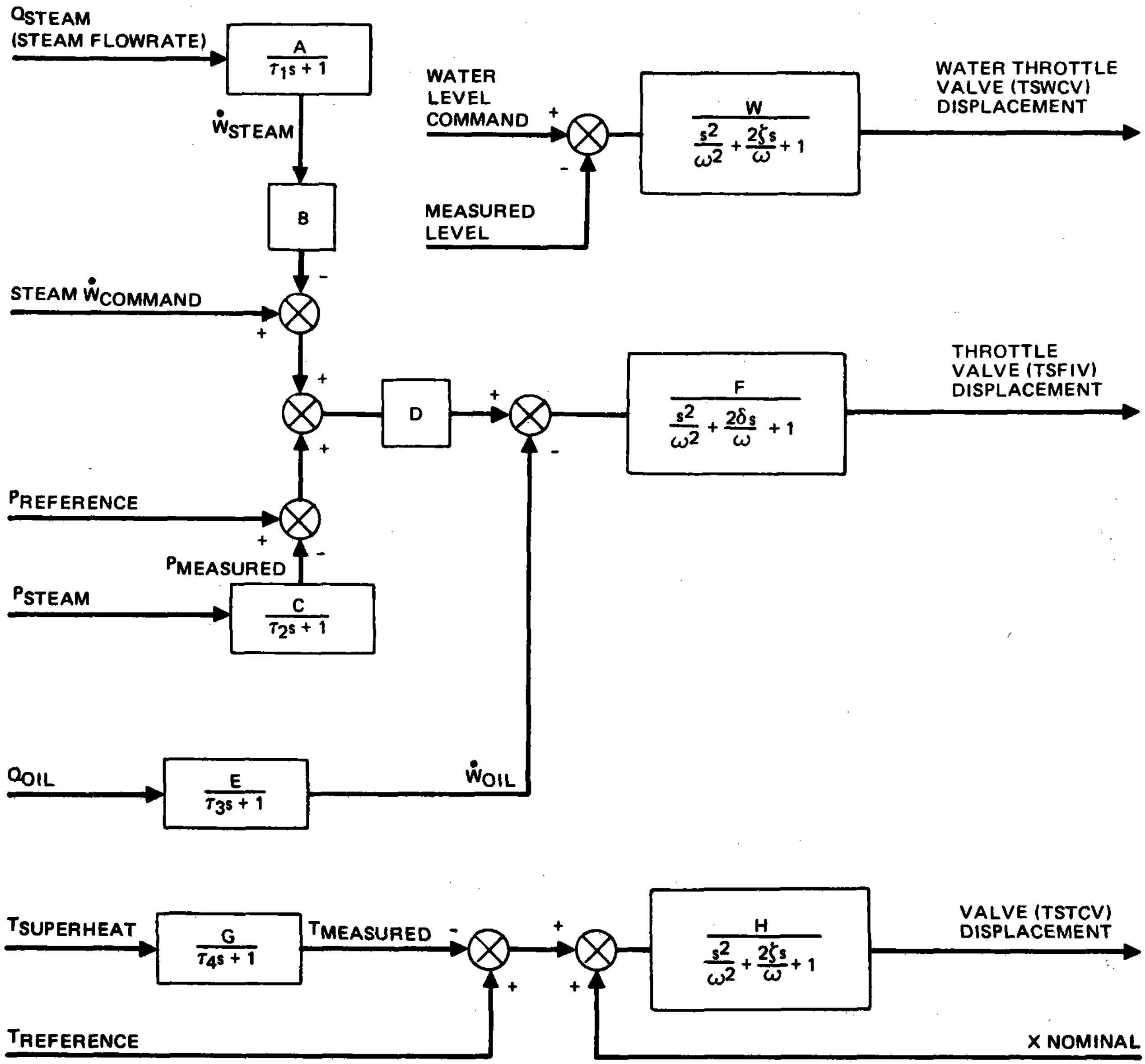


Figure 4-56. Block Diagram of Thermal Extractor Loop Controls

The superheater temperature control 3-way valve, TSTCV, is positioned as required to maintain superheater steam outlet temperature. Steam temperature at the superheater outlet is measured by redundant temperature sensors. As indicated in Figure 4-56, a first-order time lag in temperature measurement is presumed. The measured average temperature is compared with a reference temperature signal and the difference, or error signal, is used to reset the 3-way valve (TSTCV) position in such a direction as to bring the temperature back to the reference value. The flow not going to the superheater is bypassed to enter the boiler along with the exit flow from the superheater.

The steam generator water control valve, TSWCV, maintains proper water level in the boiler under all demand conditions upon command from the boiler water level sensor. A liquid-level sensor measures the boiler water level, which in turn is compared with a water level command value. The resulting difference or error signal is amplified and sent to reposition the water inlet control valve to the preheater in such a direction as to bring the liquid level back to the command level.

The steam storage volume in the steam boiler provides significant system capacitance for dynamic stability with relatively slow control response. By scheduling the total fluid flowrate as a fast-response function of steam flowrate, the other loops can be relatively slow, to prevent their interaction. If system dynamic analysis indicates a need, the simple block diagram of Figure 4-56 can be modified to incorporate analog control compensation features in combination with supplementary logic functions. The use of solid-state electronic circuits in the extraction controller permits simple mechanization and/or modification of the basic circuits.

Analog flowrate control is supplemented by extraction controller logic and sequence-of-events control for operation with either one or two of the steam generators.

With interconnect valve TSFBV open and both pumps in operation, if one pump should fail or inadvertently shut down, the other pump can supply fluid to both steam generators, within the capacity limitations of one pump.

Under such conditions, the extraction controller provides a signal to the power plant master controller to limit the steam flowrate demand. The parallel pump design enhances system reliability and minimizes electrical power requirements by permitting one pump operation during low flowrate demands.

The two parallel paths through the extraction loop steam generators are controlled automatically by the extraction controller (EC). Physically, the extraction controller is a portion of a larger assembly that is identified as the Thermal Storage Unit Controller.

Desuperheater Controls

Figure 4-57 is a schematic diagram of the plumbing system and control elements that are required for control of the thermal storage unit inlet steam desuperheater (DSH).

Under conditions for thermal storage unit charging loop operation, steam at 950°F is delivered to a DSH with a tag number TD. The steam flowrate is an independent variable, and is the difference between the flowrate from the solar energy receiver panels and the turbine steam flowrate demand. Steam turbine inlet temperature is controlled by the receiver panel subsystem. Steam turbine inlet pressure is controlled by bypassing steam to the thermal storage subsystem.

The power plant will include provisions for maintaining a back pressure in the thermal storage unit condensate water return line. The DSH steam inlet pressure will therefore vary with steam flowrate and with variations in the regulated back pressure.

Under conditions for thermal storage unit charging loop operation, DSH inflowing steam is mixed with water to regulate the DSH outlet steam temperature at approximately 650°F. As steam from the DSH passes through the thermal storage heaters, identified with tag numbers TH-1 and TH-2, the steam condenses and is returned to the back-pressure regulating subsystem. As indicated in Figure 4-57, DSH coolant water is supplied from the solar energy receiver panel feedwater pump at approximately 2100 psig. Coolant water passes through a flowmeter with tag number TDWFR, and water flowrate is throttled by water control valve TDTCV. The control valve discharge pressure is approximately 1,450 psig.

The pressure drop through the control valve under flowing conditions is approximately 30% of the valve inlet pressure. The ratio of pressure drop to inlet pressure is sufficiently great that flow cavitation would occur, or be

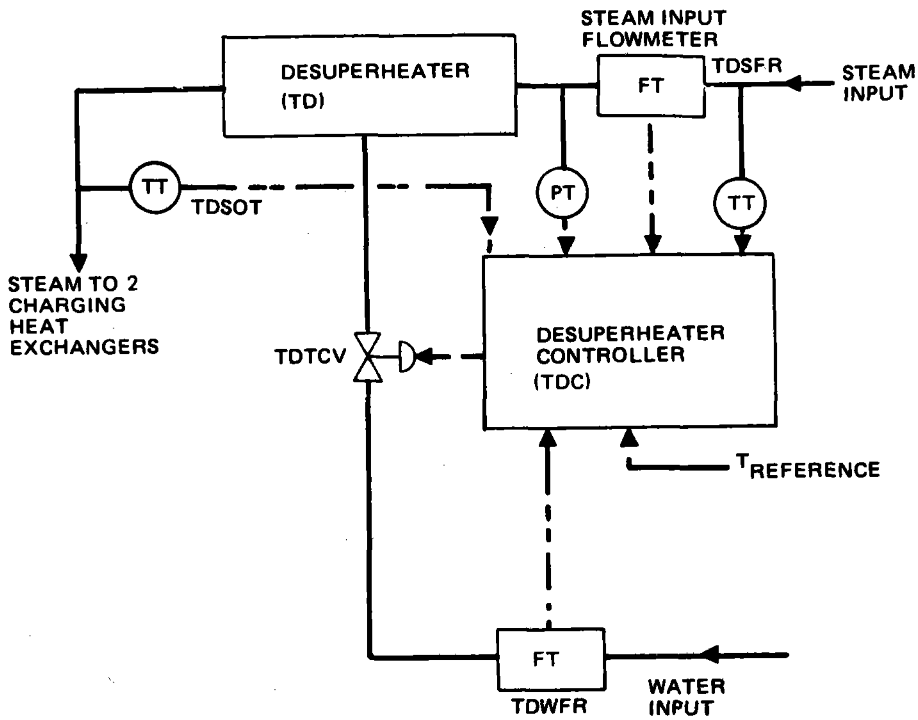


Figure 4-57. Desuperheater Control Loop Components

incipient, in the most commonly used control valve configurations, and operation would be noisy. Therefore, to ensure cavitation-free and tolerably quiet flow control, valve TDTCV will be one of the "cavitation-control" valve configurations, sometimes referred to as "quiet" valves, that have been developed by valve manufacturers in recent years.

Figure 4-58 is a block diagram of the DSH operational control concept. As indicated, the pressure, temperature, and volumetric flowrate of the DSH steam influent are measured, presuming first-order time lags in the measurements. These measurements are inputs to a process control computer that computes the total enthalpy of the steam influent. A coolant water weight flowrate command signal $\dot{W}_3$, is proportional to the incoming enthalpy. The DSH controller, shown in Figure 4-57 with tag number TDC, is physically a portion of a larger assembly which is identified as the Thermal Storage Unit Controller.

The DSH outlet steam temperature is measured, again presuming a first-order time lag in measurement response. The measurement signal is compared with a reference signal, and an integrating amplifier with a transfer function N/s delivers a signal, $\dot{W}_4$, which attains any value that is required in eliminating steady-state temperature error.

The coolant water weight flowrate command signal, $\dot{W}_3$ plus $\dot{W}_4$, is compared with a signal that indicates a measured value of water flowrate. A flowrate error results in corrective action displacement of the water control valve, TDTCV. A typical second-order response transfer function is presumed for modulating throttle valve closed-loop control.

With this temperature control concept, the commanded coolant water flowrate is primarily a scheduled function of the computed incoming heat energy; with trimming corrections applied as required in eliminating steady-state error in DSH effluent temperature. Changes in influent enthalpy result in fast-response changes in coolant water flowrate in maintaining an approximately correct effluent temperature, followed by slower responding changes that integrate control error to a nominal zero.

Ullage Maintenance and Fluid Maintenance Unit Controls

The fluid maintenance unit and ullage maintenance unit are both ancillary to the system and there is negligible physical interaction between these systems and the TSU system provided that these ancillary systems perform their

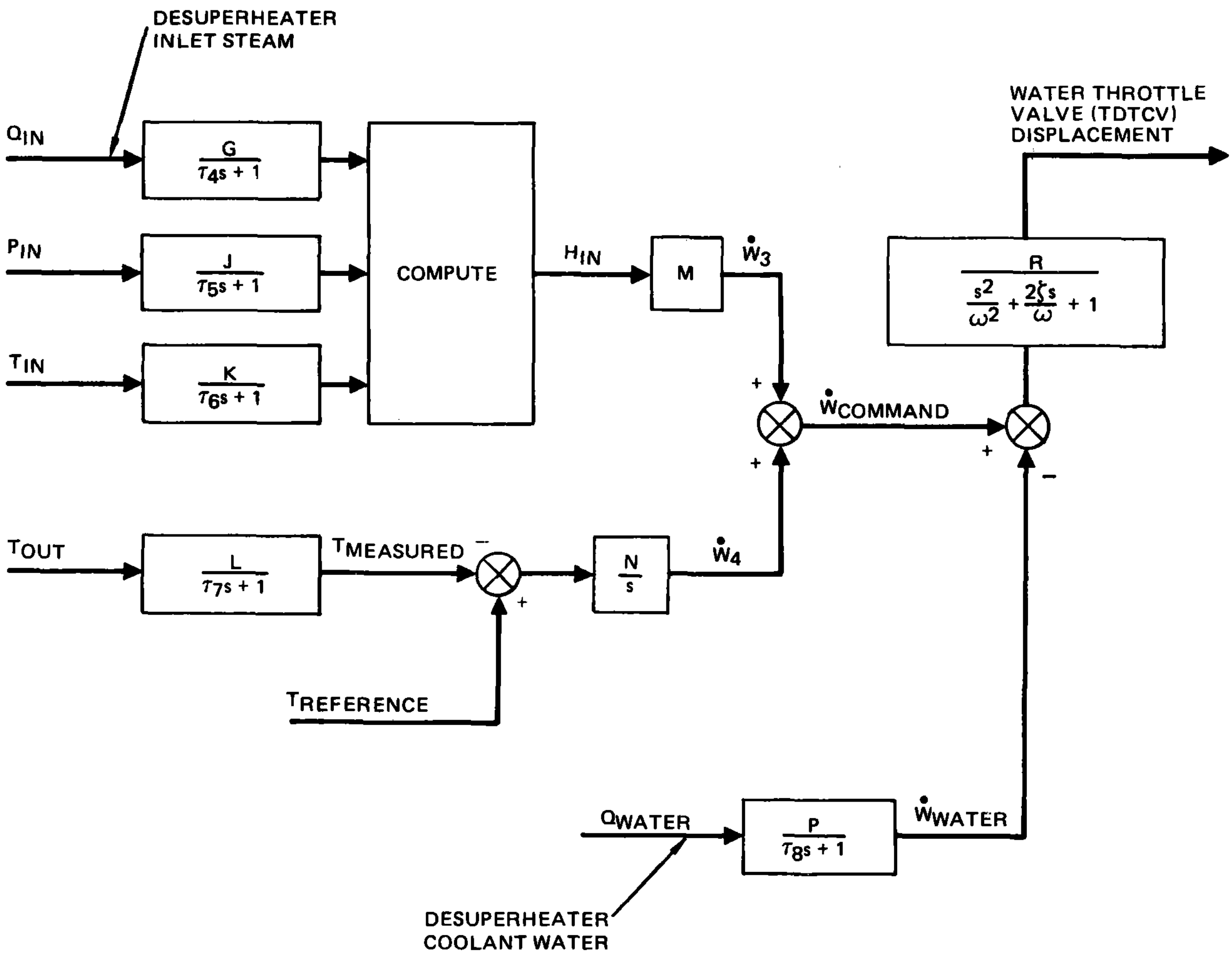


Figure 4-58. Block Diagram of Desuperheater Controls

functions within their design specifications. Consequently, discussion of these control systems are given in respective sections dealing with these functions, i.e., Ullage Maintenance, Section 4.3.2 and Fluid Maintenance, Section 4.3.3.

However, if these units do not maintain their functions within specified control limits, then they do interact with the TSU system. For example, if the filters, TFF-1 and TFF-2, which are part of the fluid maintenance system should become clogged, then these units will adversely affect the control functions of the TSU.

Monitoring of the filter condition and periodic checks to maintain them properly are discussed under Section 5.4.

Control Equipment

The TSU piping system includes manually operated shutoff valves that are used when part of the system is nonoperational and has been isolated for maintenance.

The TSU piping system also includes numerous solenoid-pilot-operated shutoff valves that are used for remote control of sequences of events, e.g., termination of TSU charging operations, initiation of TSU extraction operations. Additionally, heat-transfer fluid flowrate through each thermal storage heater and through each steam generator is under closed-loop control with a modulating butterfly for each unit. A modulating three-way valve controls heat-transfer fluid flow division through each steam superheater and its bypass. A modulating poppet valve controls water inflow to the DSH. Each of the modulating valves is pneumatically actuated in response to electrical position-command signals.

Heat-transfer fluid pumps in the TSU piping system are electric-motor powered in response to remotely controlled on-off signals and high-low speed selection signals. All parameter sensors, e.g. pressure transducers, thermocouples, fluid flow meters, liquid-level sensors, etc., deliver electrical signals.

The use of electrical signals for parameter sensing and for control permits use of versatile electrical/electronic instrumentation and control equipment.

4.4 OPERATIONAL CHARACTERISTICS

This section discusses the alternative modes of operation of the Pilot Plant Thermal Storage Subsystem (TSS), with subsections devoted to each of the major operational modes.

4.4.1 Startup Mode

There are three types of startup operation which may occur with the TSS: (1) initial startup of a plant subsystem after construction, (2) cold startup after a very lengthy shutdown period, and (3) "hot" startup, following previous operation within a few days, which is the startup mode encountered most of the time.

4.4.1.1 Initial Startup

The TSU tank is first filled with heat-transfer fluid from the bottom up, at ambient temperature, thus displacing the air in the tank upward and out an open vent line until the required volume of fluid is transferred into the system. A gaseous nitrogen supply derived from liquid nitrogen previously supplied to the system is readied for use. A flow of cold nitrogen gas at or near the maximum flow available from the unit is conducted into the ullage space above the heat-transfer fluid. This nitrogen gas is colder than the surrounding tank and fluid and will blanket the fluid and displace the air upward out of the vent, thus reducing the total amount of gaseous nitrogen which will be required (as compared to simple flushing) to reduce the oxygen content of the ullage space down to safe limits. The ullage gas compressor is now started and the compressed gas is stored in the ullage storage tank. Meanwhile, the flow of nitrogen gas is regulated to replace the gas being evacuated by the compressor. As long as the oxygen content of the gas is above acceptable limits, the compressed gas is vented to the atmosphere. This flushing operation is continued until the oxygen content of the gas in the ullage space has been brought down to acceptable limits. The anti-oxidant content of the heat-transfer fluid is sufficient to neutralize any remaining oxygen and that which is adsorbed onto the surface of the rock and sand bed. However, water is also adsorbed in the surface fissures of the rock and sand.

The heating of the fluid can now be initiated. This operation must proceed slowly, especially as the boiling temperature of water is approached, because this is the temperature range at which the greatest quantity of water will be released from the rock and sand fill. This water vapor is pulled from the ullage space by means of the ullage space compressor and condensed in the compressor intake manifold and collected in the attached sump. Should steam be released at rates higher than can be accommodated by the compressor, it will be vented to the atmosphere by the automatic pressure relief system. Close monitoring of the TSU pressure and the condensable liquid levels in the two tanks at the inlet and outlet of the ullage space compressor is required.

A check on the rate with which water is being expelled is obtained by noting the quantities of condensed water removed from the ullage gas storage tank and by the moisture monitoring instrument which has a sensor located in the compressor feed tank. This sensor is normally used as a leak detector should one of the heat exchangers become faulty. The heating rate will be closely controlled to avoid excess rate of steam production. Once all the water and air have been removed from the TSU, subsequent operation will consist entirely of venting and replacing the ullage gases resulting from the normal rise and fall of the fluid resulting from charging and extracting of heat.

4.4.1.2 Cold Startup

It is predicted that, except for the initial startup following construction, it is highly unlikely that a "true" cold startup will occur. This is because the heat leakage rate from the Pilot Plant TSU is on the order of 2.6% per day of the extractable heat and thus requires 112 days to cool down from 575° to 325° F and approximately 483 days to cool to 100°F (assuming an ambient temperature of 75°F).

The piping, heat exchangers, and pumps will cool down more rapidly than the TSU, and are expected to require a cold startup procedure after a few days of inoperation. The exact length of time after which a cold start procedure will be required can best be determined once experience has been gained with the Pilot Plant; however, it is estimated to be longer than 24 hr. One of the operational learnings gained from running the SRE is that gaseous degradation products tended to collect in the upper parts of the pump volute when the system was stopped for a number of days. This sometimes caused cavitation of the pumps when they were first restarted. The obvious solution, which will be used for the Pilot Plant, is to bleed the gases from these points of entrapment using small lines equipped with vent valves. The tendency for cavitation due to this trapped gas is greater for the hot start condition than for the cold start. However, if this bleed system is equipped with small solenoid valves which can be operated from the control room, the precaution can be taken easily of venting these spaces as a routine procedure before starting the pumps. The vented gases are conducted to the ullage maintenance system compressor intake manifold, where they are treated as if they had come from the ullage space in the TSU.

The charging pumps should be started in the slow-speed mode and the associated throttle valves set to a partly open position to limit the power required. The viscosity of the fluid increases greatly as the temperature drops. The initial fluid flowrates will be below normal since the fluid viscosity will be much higher (by a factor of 30) than for the design temperatures. The system will be started at an equivalent charging rate of 1/30 turndown ratio, and increased to higher rates. Once the pumps are started, thermal charging can begin and the temperature of the TSU and associated charging loop equipment will increase slowly. It is at this point that the valve which permits the fluid to bypass the TSU is used. With this valve open, the pumps and piping are brought up to temperature quickly and no attempt is made to heat the TSU. Once this fluid is up to temperature, the bypass valve is closed partly to force some fluid through the TSU. Thus, mixing takes place at the bottom of the TSU between the hot bypassed fluid and the cold fluid coming from the TSU bottom manifold. The ratio of these fluid flows is set, if possible, to allow the charging heat exchanger fluid inlet temperature to be at the design value of 218°C (425°F). It may not be possible to achieve balancing this until the TSU arrives at a higher temperature level.

It would be ideal to charge the TSU up to the design temperature of 302°C (575°F) directly since, once a full charge has been obtained, operation would be simplified. However, caution must be exercised as the TSU is brought up to temperature through the temperature at which water boils since any water which may have entered the system inadvertently during the cooled down period will come out of the fluid and vaporize vigorously at temperatures above 100°C (212°F).

A more cautious approach is to restrict the steam input to the charging heat exchanger and set the charging heat exchanger fluid outlet temperature to around 110°C (230°F) and charge the entire TSU up to this temperature using the bypass procedure described above. The outlet temperature is then reset upward to 218°C (425°F) and the charging is repeated until the entire TSU is at that temperature. This is also done at somewhat reduced input steam flow rates, and is then followed by a normal charging run to bring the TSU up to full temperature (with the bypass valve closed).

During these procedures, the tank ullage pressure should be watched and the water condensation in the compressor intake manifold in the UMU monitored, since water vapor may condense in these in relatively large quantities rather quickly.

Once the TSU has an average temperature of 177°C (350°F) or above, it is permissible to start the extraction pumps to slowly heat up the extraction side piping and heat exchangers, if desired. An extraction system preheating can be obtained easily by using only the small fluid pump which runs during the hot standby mode.

4.4.1.3 Hot Startup

The highest efficiency will be obtained if the thermal charge condition of the TSU is determined before startup as well as the anticipated future operating mode and heat load. Reference is then made to a chart as shown in Section 4.3.9.2 to determine whether two heat exchangers or one should be used. If the anticipated charging rate will be in the range from 1/4 to 2/3 of maximum flow, both heat exchangers should be run with both pumps running at slow speed (1,150 rpm). If the anticipated charging rate is to be below 1/4 of maximum, then only one pump and one heat exchanger are activated.

A check list is then used if startup is to be done manually. Automatic startup should be provided for but not attempted until manual control techniques have been perfected; the automatic starting sequence and timing can then be properly programmed into the control circuitry.

The pumps are activated in low-speed mode to reduce electric starting surges. Once the fluid is circulating, the steam supply valve is opened to allow steam to enter the charging heat exchangers.

The system can soon be switched to automatic mode. However, the only active pieces of machinery in the TSS are the fluid pumps. As long as precautions are taken not to damage them, no catastrophic failures will occur no matter in what sequence the system is started. The only serious precautions to be taken are in regard to the pressure level in the ullage space above the fluid in the TSU. However, this system must operate continually, even when the TSU is in a stopped or standby condition. Thus, it should not be necessary to change anything as regards the automatic controls of the Ullage Maintenance Unit during startup or shutdown or at any time, except that the UMU should be monitored, maintained, and operated properly at all times regardless of operating modes or changes in operating modes.

Even though the system is difficult to damage, the efficiency will, of course, be degraded if proper operational procedures are not followed. These procedures will be worked out during checkout runs and initial trial runs. The system also lends itself to operation at high turndown ratios, permitting experience to be gained by operating personnel at low heat rates where the possibilities of doing damage to the system are even further diminished.

The remaining precaution is in regard to overheating of the heat storage fluid. Overheating of the fluid causes degradation rates to increase exponentially with temperature. The critical point in the system for fluid overheating is at the fluid exit of the charging heat exchangers. An enunciator alarm is provided on each of these temperatures to give operating personnel warning of fluid overtemperature.

Experience gained from operation of the SRE has shown that vapors tend to collect at high points in the piping and in the top of the pump volute if the system is allowed to stand for a number of hours with hot fluid in the lines. This can cause pump cavitation upon restart. This condition is found to be more pronounced in hot restart than in cold restart modes. The cure is to provide for a venting system, as briefly described in the previous section.

4.4.2 Charging Mode

A schematic of the plant under normal charging operating conditions is shown in Figure 4-59. Steam from the receiver located on the tower feeds the turbine and charges the TSS. The bypass flow charging the TSS is first desuperheated to avoid overheating the thermal storage medium. It then passes through the thermal storage heater, and is eventually returned to the receiver. The thermal storage fluid is extracted from the bottom of the thermal storage tank and after being heated, is introduced through a manifold into the top of the TSU tank.

Operation with reduced solar power is indicated in Figure 4-60. When solar power available from the tower is reduced below the nominal value, the bypass flow that charges the TSS is reduced to maintain constant turbine inlet conditions. When this flow reaches zero, turbine extraction for feedwater heating is successively reduced starting with feedwater heater No. 3, until finally extraction is limited to the minimum required to provide deaeration in feedwater heater No. 2. Steam is supplied from the steam generator as required to maintain the temperature of the feedwater at the nominal value of 205°C (401°F) for return to the receiver.

The thermal storage system is designed to absorb a maximum of 30 MWt of energy down to the 1.5 MWt rate (a turndown ratio of 20 to 1). This energy enters the thermal storage system in the form of superheated steam and represents the independent variable while in the charging mode. The thermal storage system is designed so that it can accept any steam flow rate input within the rated range. The rate of change of steam flow input is limited to the ability of the DSH controller to follow this rate of change and to ensure that the temperature downstream of the desuperheater does not exceed or fall short of the design temperature of 343°C (650°F). The control system is designed to maintain a flow of heat-transfer fluid into the top of the thermal storage unit at a constant temperature independent of the thermal flow variations of steam at the input. This function will proceed smoothly and automatically with upsets caused only by the change in the steam flow rate and by the possible changes in the temperature of the thermal storage fluid entering the heat exchanger. These would originate in the lower temperature part of

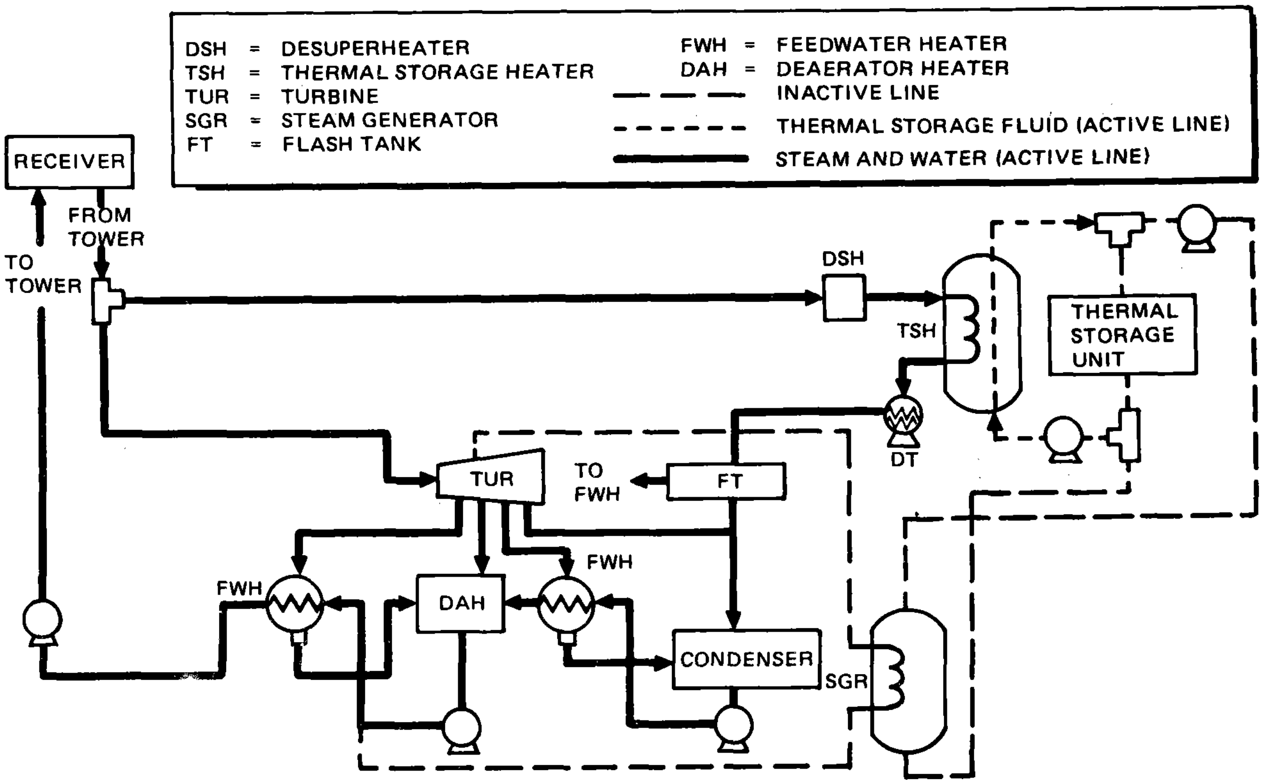


Figure 4-59. Plant Schematic for Normal Charging Operation

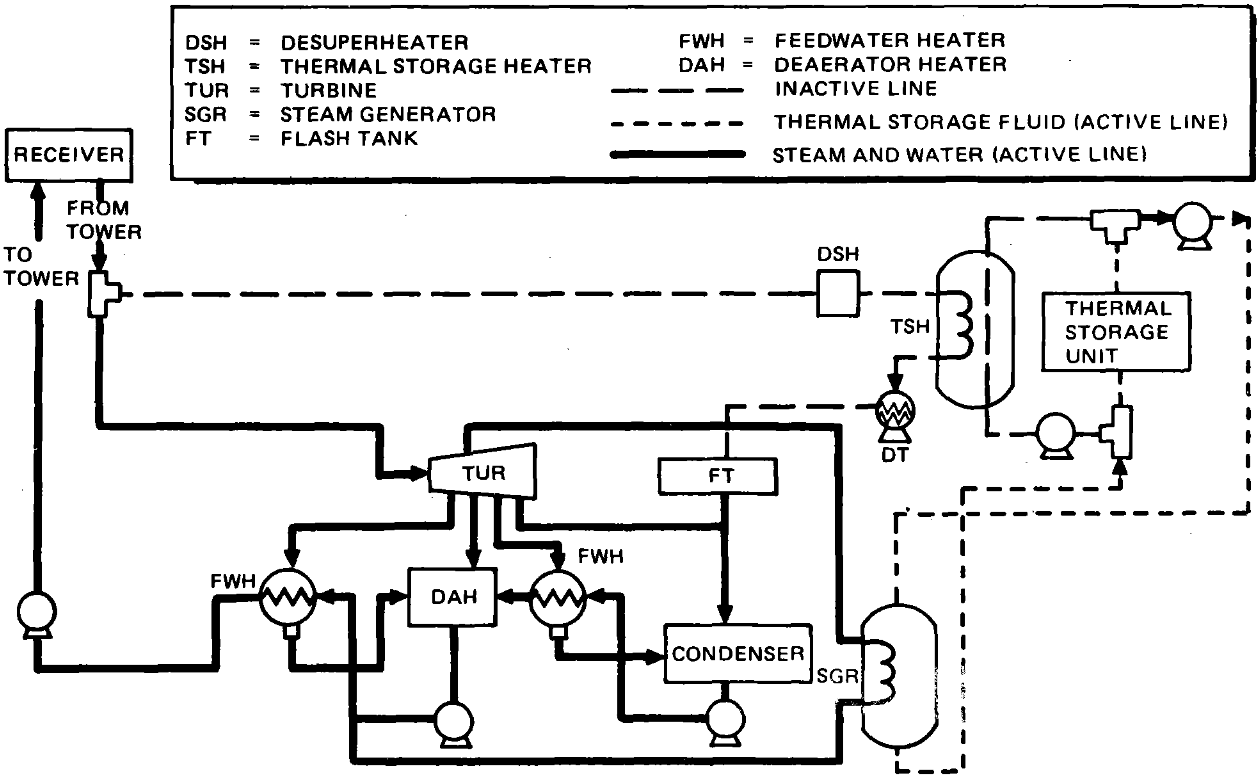


Figure 4-60. Plant Schematic for Low-Solar-Power Operation

the thermal storage unit caused by extraction of seal steam heat during the previous night's operating period.

As the thermal storage unit approaches its maximum charge condition the charging rate will become restricted, in that the fluid in the cooler end of the thermal storage unit will contain elements of fluid which are physically close to the thermocline and which will be above the design temperature of 219°C when they approach the thermal storage heater. In order to maintain a constant temperature output from the thermal storage heater, the charging controller will speed up the flow rate; however, this flow rate speed up is limited by the capacity of the pumps and the range of the control valve.

Another limitation will present itself if the cool fluid temperature and fluid flowrate rise too far. The temperature difference between the inlet fluid and the output water (condensed steam) at the cool end of the heat exchanger will not be great enough to sustain a sufficiently high heat-transfer rate. A complete condensation of the steam may not be possible with the area and ΔT available under these conditions, resulting in blow-through of the steam. As a result, the proper design solution to this is to match the excess flowrate available (above nominal maximum) to the small overdesign in the heat exchanger so that both reach their limits at nearly the same time.

Based on experience with the SRE, it is estimated that the temperature of the cool fluid will start to rise about 1/2 hr before the end point is reached and the cool fluid temperature will then have increased about 8.3°C (15°F) from 218° to 227°C (425° to 440°F).

When the TSU reaches the fully charged condition, a warning signal is sent to the master controller so that it can take action to reduce the amount of steam being conducted to the thermal storage system from the receiver.

4.4.3 Discharging Mode

Figure 4-61 shows the TSS operating loops in this mode. Operating personnel will prepare in advance to commence a discharge operation. The time required to build up a useful head of steam after the thermal discharge pumps have been started will be considered to estimate the exact time when the

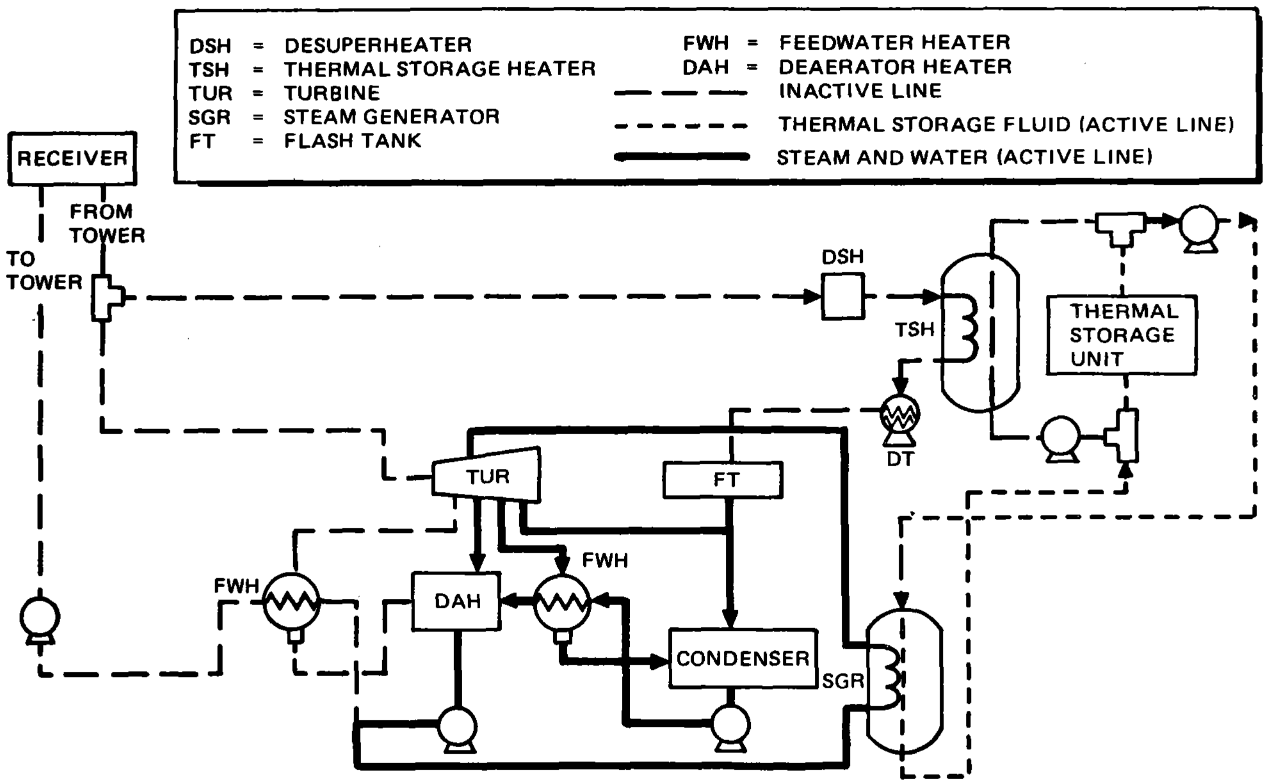


Figure 4-61. Plant Schematic for Discharge Only

discharge is to commence, so that the steam will be ready at the proper pressure and temperature when required by the turbine. Upon receiving a go-ahead signal from the central controller, the operator will start the thermal storage extraction pumps, TEP-1 and TEP-2 (equipment numbers are given in Figure 4-2). For purposes of clarity, the following discussion will follow the path of the fluid through only one set of heat exchangers. The description for the other set is identical.

The pumps pull fluid from the top manifold in the thermal storage unit through the shutoff hand valve at the exit and into the pump after passing through a strainer. At the outlet of the pump the pressure is 0.345 MPa (50 psig) as it passes through the check valve and then the flowmeter, and, finally, through the thermal storage superheater fluid inlet valve (TSFIV-1) which controls the fluid flowrate. The fluid flow is then split into two flows by a 3-way valve (TSTCV-1) which controls superheater outlet temperature. Part of the flow is conducted to the exit of the thermal storage superheater where the two flows recombine to enter the thermal storage boiler, TB-1. Downstream of the boiler, the fluid flows through the thermal storage preheater and back to the bottom of the thermal storage unit.

At the same time, water from the high-pressure condensate pumps is led through the thermal storage preheater water control valve into the preheater, where its temperature is brought up near to that of the temperature of the water in the boiler. The water then travels into the boiler and is evaporated by the heat from the fluid traveling in the opposite direction through the tubes in the boiler. The preheater water control valve is modulated by the boiler water-level measuring and control system to maintain a constant water level in the boiler. The steam generated then is conducted into the thermal storage superheater where its temperature is raised to 277°C (530°F). The superheated steam then exits this part of the system and is conducted to the low-pressure steam throttle on the turbine. The water temperature and pressure arriving at the preheater, and the steam pressure and temperature which are leaving the superheater, are both measured and recorded. The superheated steam flow rate is used by the extraction controller (EC), to modulate the flow of fluid into the superheater and into the boiler. In this way, the flow rate at the outlet of the superheater is maintained at a constant level set by the central controller. (A more detailed description of the automatic control system involved in this operation is given in Section 4.3.9.3.)

As the heat extraction proceeds, the charge monitoring system will indicate to the operator at regular intervals the charge state of the thermal storage unit. This indication will be valuable from an operational standpoint in that the required changes in operating mode can be anticipated.

As the thermocline proceeds up the thermal storage unit, the average temperature of the fluid in the system and in the thermal storage unit will decrease and the fluid will contract. To fill the void, gas is conducted into the ullage space by the UMU in such a way as to maintain a positive pressure in the ullage space between 7.6 and 12.7 cm (3 and 5 in.) of water column. An alarm will be given automatically if the ullage pressure moves outside this operating range.

As the TSU approaches the total discharge condition, the temperature of the fluid will start to drop slowly and the control valve will open further, allowing more fluid to flow to the heat exchangers to compensate for the fluid temperature drop. Increasing the pumping rate of the fluid to compensate for the dropping temperature will be adequate until a point is reached — approximately 17°C (30°F) below nominal outlet temperature — where the fluid temperature is no longer high enough to provide the heat exchanger with sufficient ΔT to achieve the required heat-transfer and to keep the steam outlet temperature high enough. Hence, the design matches the excess pumping capacity (approximately 20%) with the excess margin in ΔT provided in the extraction heat exchanger design.

4.4.4 Intermittent Cloud Mode

When substantial variations in insolation are encountered, due to intermittent clouds, the operating mode shown in Figure 4-62 is employed. In this mode the TSS will be charged at the maximum rate, as during normal charging operation. Operation of the generating plant can be maintained at approximately 70% of rated capacity, using some of the energy stored within the

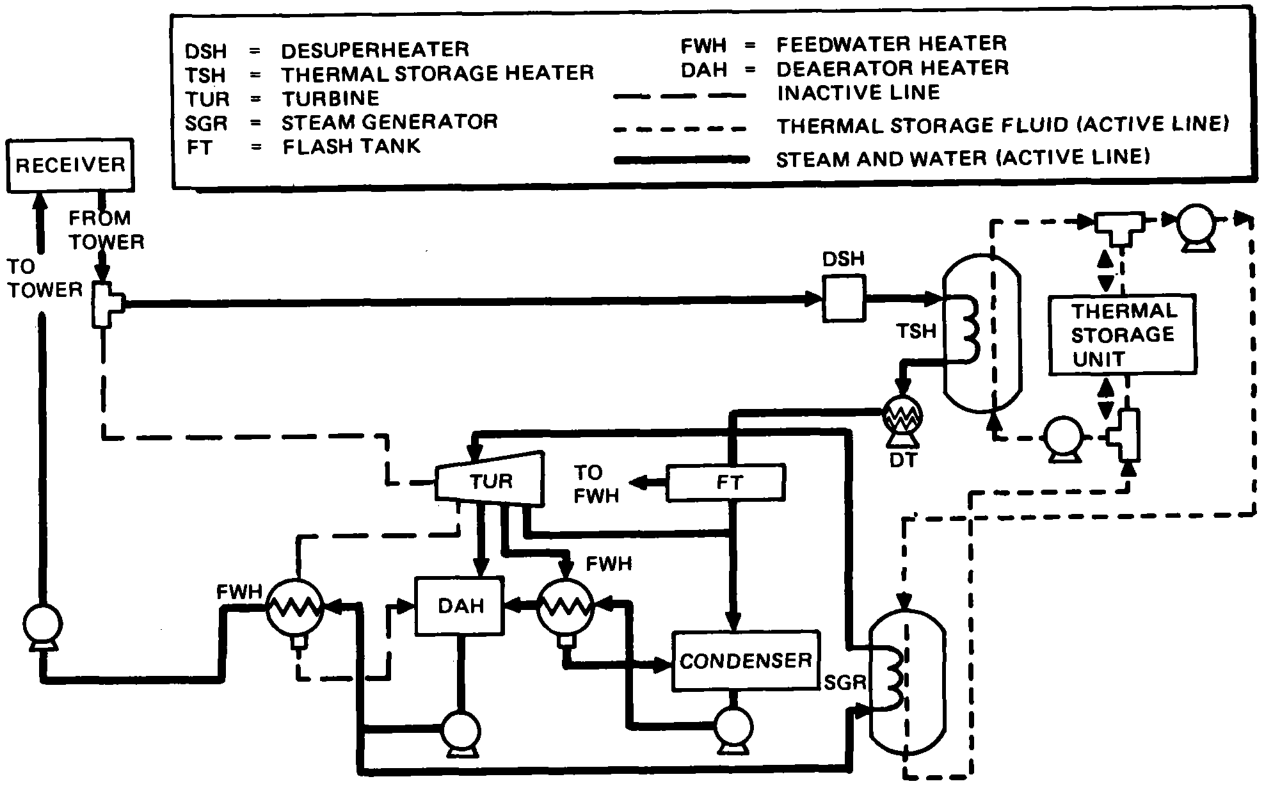


Figure 4-62. Plant Schematic for Operation with Intermittent Clouds

TSS. This operating mode requires a turbine capable of accepting two admission conditions.

When the collector field experiences intermittent cloud conditions, the turbine is switched from the receiver output to the TSU output. The operations which take place in the TSS are then a combination of those described in the previous sections under Charging Mode and Discharging Mode. The function of the components in the charging side of the TSS are completely independent of those in the extraction side of the system. They are, of course, connected at the thermal storage unit, but the TSU functions to isolate the two systems from each other, since the TSU has such a high capacity compared to the flowrates in the two sides of the system. To illustrate this, it can be imagined that the system is initially in the charging mode, receiving full power from the receiver and all of the energy is being stored in the TSU. The extraction side of the system then can be started at any time and energy can be extracted from the TSU. However, upon closer analysis it can be seen that the heated fluid coming from the thermal storage heaters can pass directly into the extraction heat exchangers and the cooled fluid returns directly to the charging heat exchangers without necessarily going into the TSU. The TSU takes automatically any differential which happens to exist in these two sets of flows. It acts as a buffer between a highly varying thermal input from a mirror field subjected to intermittent cloud conditions, and stands between this varying thermal source and the requirements of the turbine which are essentially a constant input of high-pressure, high-temperature steam. Thus, when the sun illuminates the collector field, the thermal storage unit itself is relatively idle since most of the energy is passed directly past it. When clouds obscure the sun, the TSU supplies the total energy to keep the turbine running at derated conditions. If the insolation is high most of the time, the thermal storage unit will slowly accumulate any excess energy not used by the turbine. If the insolation is lower, the TSU will be discharged slowly. Since the charge condition can be monitored easily, an impending, totally discharged condition can be anticipated easily and the cognizant personnel warned that the system may have to be brought down to a standby power condition.

The intermittent cloud condition also will cycle the charging side of the thermal storage system through its complete range of turndown ratios. It is conceivable that the desuperheater and thermal storage heaters and their associated pumps could be driven alternately from full throughput to 1/20 of full throughput and to be subjected to this treatment alternately and fairly rapidly throughout the day. This treatment may ultimately be detrimental in that it may show up in the form of thermal stress fatigue in the heat exchangers. On the other hand, this condition has been recognized from the beginning and the heat exchangers will be designed to withstand this type of thermal cycling. The severe excursions are experienced only on the charging side of the system. The discharge side is affected only by changes in the turbine load requirements, which are usually slow moving and of limited turndown ratio.

4.4.5 Shutdown/Standby Modes

The common definition of TSU shutdown will be one in which the thermal input and the thermal output are both reduced to zero. Since the unit is meant to store energy it can never really be shut down in the normal sense of the word. For it to cool off in a fully charged condition would take many days unless energy were actively extracted from it. The shutdown procedures are thus limited actually to mainly decreasing the charging rate, from whatever the value is when the command is received to shutdown, to zero charging rate. If the TSU has been discharging when the command to shutdown comes, attention will be focused on bringing the discharge rate from whatever it is down to zero. On rare occasions the command to shutdown might be received when the system is both receiving heat through its charging heat exchangers and supplying heat through the extraction heat exchangers. In this case, both sets of heat exchangers could be shut down simultaneously; if it were advantageous and allowable to shut one system down before the other this could be accomplished without any problems since the thermal storage unit would either absorb or supply the extra energy as required during the delay. Experience with the SRE subsystem indicates that the flexibility is high.

4.4.5.1 Shutdown Following a Charging Operation

Immediately after the decision to shut down is received, the steam from the receiver to the DSH will be decreased and ramped down to zero. The desuperheater controller and the charging controller will respond to this decrease in receiver steam supply by throttling down the water to the DSH and the fluid flow through the thermal storage heaters, with the objective of maintaining the temperature of the thermal storage fluid at the output of the thermal storage heaters at the design value of 304°C (580°F). When the turndown ratio of approximately 20 to 1 is reached, the steam supply from the receiver will be closed completely, the fluid charging pumps will be stopped, and the entire system will come to rest.

The only systems which will remain active at all times are the ullage and, at various times, the fluid maintenance system. The ullage maintenance system must remain active constantly since the pressure in the ullage space must be kept within close limits, slightly above the outside atmospheric pressure. Even though the thermal storage unit is in a standby condition or idling mode, the change in atmospheric pressure will affect the relative internal and external pressures sufficiently to cause the UMU to correct the internal pressures to maintain a constant pressure differential between the inside and outside of the tank at the top. In addition, as heat is slowly lost from the piping and from the sides of the thermal storage tank, the temperatures in the ullage space will change slightly and will require a slight influx of gas to make up for the contraction due to the lowering of the ullage gas temperature.

4.4.5.2 Shutdown Following an Extraction

If the TSS is in the extraction mode when the command for shutdown is received, it is possible to stop the flow of fluid almost immediately by stopping the power to the pumps. A small amount of steam will still be generated due to the heat given off by the fluid, which has been trapped in the tubes within the superheater, the boiler, and the preheater.

4.4.5.3 Hot Standby Mode

The hot standby mode is one of three standby modes which can be defined; the others are idle mode and nighttime mode. The hot standby mode is one in which the charging and/or extraction heat exchangers are maintained in a hot condition so that they can be brought on-line immediately to accept steam from the receiver or furnish steam to the turbine. Energy from the thermal storage unit is used to maintain the extraction heat exchangers in the hot standby mode. The energy required is estimated to be 0.015 to 0.02 MWt. This heat is furnished to the heat exchangers by a small pump which extracts fluid from either the top or the intermediate seal steam manifold of the thermal storage unit and pumps it through the discharge heat exchangers and then back into the bottom of the thermal storage unit. An analogous scheme for maintaining the thermal storage heaters in the charging side of the thermal storage system at a high temperature is not required since a small bleed flow can be taken from the receiver downcomer for this purpose, if desired.

4.4.5.4 Idle Standby Mode

The standby idle mode is the condition of the thermal storage unit if for any reason the entire plant should be shut down for an extended period. Some provision must be made in such a case to provide for enough ullage maintenance gas in the form of pressurized nitrogen to maintain the pressure in the ullage space within the prescribed limits. Should the TSU be required to be in the idle mode for days or weeks, the temperature within the subsystem will drop slowly as a result of heat loss through the insulation.

4.4.5.5 Nighttime Standby Mode

This mode is used at night after the main turbine has been placed in a daily standby mode, and a supply of seal steam is required to maintain a positive pressure in the turbine and to main the turbine seals at an elevated temperature. This is a substantial heat rate, 0.326 MWt for the Pilot Plant. Since a steam supply temperature of only 135°C (275°F) is required, it is not necessary to use high-grade heat for this purpose. Accordingly, heat-transfer fluid from the cool side of the TSU is used. Fluid from a special manifold a few feet above the main bottom manifold is pumped (by a special auxiliary pump, TAFP) from the TSU and into one of the steam generators (boiler and superheater section), after which it is returned to the bottom of the TSU.

The present design specifies that the TSU will be operated in extraction mode only when the average fluid temperature in the TSU is at least 177°C (350°F). This is due to the fact that cooling causes contraction of the fluid, decreasing the fluid level so that the holes in the upper manifold will be uncovered if the mean temperature falls below 177°C. Below this temperature, a charging mode will be made first to bring the average temperature of the TSU above 177°C.

The thermal charge condition and average temperature of the TSU are provided to the operator so that this limiting condition can be anticipated and a warning signal provided when the average temperature approaches 177°C.

4.4.6 Emergency Modes

Emergency conditions in subsystems external to the TSS may impinge on the TSS, e.g., causing a sudden stop of thermal power to the TSS, or requiring that the TSS shut down rapidly. In general, the TSS lends itself very well to dealing with these sudden changes of steam input and/or power output requirements. No unusual problems are anticipated in meeting these requirements because all the components are passive except for the pumps.

Similarly, should an emergency occur in which the electrical power to the TSS should be severed, shutting down all pumps and allowing all valves to return to their "normal" positions, then it has been determined that no dangerous conditions will result.

If the TSS is physically damaged in such a way as to allow fluid to spill out onto the surrounding ground, a series of dikes and drains have been provided to contain the fluid. The main danger is one of fire and a firex system has been provided as well as an underground drainage sump into which fluid will flow naturally by gravity. If this fluid should be burning, it will be snuffed out as it enters the underground sump area since little oxygen is available.

The main reason that the TSS is so immune to adverse treatment is that no source of energy other than the superheated steam is available to the system, except for the electrical power to run the pumps and compressors. The maximum temperature of any component in the entire system is limited to that of the entrance temperature of the superheated steam. This temperature only exists in a few components upstream of the DSH. For example, the thermal charging heat exchangers have metal components which are never above 343°C (650°F), a temperature which the fluid can easily withstand for short time periods under emergency conditions. Thus, the lower temperature fluid will easily absorb the heat stored in the higher-temperature components, if all steam flow and fluid flow should suddenly come to a standstill.

Should electrical power be cut off from the unit, then the entire system will be allowed to cool down gradually. The gases in the ullage space will contract and gaseous nitrogen will be required to fill this space in order to avoid collapse of the tank (due to a partial vacuum which would otherwise be created in the ullage space). This nitrogen gas supply will operate satisfactorily without the need of any kind of electrical power, since the nitrogen is stored under high pressure in a bottlebank and the gas is allowed to enter the ullage space under the control of a simple vacuum breaker valve.

Emergency conditions could be caused within the TSS, itself, if there were a massive failure of critical components or assemblies, e.g., a major piping or vessel failure, releasing hot fluid, or over- or underpressurization of the TSU. Such failures are extremely unlikely, and careful design and operational provisions have been made to effectively eliminate such events. However, if such a failure should occur, the subsystem has also been designed to contain and minimize the effects of such a failure.

4.4.7 Transitions between Operating Modes

Following is a discussion regarding the time required to switch the TSS between allowable combinations of the operating modes numbered and defined in Table 4-29.

Table 4-29

OPERATING MODE DEFINITIONS

		EGS Turbine		TSS		
Mode	REC "On"	Steam From REC	Steam From TSS	Charge	Discharge	Hold
	X	X				X
2	X	X		X		
3	X	X	X		X	
4	X		X	X	X	
5			X		X	
6	X			X		

REC - Receiver
EGS - Electrical Generating System
TSS - Thermal Storage System

The time required to essentially come to equilibrium in the new mode from the old mode is viewed from two standpoints: (1) time delays which impinge upon operating conditions outside the TSS, and (2) the effect of the time delay on the TSS itself.

Figure 4-63 shows pictorially the time constant nomenclature in which each subscript letter indicates the unit involved and whether the time constant is involved with increasing energy throughput or decreasing energy throughput. For example, t_{ci} is the time constant (t) associated with the charging (C) heat exchanger while undergoing a transition to an increase (i) in energy throughput.

The following mode transition discussion can best be presented by following the mode to mode transitions which will occur in a normal day's solar plant operation.

4.4.7.1 Mode 6 (TSS Charging Only) to Mode 2 (TSS Charging Plus Receiver Flow to Turbine)

Due to the low solar power incident on the receiver during the early morning hours, the most efficient capture of energy will be achieved by initially establishing Mode 6 and charging the TSS with derated steam until sufficient insolation exists to produce rated steam for turbine operation.

Once derated steam output conditions have been achieved into the receiver flash tank system, steam will be directed down the downcomer and toward thermal storage. Low flow will be simultaneously established in the thermal storage charging loop. When normal heat-transfer fluid temperature has been established leaving the thermal storage heater(s), charging will begin into the top of the TSU and will continue until sufficient insolation exists to permit receiver transition to rated steam output conditions.

Transition time to steady-state Mode 6 will be governed by the large relative mass of the downcomer and interconnecting piping and their temperatures and not by the thermal storage subsystem components.

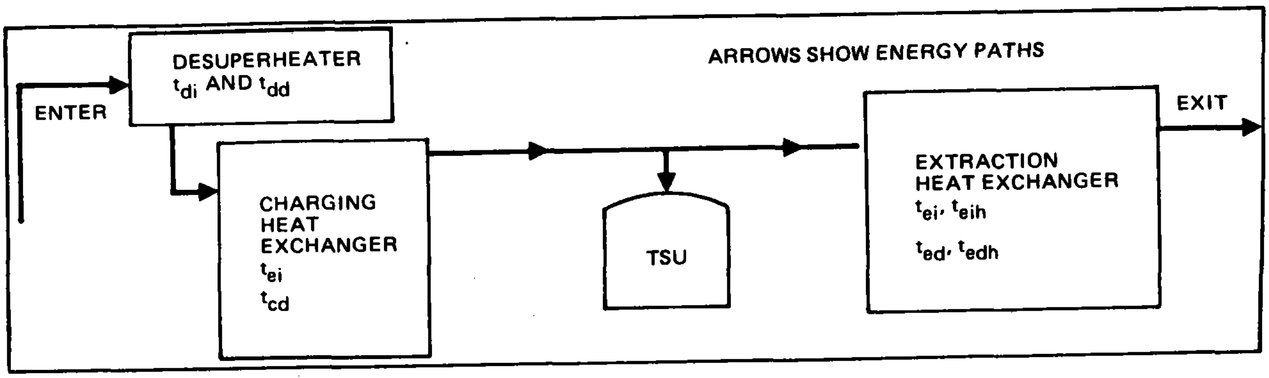


Figure 4-63. Principal Time Constants Involved in Transition Mode Study

Once rated steam conditions out of the receiver are achieved, the turbine throttle steam drain valve will be opened and steam flow will begin toward the turbine. When the proper superheat conditions are achieved at the turbine, the throttle stop valve will open and turbine spinup will begin. Here again, the interconnect piping and the turbine spinup limitations will govern the transition to Mode 2, not the TSS. Activation of the DSH during transition of the receiver from derated to rated steam conditions occurs essentially instantaneously.

4.4.7.2 Transition from Mode 2 to Mode 1 (Turbine Operation Only: TSS Hold)

In this transition the system switches from the condition where steam from the receiver is flowing both to the turbine and to the thermal storage unit to a condition in which the thermal storage system is in a hold mode. The question to be answered is how much time delay is there in re-establishing the new condition.

The steam flow to thermal storage may be cut off immediately. The charging fluid flow controller immediately begins to throttle the heat-transfer fluid flow through the charging heat exchanger so as to maintain the exit temperature of the fluid at the design value. As the heat exchanger cools, fluid flow is throttled until it reaches a value of 1/20 of the maximum flow, at which point the thermal charging loop pumps are shut down completely and the flow is stopped, completing the transition to the hold mode. If desired, a small steam bleed flow might be maintained toward the TSS to maintain the thermal storage heaters in a hot standby condition.

It is estimated that the transition to Mode 1 will require a time period approximately equal to the sum of the time constants for the DSH and the charging heat exchanger; i. e., $t_{dd} + t_{cd}$. However, this decreasing energy input influences only the TSS desuperheater and charging heat exchangers. Thus the plant outside the TSS is not influenced by these time constants in this case and only is aware of the rapidly decreasing steam flow to the TSS in which the rate of decrease is only limited by the steam shutoff valve cutting off the flow, i. e., a few seconds. It is apparent that the study of this particular transfer as regards time constants is almost trivial since the effects of changing the valves make little difference in plant operating efficiency.

4.4.7.3 Transition From Mode 1 to Mode 2

As receiver power increases during the day, the condition may be reached where the receiver is producing more steam than the turbine requires for maximum electric output. Since the buildup of excess steam output is predictable and relatively slow, the limiting transition time factor will be the rate of insolation increase rather than the thermal storage subsystem.

4.4.7.4 Transition From Mode 1 to Mode 3 (Low-Insolation Turbine Operation)

As the maximum insolation period of the day passes, the plant will transition back from Mode 2 to Mode 1 in the manner previously discussed. As the day continues to wane, admission steam flow from thermal storage is required to supplement receiver flow to maintain rated turbine output as long as possible. Since the thermal storage susbsystem is designed to maintain the steam generators in a hot standby condition, the maximum time lag would be due to the mass and temperature state of the delivery piping between the steam generator(s) and the turbine. Since the normal diurnal variation of insolation is known, under nominal conditions the line warmup could be scheduled to eliminate any delay.

4.4.7.5 Mode 3 to Mode 4 (Turbine Operation From TSS Only; TSS Charging)

With the plant in Mode 3, as the insolation continues to decrease, the thermal storage subsystem continually assumes a larger share of the turbine load. When the point is reached when the receiver can no longer maintain rated conditions, TSS steam generator flow to the turbine admission port is at its maximum value. The thermal storage steam inlet valve is modulated open to gradually divert flow away from the turbine throttle valve and into the thermal storage heater(s). The turbine throttle stop valve is then closed and the turbine operates on TSS steam alone. The receiver is then commanded to produce derated steam conditions. Here again the limiting time is the slow transition time of the insolation down to an inadequate level to maintain rated receiver steam output.

4.4.7.6 Mode 1 to Mode 4 (Intermittent Clouds)

If during the day intermittent clouds conditions develop, the resulting rapid changes in insolation onto the receiver will cause fluctuations in receiver output conditions too rapid for the turbine to accept. It is planned that a weather monitoring system will provide advance warning of developing cloud conditions and will permit time to initiate flow from the thermal storage steam generators as in the transition from Modes 1 to 3 and to divert full receiver flow from the turbine to thermal storage as in the transition from Modes 3 to 4, as previously discussed.

4.4.7.7 Mode 4 to Mode 5 (TSS Discharge Only)
As the insolation declines below the point where even derated conditions can no longer be maintained out of the receiver, charging flow to the TSS is also reduced and then stopped as previously discussed in the transition from Mode 2 to Mode 1. The transition may be essentially instantaneous if desired.

4.4.7.8 Emergency Transition, Mode 1 to Mode 6
The most difficult and time-critical transition will occur if a turbine or other electrical power subsystem problem occurs while the plant is in Mode 1 with all steam flow to the turbine and none to the TSS. Although the thermal storage charging system could be maintained in a hot standby condition to be best prepared for this eventuality, the ability of the TSS to pick up the full receiver load quickly enough the avoid some loss of energy will depend entirely on the type of problem and the advance warning time given by the problem. Although under most conditions the few minute warning time required will be available, under immediate trip-offs of the turbine, either short steam dump periods or trip-off of the receiver will also be required.

4.4.7.9 End of TSS Charging and Discharging Transition
Following is a discussion regarding the practical limitations on operation of the thermal storage system near the end of charging on those infrequent occasions when the TSU is fully charged. There is a slight limitation on charging rate when the thermal storage unit approaches a fully charged condition and the exit fluid temperature begins to rise above its normal value.

If the TSS is receiving a full charging rate of thermal energy when the end condition is approached, then the following conditions should take place. The temperature monitoring system on the TSU plays an important role in this particular transition, since it permits a fairly accurate determination of the position of the thermocline in the TSU. It will be possible for either the computer, or the operator, or both to anticipate the time of reaching the full charged condition of the TSS. When the thermocline approaches the bottom of the TSU, the control system would call for a decrease in the steam charging rate. Provided there was no other course of action (e.g., using extra steam in the turbine) some spillage of solar energy would be required.

The master control can be programmed to include the various time constants involved so that plenty of warning time is given before this condition is reached. Therefore, no problems are anticipated, provided that these monitoring systems are functioning properly and that personnel take appropriate action. The required time constants to deal with this problem are on the order of tens of minutes, while the control time constant in the TSS with the greatest lag is on the order of tens of seconds. It can be seen that the TSS control system time constant will be adequate.

Also, when the thermal storage unit is being discharged and it nears its completely discharged condition, there will be limitations on the rate with which heat can be discharged near the empty condition. Again, the thermal storage unit's charge condition will be monitored by Master Control, and the approach of this condition will be shown by a warning light. Steps can be taken to avoid an unexpected power reduction, and transition to other operating modes can be made smoothly in view of the fact that enough time is given for these new modes. Thus, if it is known that the end of the charge is approaching, the electrical power dispatcher can be notified that he will be somewhat limited in the load commands that he is permitted to give the system during this final period.

In view of these operating requirements, the number of thermocouples per unit of height along the thermal storage unit is increased near the top manifold and near the bottom manifold so that greater accuracy can be obtained in predicting the time when corrective actions need to be taken in regard to approaching the fully discharged (or charged) condition of the TSU. As a backup to the master control monitoring and calculating capabilities there will be an additional display on the control panel for the TSS which will give an indication to the operator that a fully charged or fully discharged condition is approaching. In this way, any problems which may arise due to malfunction of the computer as regards this mode of operation can be avoided.

It should also be remembered that the charging rate of the thermal storage unit and the discharging rate are completely independent. Any difference between the two is automatically either absorbed or made up by the TSU. Because of this, the time constants associated with the charging cycle do not influence the time response behavior when in the discharging mode.

4.4.7.10 Operational Summary

From the foregoing, it can be seen that the thermal storage design provides a flexible and responsive subsystem and under normal operating conditions does not create the limiting time lags in plant operating mode transitions.

4.4.8 Thermal Storage Availability

The availability analysis of the thermal storage system is shown in Table 4-30. The configuration of this systemis shown in Figure 4-64.

It was assumed that the required operating time of the thermal storage charging (input) circuit is 8 hr per day and 330 days per year or 2,640 hr. The duty cycle for the discharge (outlet) side is 2 hr per day or 990 hr per year. The TSU and its associated components must operate during both of these operations and also during the time that steam is being provide for feedwater blankets and for the turbine seals; therefore, it is assumed that these components have a 24-hr/per day duty cycle or 7,920 hr/yr.

The major failure item in the thermal storage subsystem are the pumps, where about one failure per year is expected. The heat exchangers (TH, TS, TB, and TP) require periodic tube cleaning and it is estimated that this will require one week every 18 months or about 112 hr/yr. However, this will be performed simultaneously and at the same time that the tubes of the receiver and the feedwater heaters are cleaned and the preventive maintenance is performed on the turbine and the generator.

It is assumed that failures of the sensors and controllers will not affect system unavailability in accordance with current commercial power plant experience. The thermal storage unit has dual input and dual output paths. Therefore, a failure in one path will not cause a system shutdown but will only cause a reduction in charging or discharging to 50% of the rated value.

The results of the thermal storage analysis gives 5.16 unavailable hours in the portion of the system with 2,640 operating hours, 16.80 unavailable hours in the 7,920 operating hour portion and 1.40 unavailable hours in the 990-hour portion. This gives subsystem unavailable percentages of 0.195, 0.212, and 0.141 % or a total of 0.549% which meets its availability goal of 99.45%. The planned outage is 112 hr in the 7,920-hr portion or 1.414%.

Table 4-30 (Page 1 of 3)

AVAILABILITY ANALYSIS- PILOT PLANT THERMAL STORAGE

Item No.	Component	Operating Hr/Yr	Mean Time Before Failures (hr)	Failures (Yr)	Mean Time To Repair (Hr)	Component Forced Outage (Hr/Yr)	Component Planned Outage (Hr/Yr)	System Unavailability (Hr/Yr)	Comments
TD	Desuperheater	2,640	31,250	0.17	14.0	2.4	37	2.4	
TDTC	Controller	2,640	27,400	0.10	2.0	0.20	0	0	
TDSFR	Flow Meter	2,640	83,000	0.03	4.5	0.14	0	0	
TDTCV	Control Valve	2,640	23,800	0.11	4.0	0.44	0	0.44	
TDWFR	Flow Meter	2,640	83,000	0.03	4.5	0.14	0	0	
TDWCK	Check Valve	2,640	250,000	0.01	4.0	0.04	0	0.04	
TDSIT	Temp Sensor	2,640	1,000,000	0.003	2.5	0.01	0	0	
TDSIP	Pressure Sensor	2,640	1,000,000	0.003	2.5	0.01	0	0	
TH-1,2	Thermal Storage Heaters	2,640	31,000	0.17	14.0	2.4	112	0.6	
CC	Controller	2,640	27,400	0.1	1.5	0.15	0	0	
THFFR-1,2	Flow Meters	2,640	83,000	0.06	4.5	0.27	0	0	
TCP-1,2	Pumps	2,640	14,000	0.38	4.5	1.71	0	0.43	
THWCK-1,2	Check Valves	2,640	250,000	0.02	4.0	0.08	0	0.02	
THSIV-1,2	Control Valves	2,640	23,800	0.22	4.5	1.0	0	0.25	
THFIV-1,2	Control Valves	2,640	23,800	0.22	4.5	1.0	0	0.25	
THSFR-1,2	Flow Meter	2,640	83,000	0.06	4.5	0.27	0	0.065	
TDSOT-1	Temp Sensor	2,640	1,000,000	0.003	2.5	0.01	0	0	

Table 4-30 (Page 2 of 3)

AVAILABILITY ANALYSIS-PILOT PLANT THERMAL STORAGE

Item No.	Component	Operating Hr/Yr	Mean Time Before Failures (Hr)	Failures (Yr)	Mean Time To Repair (Hr)	Component Forced Outage (Hr/Yr)	Component Planned Outage (Hr/Yr)	System Unavailability (Hr/Yr)	Comments
THFOT-1,2	Temp Sensors	2,640	1,000,000	0.006	2.5	0.02	0	0	
TDSOP	Press. Sensors	2,640	1,000,000	0.006	2.5	0.01	0	0	
TDSOT-2	Temp Sensors	2,640	1,000,000	0.003	2.5	0.01	0	0	
TFF-1,2	Filter	2,640	125,000	0.04	4.0	0.08	0	0.02	
TFFDP-1,2	Delta P Sensor	2,640	1,000,000	0.006	2.5	0.02	0	0	
UMU	Ullage Monitor	7,920	27,400	0.29	2.0	0.58	0	0	
TUFL-1,2	Level Sensors	7,920	1,000,000	0.02	2.5	0.05	0	0	
TUFT-1,9	Temp Sensors	7,920	1,000,000	0.07	2.5	0.17	0	0	
TUET-1	Temp Sensor	7,920	1,000,000	0.01	2.5	0.03	0	0	
TUEV	Control Valve	7,920	23,800	0.33	4.5	1.5	0	0	
TAFV	3-Way Valve	7,290	23,800	0.01	4.0	0.04	0	0.04	
TAFFR	Flow Meter	7,290	83,000	0.10	4.5	0.45	0	0	
TAFP	Pump	7,920	14,000	0.57	4.5	2.6	0	2.6	
TSTCV	3-Way Valve	7,920	23,800	0.01	4.0	0.04	0	0.04	
TAWOC-1,2	Check Valve	7,920	250,000	0.06	4.0	0.24	0	0.24	
TEP-1,2	Pump Valve	990	14,000	0.14	4.5	0.63	0	0.16	
EC	Outlet Controller	7,920	27,000	0.29	2.0	0.58	0	0	

Table 4-30 (Page 3 of 3)

AVAILABILITY ANALYSIS- PILOT PLANT THERMAL STORAGE

Item No.	Component	Operating Hr/Yr	Mean Time Before Failures (Hr)	Failures (Yr)	Mean Time To Repair (Hr)	Component Forced Outage (Hr/Yr)	Component Planned Outage (Hr/Yr)	System Unavailability (Hr/Yr)	Comments
TUBV	Bypass Valve	7,920	23,800	0.33	4.5	1.5	0	1.5	
TEPCK-1,2	Check Valves	990	250,000	0.01	4.0	0.04	0	0.01	
TSFBV	Shutoff Valve	990	23,800	0.04	4.0	0.17	0	0.17	
TSFIV-1,2	Control Valves	990	23,800	0.08	4.5	0.36	0	0.09	
TSTCV-1,2	3-Way Valves	990	23,800	0.08	4.0	0.32	0	0.08	
TSSFR	Flow Meters	990	83,000	0.02	4.5	0.09	0	0	
TSWCV-1,2	Control Valves	7,920	23,800	0.67	4.5	3.02	0	0.76	
TS-1,2	Superheaters	7,920	91,000	0.17	82.0	14.3	112	3.58	
TB-1,2	Boilers	7,920	91,000	0.17	82.0	14.3	112	3.58	
TP-1,2	Preheaters	7,920	91,000	0.17	82.0	14.3	112	3.58	
TBWL-1,2	Level Monitors	7,920	1,000,000	0.02	2.0	0.05	0	0	
TSFFR-1,2	Flow Meters	990	83,000	0.02	4.5	0.09	0	0	
TSWFR-1,2	Flow Meters	7,920	83,000	0.19	4.5	0.86	0	0	
TSSOT-1,2	Temp Sensors	7,920	1,000,000	0.02	2.5	0.05	0	0	
TSSOP-1,2	Pressure Sensors	7,920	1,000,000	0.02	2.5	0.05	0	0	
TEPOP-1,2	Pressure Sensors	990	1,000,000	0.002	2.5	0.005	0	0	
TU	Thermal Storage Unit	7,920	1,000,000	0.01	82.0	0.82	112	0.82	

THERMAL STORAGE SUBSYSTEM (TSS)

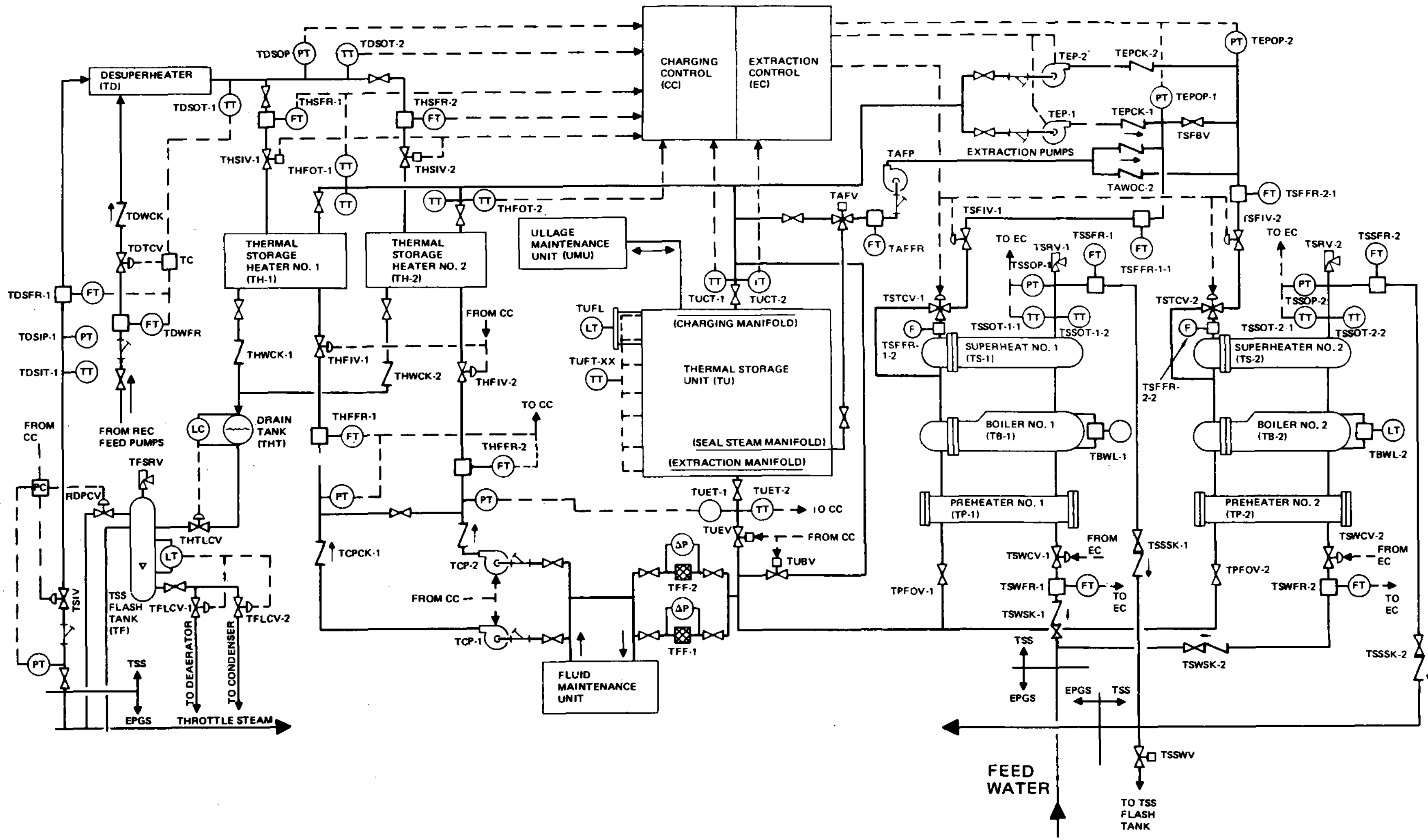


Figure 4-64. Thermal Storage Schematic

4.5 SAFETY CHARACTERISTICS

The Thermal Storage Subsystem (TSS) is designed to meet all personnel and structural safety requirements typically applied in today's industrial community. The complete listing of Government and other applicable specifications is included in Paragraph 2.1 of Appendix A, covering the TSS design. Detailed requirements relating these specifications as they apply specifically to portions of the central receiver thermal power plant have been expanded in Ref 4-1, which includes specific recommendations for each subsystem as well as general design safe practices. The following sections discuss potential hazards as well as addressing the specific requirements outlined in Ref. 4-1.

4.5.1 Potential Hazards

During normal operations of the TSS, a very low degree of hazard will exist to operating personnel when basic procedures are followed. These procedures have been proven to be effective over many years of commercial operating experience using high-temperature heat transfer fluids in the chemical processing industry and petroleum processing industry.

The fluid that will be used in the TSS is Exxon's Caloria HT43 which is a petroleum based product, non-toxic, and non-carcinogenic for external exposure over most of the skin areas of the human body. The greatest potential hazard exists if personnel contact hot metal surfaces or the hot heat-transfer fluid. However, pain or damage to skin areas will not occur if basic procedures are followed. Insulation covers all portions of the line, valves, and fittings, and is designed with a thickness to allow contact without producing a skin burn. The hot fluid will be in a totally enclosed system. Only in the event of an equipment failure is there a possibility of hazard of exposure to hot fluids. During normal operation personnel will need not be adjacent to equipment that may be potentially hazardous. Totally automatic operations of the thermal storage subsystem are planned, with personnel proximity needed only during maintenance. Maintenance can be scheduled on portions of the systems that have cooled down well below the combustion or possible burn hazards limit. Simple precautions with operating personnel will be enforced. It is essential that personnel wear protective eye cover and nonporous insulated gloves at all times when they expect to come in contact with the system; however, this is no more of a precaution than is needed with any type of electrical power-generation station.

The greatest potential hazard exists when personnel break into a line either through fluid filter element changing or component removal or repair. Filter cleanout, if conducted according to procedures that were followed in the SRE systems test, is not hazardous and can be dangerous only if personnel disregard common-sense practices.

A second source of hazard exists with the possibility of fire. At the maximum temperature the fluid is circulated (316°C) it is below the auto-ignition point with air (404°C), but it is above the flash point (216°C), and thus is considered flammable. Caloria HT43 has a low vapor pressure (0.1 psi at 216°C for weathered fluid, which results in a high flash point). Sustained combustion will not occur below the fire point of 204°C (400°F) even with an external ignition source present. Thus, small leaks are not considered to be a fire hazard because immediate cooling of the fluid occurs upon exposure to air. Because there is a remote possibility of a massive spill, all ignition sources are eliminated through use of explosion-proof equipment. A range of fire-fighting equipment will be included in the thermal storage subsystems area because of the possibility of both small and large fires. These will include high flow flooding and spray nozzles using water, small and large hose reels, portable chemical fire extinguishers, and shower and eye sprays to provide immediate relief for personnel if contact with the hot fluid occurs. Massive spills will be contained by a catch basin around the TSU and directed to an adjacent rock-filled catch basin underground. The catch basin will be capable of holding the complete fluid charge in the system. Large interconnecting drain pipes will rapidly channel any spillage immediately into the catch basin. The rock will immediately cool the fluid as well as provide a suffocating effect through the elimination of circulating air to restrict combustion. The low vapor pressure of the fluid, if in the catch basin, will help eliminate fuming vapors over the basin which could otherwise be a source of secondary combustion.

Water sprinkler heads will be located strategically in the primary equipment area (containing pumps, heat exchangers, TSU, etc.). For almost all conventional operations the heat-transfer fluid will be handled in closed containers for any significant quantities. Venting and bleeding of lines and draining during repair and maintenance will be done through a small auxiliary system that will prevent accumulation of the fluid in the work area. Also, the general work area will be kept clean and free of oily deposits, which enhances safety as well and eliminates the potential health hazard of falls and objectionable fumes.

In addition to the fire control and containment systems, the TSU and equipment area will be under constant surveillance and smoke detection sensors with appropriate alarms to the manned stations (i.e., control center, work shop area, etc.). A closed-circuit television will be installed with a TV camera that can scan the area under control of operating personnel in the control center. Use of sensors in combination with the TV camera will provide instant response. The sensors will provide an audible alarm and a TV camera can scan the area to observe the extent of a hazardous situation and determine if sensors have generated a false signal.

4.5.2 Requirement Criteria

This section addresses itself to the specific requirements highlighted by Ref. 4-1 and discussion of the design features as they meet the requirements. Rocketdyne's evaluation of the specific requirements is that they are necessary and sensible and not overly restrictive.

4.5.2.1 Containment

Earth berms and/or retaining walls must be provided around thermal storage tanke containing flammable or combustible liquids. The design should be in accordance with Title 29 - Labor (Reference CFR Parts 1910.106 and 1926.152).

The thermal storage tank is contained in a 2m (6 ft) deep depression of a size that will hold 100% of the liquid in the tank. In addition, an adjacent drain pit with large opening access to the tank area will hold 100% of the volume of liquid in the tank.

4.5.2.2 Barrier Protection

The protective barrier must be provided between noncompatible substances to prevent mixing.

The principal source of comingling of fluids is in the heat exchanger where there is a possibility of the fluid draining into the water side or the water draining into the fluid. Those heat exchangers where a joint is made at a tube sheet, which is traditionally a source of leakage, will use double the sheet construction to provide a drain area between the water and heat-exchanger fluids. The water and the heat-exchanger fluids will not react if mixed; however, there is the possibility of excessive outgassing from conversion of the water to steam by the hot heat-transfer fluid.

4.5.2.3 Automatic Shutdown

The thermal storage subsystem will accept and execute control commands, detect a malfunction, and initiate fail-safe shutdown.

The system will automatically shut down at the detection of a safety hazard or hazardous deviations from specification performance. Automatic shut-down as well as the reason for shutdown will be flagged to the operator at his control panel.

4.5.2.4 Hazardous Leak Detection

The ability must exist to detect hazardous leaks in the thermal storage system and provide a method to isolate this leak from the rest of the system.

Hazardous leaks could originate from high pressure steam, high pressure high temperature water or the heat transfer fluid loops. Fluid and possibly steam leaks would activate the smoke sensors. The operator then can scan the area with the TV monitor and take the appropriate corrective action (i.e., terminate, isolate, activate firex, etc.).

4.5.2.5 Leak Monitoring

The ability must exist to detect hazardous leaks in the thermal storage system and provide a method to isolate this leak from the rest of the system.

The most serious internal leakage affecting the TSS is water in the heat transfer fluid. Small leaks of water into the fluid side will involve steam formation which will be vented ultimately to the TSU and then to the ullage maintenance unit. Large amounts of water leakage may produce excessive pressure in the fluid side with the possibility of rupture of the heat exchange shell. The heat exchangers will be supplied with safety valves to prevent rupture if an overpressure should occur on the fluid side. Any leaks of water into the fluid will be detected with the water detection sensors in the ullage maintenance system.

4.5.2.6 Safety Showers and Eye Wash Fountains

Safety showers and eye wash fountains will be provided in the vicinity of tanks containing toxic materials in accordance with Title 29 - Labor Requirements (Reference 1910.151).

Safety showers and eye wash fountains will meet the requirements and be located clear and free in readily accessible areas in several locations in the TSS.

4.5.2.7 Closed-Cell Insulation

Closed-cell insulation is required for all areas where a fluid leak may occur. The principal candidate areas for closed-cell insulation are at all system penetrations or nonwelded connections such as instrument connections, valve packing glands, flanges, etc.

The thermal storage subsystem will use closed-cell insulation in areas where a leak may occur. These have been identified as piping and equipment connecting flanges, and instrumentation penetrations.

4.5.2.8 Heat-Transfer Media Temperature Monitoring

The design must include provisions to monitor heat-transfer media temperatures throughout heat-up operations to ensure that localized temperatures do not exceed systems design value.

The system has been designed with temperature sensors throughout the fluid network. In addition, the possibility of damaging the heat-transfer media will be minimized by establishing thermal characteristics of the various pieces of equipment so that predictions can be made as to warmup times. These predictions will be part of the Master Control computer program so that, if temperature responses from sensors do not correlate with the past temperature history of the component, the system will be shut down or display a warning flag. In addition, certain operations will be forbidden and control systems will be programmed appropriately. For example, steam input to the thermal storage heater will not be allowed unless a minimum flow of heat-transfer fluid through the warmup circuit is registered to avoid stagnation conditions that could damage the heat-transfer media at this point.

4.5.2.9 Inert Gas Blanket

The inert gas ullage system for the TSU will have a safety release device to protect against pressure buildup which may exceed design levels. The inert gas blanket in the TSU is controlled by the ullage maintenance unit which provides a three-step activation in case of overpressurization. The nominal operating pressure range in the thermal storage unit is from 7.6 to 12.7 cm (3 to 5 in.) of water gage pressure. At 11.4 cm (4.5 in.) of water pressure the ullage scavenge pump starts to remove gases from the TSU tank ullage space. At 14 cm (5 in.) of water pressure, the ullage pressure transducer signals a warning in the instrumentation system. At 15.2 cm (6 in.) of water pressure, a high-pressure alarm switch signals the operator that the pressure is excessively high. At 17.8 cm (7 in.) of pressure a high volume relief valve is activated in the ullage space and simultaneously a second high-pressure warning is displayed for the operator. In addition to the normal control systems as described, the nitrogen supply system to the ullage space has a smaller capacity than the venting system in case the supply is not regulating correctly. At maximum rate the volume of the space is such that a considerable amount of time would occur before pressures would build to dangerous levels.

Even if all normal pressure control units in the TSU ullage space fail and pressure builds up to a failure point (an almost impossible event), the tank will be designed so that the roof will fail before the wall and bottom, thus ensuring that the flammable liquid will be retained within the structure and not present a fire hazard. A failure in the roof will allow fuel rich gases to escape that could burn, but combustion would be limited to the area adjacent to the location of rupture. The fire systems and water nozzles will be designed to provide enough water to control a fire in this location.

4.6 PILOT PLANT THERMAL STORAGE DESIGN SUMMARY

4.6.1 Subsystem Summary

The 10-MWe Pilot Plant TSS employs sensible heat storage using dual liquid and solid media for the heat storage in a single tank, with the thermocline principle applied to provide high-temperature extractable energy at constant rates and temperatures independent of the total energy stored.

The thermal storage subsystem is required to have an extractable storage capacity of at least 103.8 MWHt, which is composed of 7.5 MWHt to provide a turbine hot start and 96.3 MWHt to permit the turbine-generator to supply 7 MWe net (7.8 MWe gross) for 3 hr following turbine startup. The required charging rates are 1.5 MWt to 30 MWt. The maximum allowable heat loss is 3% of extractable capacity in 24 hr, starting in a fully charged condition. The subsystem is required also to provide nighttime seal steam at a temperature of at least 135°C (275°F) and at a rate of 0.33 MWt for approximately 16 hr.

There are five fluid streams crossing the boundaries of the TSS, all water or steam flows. A major requirement is to accept steam from the receiver at 343°/510°C (650°/960°F) and 10.1 MPa (1465 psia), where the two sets of temperatures correspond to derated and rated steam operation, respectively. Another major requirement is to supply steam from the TSS steam generator for the turbine at 277°C (530°F) and 2.76 MPa (400 psia). Figure 4-64 is a schematic diagram of the thermal storage subsystem, showing all major components, lines, and major control concepts. Process flow conditions are shown at various points in the subsystem on the subsystem process schematic of Figure 4-65.

The TSS is divided into nine components which are described in the subsections which follow. As shown in Figure 4-64 they are: (1) the TSU, a tank which stores and dispenses thermal energy via the Caloria HT43 heat-transfer fluid, (2) the UMU which provides an inert nitrogen gas cover over the fluid surface in the tank, (3) the fluid maintenance unit, which removes suspended and dissolved impurities from the fluid, (4) the DSH which limits

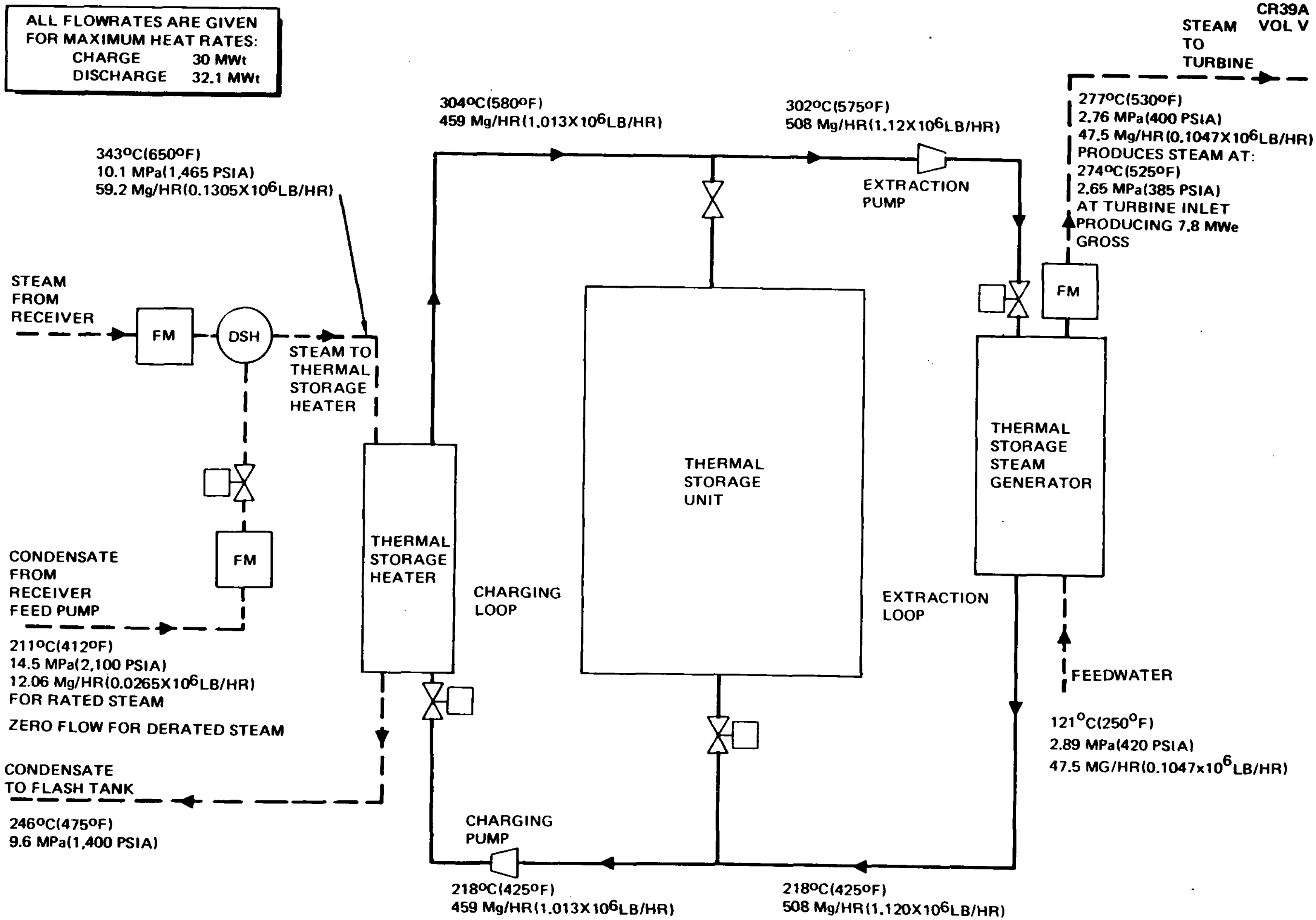


Figure 4-65. Schematic Flow Diagram of Thermal Storage Subsystem for 10-MWe Pilot Plant

incoming steam temperature, (5) the thermal storage heater, which served to transfer heat from the condensing steam to the fluid, (6) the thermal charging loop comprising the charging fluid pump and associated equipment connecting items 5 and 1, (7) the steam generator, which transfers heat from the fluid to make steam for the powerplant, (8) the extraction loop comprising the extraction fluid pump and associated equipment connecting items 7 and 1, and (9) the controls and instrumentation for the subsystem.

Each subsection briefly states the purpose of the component and its principal design requirements insofar as not stated in this section and concludes with a description of the main features of the component as designed.

4.6.2 Thermal Storage Unit

The function of the TSU is to act as a reservoir for solar thermal energy, which charges the TSU during hours of high insolation. At other times, when insolation is partially or completely unavailable, thermal energy is extracted from the TSU to produce steam for the electrical generation subsystems.

In addition to the requirements set forth in Section 4.6.1, the TSU must be able to supply thermal energy to the steam generator at rates varying from 3.1 to 32.1 MWt, and at fluid temperatures in the range from 302°-294°C (575° to 560°F). It must be able to operate over a maximum-minimum fluid temperature range from 219°C (425°F) to 302°C (575°F).

The dimensioned sketch of Figure 4-66 shows the Pilot Tank TSU design, including data on steel tank, soil foundation and packed bed. Additional numerical information is given in Table 4-31.

4.6.3 Ullage Maintenance Unit

This unit is to provide a controlled-pressure, oxygen-free gas atmosphere above the fluid in the TSU. It is necessary to have an oxygen-free gas above the heat-transfer fluid surface to prevent fire hazards and long-term oxidation of the fluid.

The ullage pressure must be controlled within a moderately narrow band to avoid underpressurizing or overpressurizing the tank since the specific

SHELL COURSE SCHEDULE (ASTM 537 CLASS 2 STEEL)				
COURSE	HEIGHT M (FT)		PLATE THICKNESS MM (IN.)	
1 (BOTTOM)	1.83	(6)	19.0	(0.75)
2	1.83	(6)	16.3	(0.64)
3	1.83	(6)	13.5	(0.53)
4	1.83	(6)	10.7	(0.42)
5	1.83	(6)	7.9	(0.31)
6	1.83	(6)	6.4	(0.25)
7 (TOP)	2.44	(8)	6.4	(0.25)

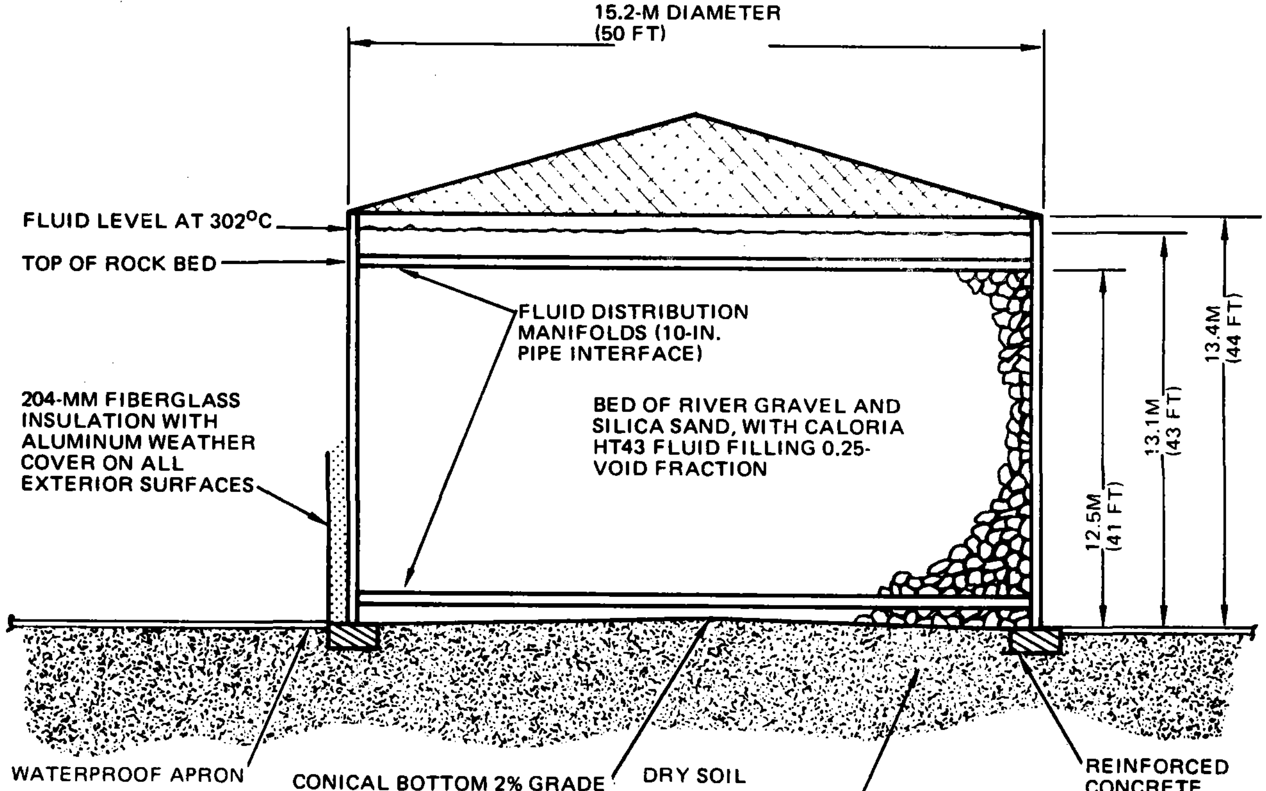


Figure 4-66. Design for 10-MWe Pilot Plant Thermal Storage Unit

Table 4-31 (Page 1 of 2)

DESCRIPTION OF DESIGN FOR 10-MWe PILOT PLANT THERMAL STORAGE UNIT

TSU Configuration

One cylindrical tank, axis vertical, installed above ground, supported on dry soil of 0.29 MPa (6,000 psf) bearing strength by excavation to 1.83m (6 ft) below grade.

Tank Dimensions (Heights Measured at Circumference)

Inside diameter 15.25m (50 ft)
Overall height 13.4m (44 ft)
Packed bed height 12.5m (41 ft)
Free fluid surface height 13.1m (43 ft) at 302°C (575°F)
Effective height of top manifold 12.4m (40.6 ft)
Effective height of bottom manifold 0.23m (0.75 ft)
Effective height of seal steam manifold 0.9m (3.0 ft)
Tank cross-sectional area 182m^3 (1963 ft^2)

Tank Thermal Performance

Extractable capacity after 20-hr hold time: 103.8 MWHt
Design storage temperatures: maximum 302°C (575°F)
minimum extraction 294°C (560°F)
minimum 219°C (425°F)
Thermal rates: maximum charge 30 MWt, maximum extract 32.1 MWt
minimum charge 1.5 MWt, minimum extract 3.1 MWt
Heat losses during 24-hour hold less than 3% of extractable capacity

Solid Storage Medium

25 mm (1 in.) river gravel and 1.5 mm (1/16 in.) No. 6 silica sand in 2:1 ratio, with 0.25-bed void fraction.
Superficial bed volume per tank 2447m^3 (86,400 ft^3)
Weight of solids per tank 4,530 Mg (4,990 tons)
Total weight of solids (all tanks) 4,530 Mg (4,990 tons)

Table 4-31 (Page 2 of 2)

DESCRIPTION OF DESIGN FOR 10-MWe PILOT PLANT THERMAL STORAGE UNIT

Liquid Storage Medium

Caloria HT43 heat transfer fluid
525 m^3 (139,000 gallons) at 21°C (70°F)
Two manifolds, each with 5,900 holes of 3.1-mm (0.125 in.) diameter uniformly spaced over cross-section
One seal steam manifold

Tank Structural Details

Fabricated of A537 Class 2, structural steel with field-welded construction
Upward conical bottom plate 6.35-mm (0.25 in.) thick, 2% slope
Plate thickness for 1.83m (6 ft) high shell courses varies from 19mm (0.75 in.) at bottom to 6.35mm (0.25 in.) at the top
Roof is single skin with trusses, 1-in-12 pitch conical
Roof and sides covered with 204 mm (8 in.) fiberglass blanket insulation with corrugated aluminum weather cover

Interfaces (Flow Penetrations)

Caloria HT43 piping for primary thermal charging and extraction: top and bottom manifold, each 10-in. pipe
Caloria HT43 for night-time seal steam, 1.5-in. pipe in sidewall at 0.91m (3 ft) height
Nitrogen gas for ullage gas blanket 7-in. pipe into roof.

volume of the fluid (Caloria HT43) expands by about 9.5% as it is heated fror 218° to 302°C (425° to 575°F). If the gas in the ullage space were not released, the pressure would rise above the allowable upper limit. The gas released must be replaced or stored and returned to the ullage space during the cooling cycle since an equal amount of gas is required to prevent the pressure from going below the allowable lower limit as the fluid cools and contracts. Specifically, the pressure in the ullage space must remain within the gage pressure range of zero (i.e., at the ambient absolute pressure) to 2 KPa (0.3 psi). A pressure below ambient is not tolerable because this might permit air to enter the tank.

Since the emerging gases will be hot and will carry with them vapors given off by the fluid, they must be cooled and the condensate returned to the tank or properly disposed of. The fluid vapors are produced at a rate dependent on the time-temperature history of the fluid and must be removed as produced.

The UMU for the 10-MWe Pilot Plant uses nitrogen as the inert gas. The unit recovers, compresses, stores, and reuses the ullage gases for each daily storage cycle. Basically, the ullage pressure in the TSU tank is maintained at 10-cm water gage by an automatic control system, with pressure and vacuum relief valves for ultimate tank protection. As dictated by the ullage pressure sensor, the ullage gases are pumped out of the TSU tank by a 10-cfm compressor to a 3.8m^3 (1,000 gal) storage tank at an operating pressure of 1.2 MPa (175 psig). This pressure condenses some of the fluid vapors pumped out of the TSU tank ullage and the condensate is removed. As the TSU undergoes thermal cycling, a mixture of noncondensables and nitrogen alternately flows into and out of the TSU tank ullage. Makeup nitrogen from a liquefied supply is required only when not enough noncondensables are generated in the TSU. Any surplus noncondensables are flared or otherwise put to beneficial use.

4.6.4 Fluid Maintenance Unit

The basic functions of the fluid maintenance unit are (1) the removal of very low-volatility compounds (the polymerized material formed over long periods of time in the hot fluid via pyrolysis) and solids from the heat-transfer fluid (Caloria HT43), and (2) the addition of fresh fluid to maintain a constant fluid inventory. The results of the prequalification tests, discussed in Section 6.2, provided a rate equation for the fluid loss by volatilization of species produced by thermal cracking. No rate was obtained for the production of polymeric residue. Measurements of kinematic viscosity and gel permeation chromatograph analyses (GPC) have yielded some information on polymer formation.

The principal requirements for the fluid maintenance unit are (1) to keep the level of polymerized materials below a given value, realistically taken as 10%, (2) to remove suspended solids from the circulating fluid, (3) to provide fresh heat-transfer fluid to make up for materials removed from the circulating fluid, and (4) to provide virtually automatic operation using state-of-the-art technology.

The fluid maintenance unit designed for the Pilot Plant uses existing commercial components to keep the fluid in good operating condition by filtration to remove suspended solids, distillation of a side stream to remove high boiling, polymeric compounds, and addition of fresh make up fluid to replace the material removed by this unit and the ullage maintenance unit.

Two 80-mesh filters are provided in the main line at the entrance to the pumps in the thermal charging loop. These filters will remove particles greater than 177 microns (0.0177 cm); this should include most of the rock dust, pipe scale, and other foreign matter likely to be present in the system after assembly. For side-stream distillation a mechanically aided thin-film evaporator is used, operating at a pressure of about 133 Pa (0.0193 psia) to 67 Pa (0.0096 psia). This operates continuously over an 8-hr period, but can function continuously (24 hr/day) if necessary. The distillation unit comprises a 0.74 KEw (1 hp) evaporator motor and a condenser, both constructed of 316 stainless steel, along with a variable-speed drive 60 l/hr feed pump and a 255 l/min vacuum pump. The hot feed stream entering the vacuum evaporator is thrown by centrifugal force against a heated wall to form a turbulent film. The vapors pass from the evaporator to a 0.7m^2 (7.5 ft^2) condensor where the distillate is collected and returned to the main circulating fluid loop by a small distillate return pump. The polymeric unevaporated material is pumped to a waste receptacle. The three liquid pump motors together consume 1.30 KWe (1.75 hp) and the vacuum pump motor 0.56 KWe (0.75 hp).

For fluid replenishment a 30-day supply of fresh Caloria HT43 will be kept on hand. Using current data, the tank size for a 30-day supply is estimated at 4,050 liter (1070 gal). The fluid loss will be replaced once every several days, by pumping fresh fluid from the 2,050-liter storage tank into the TSU main line to the charging thermal storage heater. The volume of makeup fluid actually required will be determined by recording the amount of low and high boilers lost.

4.6.5 Desuperheater

The DSH reduces the temperature of the receiver steam (nominally 510°C, 950°F) enroute to the thermal storage heater to a level not exceeding 343°C (650°F) to avoid shortening the life of the heat-transfer fluid. This is achieved by mixing the receiver steam with boiler feedwater incoming at 205°C (401°F). At the maximum charging rate the required process flows entering and leaving the DSH are:

Steam from Receiver	
Pressure	10.1 MPa (1,465 psia)
Temperature	510°C (950°F)
Flow	16.5 Kg/sec (130,500 lb/hr)
Water from Feedwater Pump	
Pressure	14.5 MPa (2,100 psia)
Temperature	211°C (412°F)
Flow	3.35 Kg/sec (26,500 lb/hr)
Steam Out to Thermal Storage Heater	
Pressure	10.1 MPa (1,465 psia)
Temperature	343°C (650°F)
Flow	16.5 Kg/sec (130,500 lb/hr)

Other requirements are operation at reduced receiver steam flowrates (turndown ratio 20.5), use of state-of-the-art technology, and ease of maintenance.

The design selected is a direct-contact, mixing chamber type with water injected through multiple atomizing nozzles. The single chamber $0.61m^2$ (2 ft) diameter, 1.8m (6 ft) in length, made of carbon steel and insulated, is, in effect, an expansion in the main steam line. Three spray nozzles in parallel introduce the feedwater coaxial with the pipe axis in the direction of flow. The spray nozzles are mounted on a common manifold which is easily withdrawn from the chamber for cleaning the spray nozzles. A controller sensing DSH outlet temperature and flowrate adjusts the water inflow to produce outlet steam at the required temperature.

4.6.6 Thermal Storage Heater

The thermal storage heater is a heat exchanger (two units in parallel) which transfers heat from the incoming steam to the heat-transfer fluid during charging periods. Thermally, on the steam side, superheated steam at 343°C (650°F) and 10.1 MPa (1465 psia) over a flowrate range for each heater from 29,660 Kg/hr (65,250 lb/hr) to 2,970 Kg/hr (6,525 lb/hr) is condensed to an outlet condition of 246°C (475°F) at 9.66 KPa (1400 psia). The steam heats the Caloria HT43 from an inlet temperature of 218°C (425°F) to an outlet temperature of 304°C (580°F), with an allowable pressure drop of 0.17 MPa (25 psid), over a flowrate range for each heater from 230,230 Kg/hr (506,500 lb/hr) to 23,030 Kg/hr (50,660 lb/hr). The heater operates approximately 8 to 10 hr/day, with daily thermal cycling, with a 30-hr service life. Also, transportability by road and rail is to be feasible.

The type of heat exchanger design selected is the TEMA type "DFU," removable bundle, two shell pass, 6 tube-pass, U-tube heat exchanger. Two such units, built of carbon steel and insulated, are in parallel, each rated at 15 MWt, with steam/water in the tubes and Caloria HT43 in the shell. The U-tubes are in the horizontal plane and there is a permanent longitudinal baffle in the shell. A rotated square tube pitch facilitates mechanical cleaning. The geometry and weight of each of the two heaters are as follows:

Tubes:	Number	450
	Average Length	17.2m (56.6 ft)
	ID	1.5cm (0.6 in)
	OD	1.9cm (0.75 in)
	Pitch/Diameter Ratio	1.25

Shell:	Diameter	104 cm (41 in)
	Length	9.6m (31.5 ft)
Heat-Transfer Area		464.5m^2 (5000 ft^2)
Weight		24,970 Kg (55,000 lb)

4.6.7 Charging Loop

The charging loop consists of motor-driven pumps, piping, and valves which make up the heat-transfer fluid circuit connecting the thermal storage heaters to the thermal storage tank. Using existing technology, the requirements for the charging loop design include ability to operate over the range of flowrates through the heater at high reliability with parasitic energy consumption and lower capital cost. These requirements were met by an optimization in which lower energy consumption and other operating costs through multiple pumps and lower fluid velocities were traded off against higher capital costs, in light of a variable diurnal duty cycle.

For the optimum design, the fluid velocity is 3.9 m/s (13 fps), resulting in a pump size of 25.4 cm (10 in). Two identical pumps, in parallel with a total maximum capacity of 0.09 m^3/sec (1,450 gpm), were selected on the basis of power consumption and reliability. The pump drivers are dual-speed (1,750/1,150 rmp) 80-hp squirrel-cage induction motors. Since motor speed is inversely proportional to the number of poles in the starter windings, two speeds can easily be obtained by wiring the motor appropriately. The motor is 480V, 3-phase, two-winding type.

The charging loop main lines consist of 61m (200 ft) of 10-in. Schedule 40 piping, and the segmented flow is carried in 21m (70 ft) of 8-in. Schedule 40 pipe. A total of 14 valves, 6 tees, and 9 90° elbows make up the remainder of the loop.

4.6.8 Steam Generator

The thermal storage steam generator uses heat extracted from the TSU by the fluid (Caloria HT43) to provide steam for turbine startup and operation during periods of 10w- and zero-insolation as well as for blanket heating of heat exchangers and turbine seals during nighttime shutdown periods.

The principal subsystem performance requirements stated in Section 4.6.1, result in the following specific requirements for the steam generator, based on "pinch point" considerations between the steam and Caloria HT43:

Feedwater/Steam Flow Rate	
Maximum	13.21 Kg/s (104,700 lb/hr)
Minimum	1.27 Kg/s (10,100 lb/hr)
Feedwater Inlet Conditions	
Temperature	121°C (250°F)
Pressure	2.90 MPa (420 psia)
Outlet Steam Conditions	
Temperature	277°C (530°F)
Pressure	2.76 MPa (400 psia)

The optimized design was developed through additional considerations, such as reliability and economics, while restricting sizes and other parameters to those commonly used in the commercial heat-exchanger manufacturing industry. Based on reliability factors, the concept of a kettle boiler with separate preheater and superheater elements was selected, all made of carbon steel. Identical twin units are employed in parallel for each of the three heat exchangers. Table 4-32 gives the geometry and other data for all steam generator components.

Because of the undesirable effects which may result from leakage of Caloria into the feedwater circuit, tube to tube sheet joints are rolled to give a primary seal and then welded to provide backup protection. In addition, a GN_2 blanket is provided on the shell (water/steam) side of all elements during shutdown periods to ensure that the tendency for leakage would be toward the Caloria.

4.6.9 Extraction Loop

The extraction loop consists of the same types of components as the charging loop, except that the extraction loop connects the TSU to the steam generator. Further, its duty cycle normally calls for operation at or near maximum flow and temperature conditions for the duration of the extraction period, in contract to the charging-loop duty cycle which varies throughout the day.

Table 4-32

PILOT PLANT STEAM GENERATOR DESIGN SUMMARY

	Preheater	Kettle Boiler	Superheater
No. of Units	2	2	2
Configuration	Straight Tube, Floating Head	Horizontal U-Tube	Horizontal U-Tube
Tube Side Fluid	Caloria HT43	Caloria HT43	Caloria HT43
Shell Side Fluid	Water	Water/Steam	Steam
No. of Tubes	431	650	225
Tube Length	8.54m (28 ft)	22.9m (75 ft)	7.0m (23 ft)
Tube OD	1.9 cm (0.75 in.)	1.9 cm (0.75 in.)	1.9 cm (0.75 in.)
Tube Wall Thickness	0.21 cm (0.083 in.) (BWG-14)	0.21 cm (0.083 in.) (BWG-14)	0.21 cm (0.083 in.) (BWG-14)
Pitch	2.38 cm (15/16 in.) (Staggered)	2.38 cm (15/16 in.) (Staggered)	2.38 cm (15/16 in.) (Staggered)
Shell Length	9.15m (30 ft)	10.82m (35.5 ft)	3.65m (12 ft)
Shell Diameter	0.64m (2.1 ft)	1.1/1.67m (3.6/5.5 ft)	0.64m (2.1 ft)
Shell Wall Thickness	1.11 cm (0.438 in.)	2.54 cm (1 in.)	1.11 cm (0.438 in.)
Mean Area	$196 m^2$ ($2,106 ft^2$)	$791 m^2$ ($8,513 ft^2$)	$84 m^2$ ($904 ft^2$)
No. of Passes	4	1	2

The extraction loop requirements include, as for the charging loop, minimum parasitic energy consumption at high reliability and low capital cost.

For the optimized design the fluid velocity is 5.2 m/s (17 fps), which corresponds to a pipe diameter of 22 cm (8.6 in.). However, the closest standard pipe sizes, i.e., 10-in. Schedule 40, was selected, giving fluid actual velocities at maximum conditions of 4.7 m/s (15.3 fps). Two identical pumps in parallel with a total maximum capacity of 0.13 m^3/s (2,000 gpm) are used, because of enhanced reliability. The pump drivers are 70-hp single-speed (750 rpm) induction motors, 480V 3-phase.

The extraction loop main lines consist of 48.8m (160 ft) of 10-in. Schedule 40 piping, and the segmented flow is carried in 19.8m (65 ft) or 8-in. Schedule 40 piping. A total of 10 valves, 9 tees, and 6 elbows make up the remainder of the loop.

4.6.10 Controls and Instrumentation

The overall control system is required (1) to respond automatically to commands from operating personnel and/or from master control, (2) to change smoothly and rapidly from one operating mode to another, and (3) to respond smoothly and rapidly to changes in thermal charging rate or extraction rate or both. The overall control system can be divided into three parts: charging loop control, extraction loop control, and DSH control.

The charging loop control maintains the fluid output temperature from the charging heat exchangers at the command set point, nominally 302°C (575°F), within ±1°C of the nominal setting as a steady-state error band. Transient overshoot is allowed to be about ±4°C from the nominal final value. In general, operating above the nominal maximum steady-state temperature increases the degradation rate of the fluid. Operation below the nominal reduces the total heat stored and reduces turbine efficiency. Transient errors in temperature can be allowed to be relatively large.

The extraction loop control maintains steam delivery at the nominal maximum temperature and pressure of 276 ± 4°C (530 ± 7.2°F) and 2.76 ± 0.069 MPa (400 ± 10 psia), respectively, with a rather broad tolerance on the temperature and pressure, since tighter requirements by the turbine are not necessary.

The desuperheat control must hold the desuperheated steam output temperature at 343 ± 1.1°C (650 ± 2°F). Transient response can allow ±5.6°C (10°F) overshoot from the final value.

In general, response times need not be rapid since the large thermal inertias involved are expected to provide the longest controlling lag time in all these control loops. In addition, TSS damage will not result from control errors or overshoot because the subsystem is inherently a passive one.

Instrumentation capabilities are required for direct monitoring of temperatures, pressures, flowrates, and liquid levels with display and recording capabilities at a remote control panel, and use of these signals by data loggers or computers to store data and calculate performance and operational parameters. A combination of instrumentation, control, and logic functions are required in the charging loop to aid personnel in automatically switching from one combination of pumps and heat exchangers to another as the demands on the system are changed.

The design features of the three controls and their interaction are described briefly in the following paragraphs and diagrams. A total of 18 temperature sensors are used as direct control system inputs, another 20 are used for manual control, another 20 determine the TSU bed thermal charge condition, and 72 more are used elsewhere in the TSS. There are 18 pressure measurements and 14 flowmeters to measure major flows in the system. Seven major throttle valves are used, two in the charging loop, four in the extraction loop, and one in the DSH loop.

Figure 4-67 is a schematic diagram of the charging loop control. Heat transfer fluid from the lower manifold of the thermal storage unit (TSU) is pumped through the dual thermal storage heaters, TH-1 and TH-2, in which the fluid temperature is increased by heat transfer from a supply of steam from the DSH. Two centrifugal fluid pump units in parallel, TCP-1 and TCP-2, are powered by two-speed alternating current motors. Flowmeters THFFR-1 and THFFR-2 measure the flowrates of the pumps, which may be run singly or in tandem.

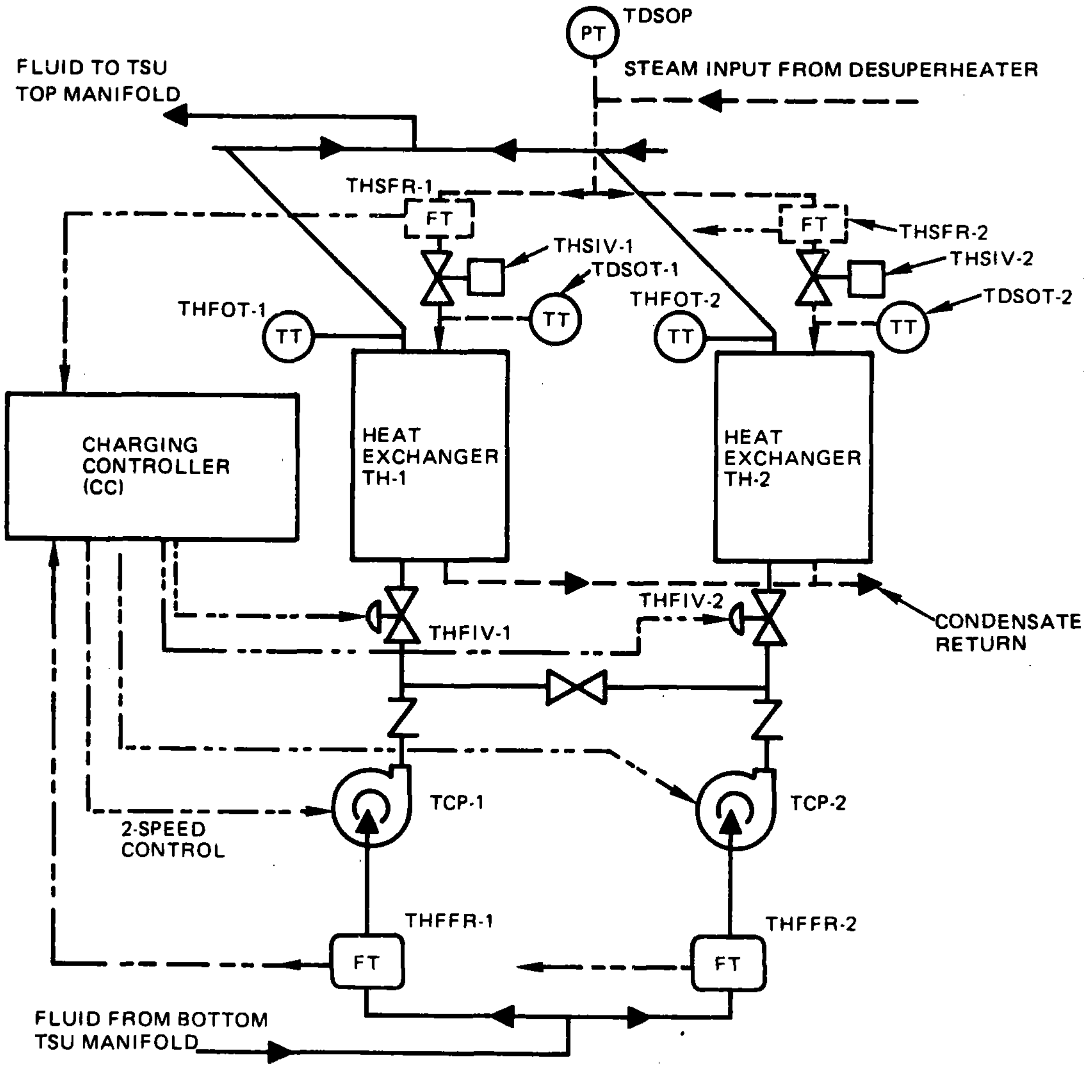


Figure 4-67. Charging Heat Exchanger Control Loop Components

The charging controller (CC), which physically is a portion of a larger assembly identified as the Thermal Storage Unit Controller, modulates thermal storage heater fluid inlet valves THFIV-1 and THFIV-2 for continuous adjustment of each pump flowrate to maintain constant heater fluid outlet temperatures THFOT-1 and THFOT-2, as heating steam flowrates vary. Solenoid pilot operated steam inlet valves THSIV-1 and THSIV-2 provide for on-off control of steam flow through each heater. Steam in flowrates to the heaters are measured by flowmeters THSFR-1 and THSFR-2. Steam pressure and temperature measurements obtained with transducer PDSOP and redundant sensors TDSOT-1 and TDSOT-2 provide data to the charging controller for relating steam weight flowrate to the measured volumetric flowrate, and for determining the incoming available heat energy.

Figure 4-68 is a schematic diagram of the extraction loop control. Heat-transfer fluid is pumped from the upper hot end of the TSU, through two parallel steam generators, to the lower cooler end of the TSU. Two centrifugal pumps operate in parallel. A manually operated hand valve, TSFBV, interconnects the pump outlets so that either pump can be used to pump fluid through either of the two steam generators. With interconnect valve TSFBV open and both pumps in operation, if one pump should fail or inadvertently shut down, the other pump can supply fluid to both steam generators. If needed, the extraction controller provides a signal to the power plant master controller to limit the steam flowrate demand. When the two steam generators are operating in parallel, each is controlled independently, e.g., for steam generator No. 1, steam flowrate is measured by flowmeter TSSFR-1, whose outputs, together with superheater exit pressure and temperature, are transmitted to the extraction controller.

The DSH control regulates the mixing of incoming steam at 510°C (950°F) and variable pressure with coolant water to produce outlet steam at 343°C (650°F). The pressure, temperature, and volumetric flowrate of the incoming steam are measured for input to a process control computer that computes the total enthalpy of the steam. A coolant water weight flowrate command signal is proportional to the incoming enthalpy. The desuperheater outlet steam temperature is measured and compared with a reference signal, and an integrating amplifier delivers a signal which is compared with a signal

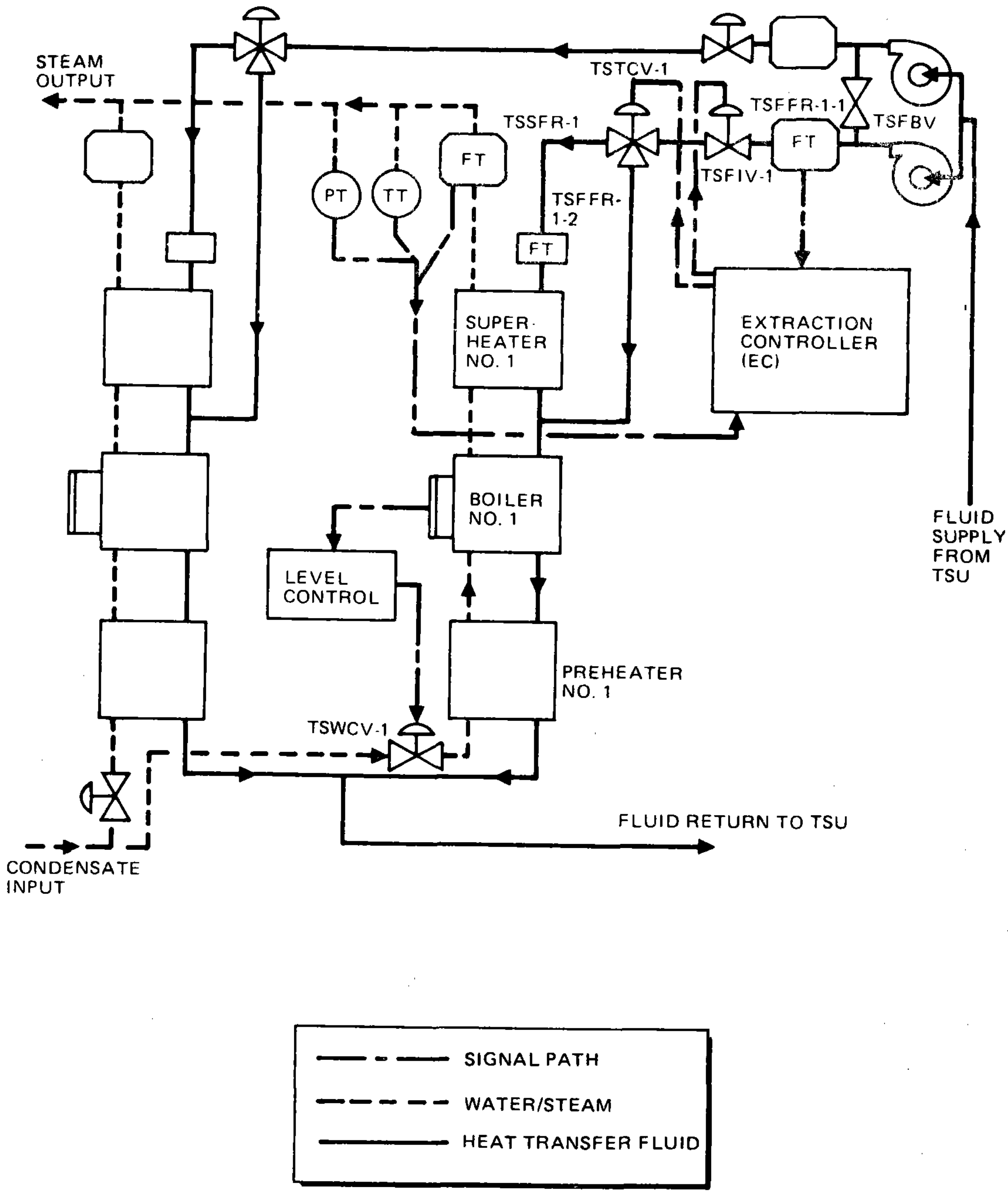


Figure 4-68. Block Diagram of Extraction Loop Controller

that indicates a measured value of water flowrate. A flowrate error results in corrective action displacement of a water control valve which, typically, throttles the coolant inlet pressure from 14.5 MPa (2,100 psig) to 10 MPa (1,450 psig). With this temperature control concept, the commanded coolant water flowrate is primarily a scheduled function of the computed incoming heat energy.

Additional control and instrumentation components are found in the UMU and fluid maintenance unit, which operate independent of the charging and extraction loops. Further, numerous solenoid-pilot-operated shutoff valves in the TSS are sequenced for startup and shutdown operations, chiefly with electrical signals and pneumatic actuators. The total thermal storage subsystem may be operated automatically, semiautomatically with operator supervision, or manually, as desired.

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Section 5
PILOT PLANT PLANS

This section contains the plans for designing, fabricating, installing, activating, and maintaining the thermal storage subsystem.

The overall summary design, fabrication, installation, and checkout schedule is shown in Figure 5-1. The schedule assumes that the program ATP is February 1978 and that overall TSS subsystem requirements are established at that time. Detailed specifications and drawings are completed and the bid and procurement cycle is initiated during the first 8 mo. Shop components are fabricated during a 10-mo period and delivered on-site by the beginning of installation and assembly on 1 July 1979. All checkout and activation occur during 1980 with the subsystem ready for 24-hr on-line electrical power generation for the 2-yr development program by 1 January 1981.

5.1 SUBSYSTEM MASTER SCHEDULE

The subsystem master schedule is shown in Figure 5-2, covering the 3-yr design, fabrication, installation, and checkout of the Pilot Plant thermal storage subsystem. The elements shown in the schedule are integrated with the overall construction and program plan for the central receiver solar thermal plant. The 35-mo schedule, Figure 5-2, assumes an authorization to proceed ATP on 1 February 1978, and the design, fabrication, and construction is predicted on the fact that overall specifications for subsystems and major subsystem components will be established within the first 6 mo of the program.

The analysis and detailed design phase for the first 8 mo of the program will involve converting overall requirements to detailed specifications and drawings that will be used for the purpose of bidding and constructing elements of the system. The first program milestone is the preliminary design review, which is scheduled for 1 August 1978. At that time detailed specifications, plus those long-leadtime items necessary for early procurement, will be presented for approval.

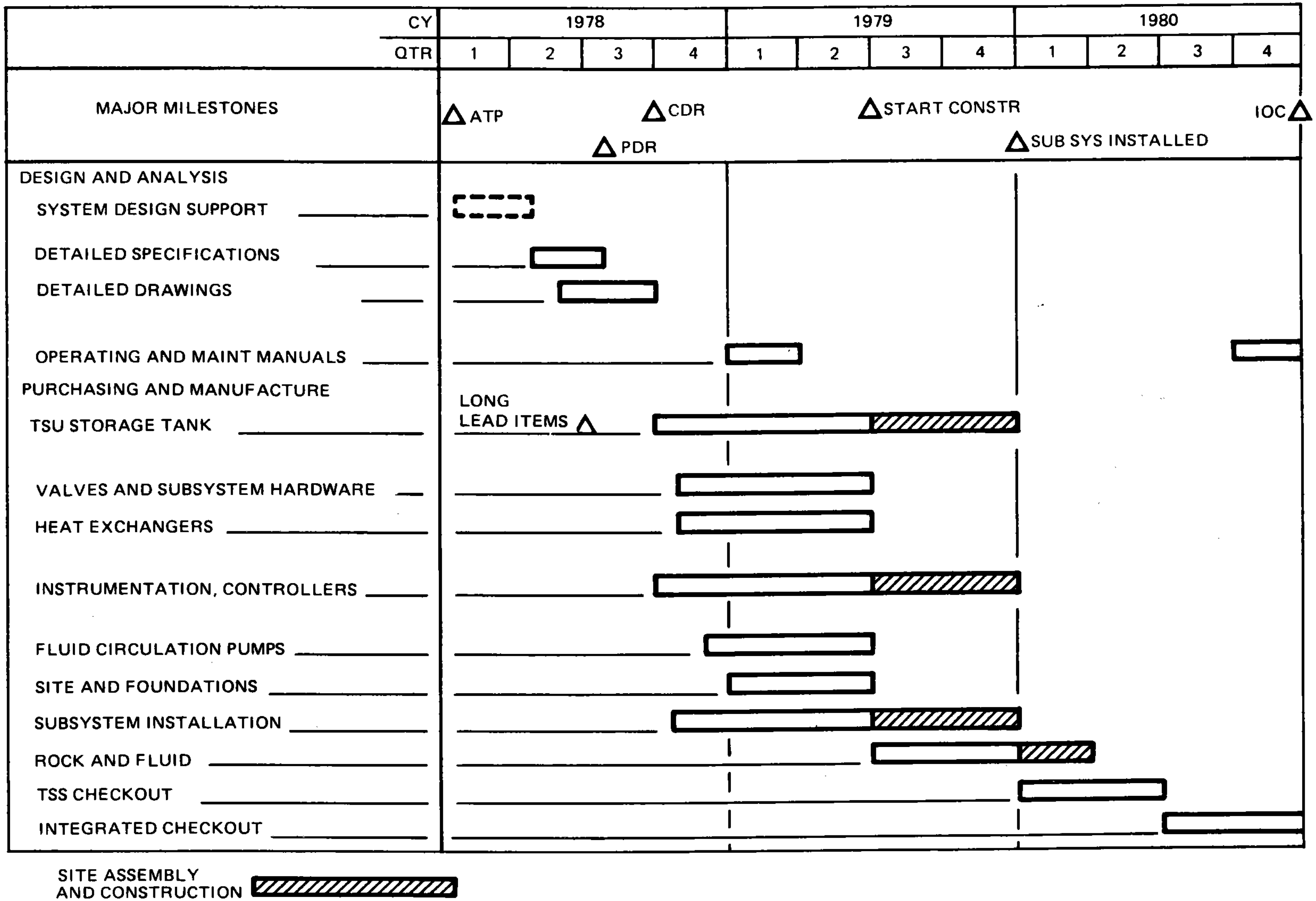


Figure 5-1. Thermal Storage Subsystem, Pilot Plant — Design, Fabrication, and Checkout Summary Schedule

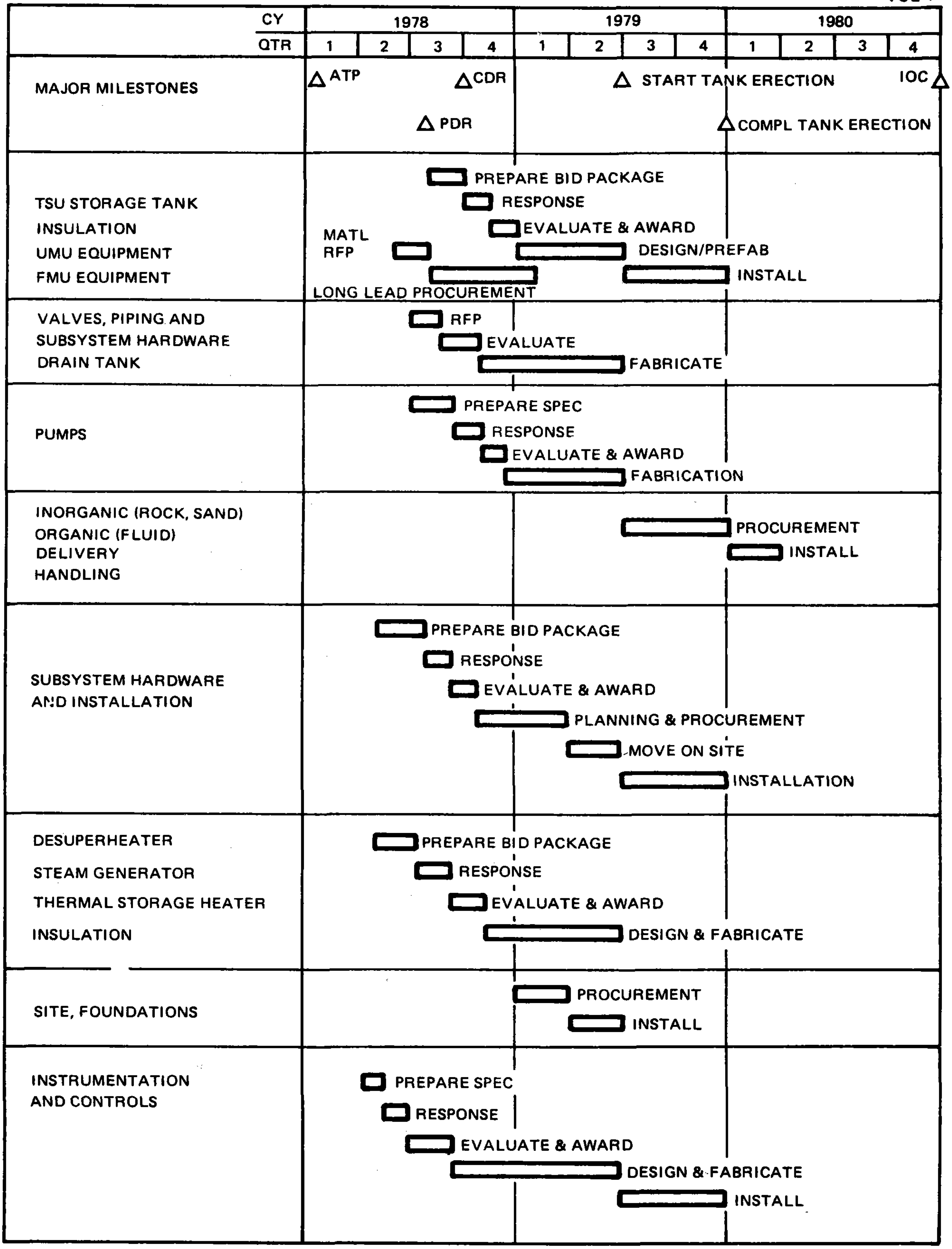


Figure 5-2. Thermal Storage Subsystem: Pilot Plant Procurement, Manufacturing, and Installation Schedule

5.2 PRODUCTION PLAN

Off-site fabrication (production) will include prefabrication of segments of the TSU, and production of controllers, heat exchangers, pumps, instrumentation, filters, and other stock to be delivered to the construction site for warehousing and installation. No unusual production techniques or use of other than existing commercial manufacturers and fabricators will be required.

5.2.1 Summary Schedule

The schedule for the production plan consists primarily of a 10-mo period beginning in mid 1978 and ending with the beginning of installation and subsystem assembly on 1 July 1979. This subsystem is different from the other subsystems in that no production tooling is required. Production for the TSS consists of purchasing state-of-the-art technology from well-established suppliers. The system is primarily unique in the arrangement of standard components, the flexibility of operation, and the cost-saving storage medium. Subsequent paragraphs describe the portions of the plan which deal specifically with design, procurement, transportation, and quality assurance (during this portion of the program).

5.2.2 Design Plan

The design plan consists of the following tasks and assumes that complete TSS specifications have been established at the beginning of the program.

1. Conduct detailed analysis of the TSU, and prepare specifications for long-leadtime material procurement. The structural steel is a long-lead item and will be purchased after approval at the preliminary design review. Tank construction specifications will be prepared and released for bids after approval at the critical design review. Tank construction will use shop prefabrication as much as possible to minimize cost and erection time. Shop drawings of the successful bidder will be reviewed prior to initiation of fabrication.
2. Prepare performance and design specifications for the heat exchangers and reaffirm predicted procurement leadtimes. Shop drawings of the successful bidder will be reviewed and approved prior to initiation of fabrication.

3. Design site preparation, including TSU foundation, drain sump and interconnecting drain lines, foundation, and supports for all pieces of equipment and interconnecting lines and containment dike.
4. Size all lines and systems and define all equipment with initial attention given to those items that require long procurement time.
5. Complete installation design which will describe subsystem in full detail permitting complete material and equipment "takeoffs" for bidding, procurement, and construction.
6. Specify all cleaning, testing, and quality control requirements to provide quality assurance personnel with procedures needed to guarantee that fabrication and construction meet all design requirements.
7. Prepare material and equipment lists and cost estimate and deliver design package, consisting of construction drawings, equipment lists, specifications, and estimated construction cost to Program Manager for procurement.
8. Produce preliminary version of operating and maintenance manuals.

5.2.3 Procurement Plan

At the inception of the Phase 2 program, a review of all critical components will be conducted to identify long-leadtime items. In the past, the impact of major changes in the national economy or major construction programs have resulted in excessive leadtimes for items that require major shop or field construction. At present, the longest leadtime items are the TSU tanks and the control subsystem, and the only item scheduled for advanced procurement approval at PDR is the TSU tank material. However, all specifications will be ready for review at PDR. There will still be time then to modify these specifications, if necessary, before item procurement.

The critical purchasing and fabrication leadtime is between the end of the second quarter of 1978 and 1 July 1979, which will be the start of construction. The leadtime requirements for valves, subsystem hardware, heat exchangers, fluid circulation pumps, foundations, and site preparation are approximately 9 mo, and purchase of these elements can be delayed until 1 November or later before contracts need be let. The critical design review (CDR), on 1 July 1978, will include the review of all detailed drawings and specifications.

With appropriate approval, the two principal long-leadtime contracts (i.e., for the TSU tank and the subsystem controller) will be let immediately after CDR.

After release of all purchase orders by 1 January 1979, engineers associated with the previous design work will produce a preliminary version of the operating manual and maintenance manual. These will be updated during the last quarter of 1980 after 6 mo of experience with operation of the subsystem. The subsystem installation and assembly contract will be let on 1 November 1978 and consist primarily of the labor involved in assembling (Figure 5-2) and installing and connecting the various components that will be delivered to the construction site by 1 July 1979. Preparation of site and pouring of foundations will occur during the second quarter of 1979 in preparation for the installation of the components by 1 July 1979. The site construction is scheduled between July 1 and December 31 of 1979. Prior to beginning of site assembly, the foundations for the TSU heat exchangers, pumps and other components will be completed in order to begin all installation by 1 July. During the construction period three major installation tasks will be involved: (1) the thermal storage unit tank, (2) the controllers, and (3) the mechanical and electrical subsystems. By 1 January all subsystems will be installed and the rock and sand will be on site. Filling of the TSU tank with rock, sand, and fluid will be initiated and completed during the first quarter of 1980.

TSS checkout will begin on 1 January 1980 with electrical and electronic checks of the control system. After filling the TSU on 1 April 1980, the checkout procedures will be expanded to include the all fluid networks. During the second quarter of 1980 all systems will be operated with the thermal storage subsystem control console, and bed conditioning will begin. Bed conditioning involves both water removal and dust removal and will depend to a great extent on the amount of the heat available at this time. If portions of the heliostat and receiver are operating it will be possible to receive steam anytime during the checkout period in the second quarter and this will facilitate water removal from the thermal storage unit. Water removal does not require steam at maximum conditions and it is possible that partial tests of the heliostats and receiver running under lower rated conditions or portable steam generators can be used to produce enough heat to eliminate a significant portion of the water during the second quarter. If not, bed conditioning will occur during the integrated checkout of the complete power plant using all subsystems, beginning the third quarter of 1980. It is expected that integrated system checkout will allow fine tuning of control settings, checking of overall thermal response and stability and allow control valves to be set that will provide optimal operation with the other principal subsystems. At the end of 1980 it is expected that all subsystems will have operated repeatedly with each other, simulating or duplicating complete operational duty cycles, and be ready for the 2-year development program beginning 1 January 1981.

Major components will be purchased separately and delivered to the construction site to eliminate installation contractors' markup, where possible. The site preparation, including all foundations, will be contracted directly for reasons of schedule and quality control, and to eliminate general contractors' markup on this portion.

The remainder of the scope will be described by specifications and drawings for construction by a mechanical-type general contractor who will subcontract electrical scope, control work, and other tasks, including insulation and painting.

5.2.3.1 Make-or-Buy Plan

Application of Rocketdyne make-or-buy policy has resulted in the following decisions:

1. The heat exchangers will be fabricated by firms specializing in the production of commercial shell and tube exchangers with special consideration given to those firms having designs and standard components that most closely approximate the required units.
2. The TSU tank will be purchased from a fabricator specializing in on-site construction of API storage vessels. Special consideration will be given to those firms having complete off-site, prefabrication facilities.
3. The equipment items will be catalogue equipment or supplier modifications of catalogue equipment. Modifications will be described by specification permitting competitive bidding by all suppliers of

similar equipment. Long-lead equipment items will be purchased separately. All other equipment will be purchased by the mechanical contractor.

4. Field construction will be performed as three tasks by construction contractors licensed and experienced in the tasks that they will be asked to bid. Each of the three selected builders will be responsible for his subcontractors and suppliers, minimizing exposure to scope omissions and jurisdictional disputes.

5.2.3.2 Raw Material Purchases

All raw material will be specified, by design, in a manner that provides definition and control of physical and chemical properties, dimensions, cleanliness, testing, and packaging. Raw material will be purchased by the mechanical construction contractor, inspected at delivery, and warehoused at the construction site.

Rockwell will perform source inspections and surveillance inspection of contractor receiving and storage operations.

5.2.3.3 Delivery Plans

The TSU will be erected at the construction site with inspection and acceptance done on the completed unit. Prefabricated work will be delivered as needed by the erecting foreman. Special site storage will not be needed.

The heat exchangers will be offloaded onto foundations by the mechanical contractor.

Equipment items and raw stock will be warehoused, at the construction site, by the mechanical contractor. Off-loading of carrier vehicles will be performed by the mechanical contractor.

The construction contractor will employ a receiving inspector and a warehouse clerk to control quality of all received equipment and stock and to process receiving and stock reports.

5.2.4 Transportation Plan

Special transportation is not required. Off-loading at the construction site will be performed by the mechanical contractor.

Packaging will be specified by the procurement specifications when special packaging is needed to retain essential characteristics.

Lowest delivery rates compatible with requirements will be used. Government bills of lading will be used for major items, if this service is available.

Shipping clearances to the site will determine the largest shop fabrication subassemblies. Shop fabrication will be preferred and used to the full economic advantage over field fabrication, wherever possible.

Equipment and stock will be received at the construction site, off-loaded by the mechanical contractor, and inspected and warehoused by the mechanical contractor. The receiving inspection function and the warehousing operation will be monitored. The receiving inspector will retain records showing that received material conforms to the procurement specifications and all certifications of physical and chemical properties have been filled with the receiving records. Acceptance of equipment and stock by the receiving inspector will be the basis for payment of the supplier. The warehouse function will maintain the inventory status, as well as to provide physical protection for the material. The warehouse function will also maintain records showing equipment and stock removed from the warehouse for contractor installation.

5.2.5 Quality Assurance Plan

Inspection will be performed at supplier's plant to guarantee conformance to engineering specifications, witness tests, and reveal production flaws requiring inspection.

All equipment and stock will be inspected at the time of receipt and prior to payment to guarantee the following:

1. Material is properly identified.
2. Material certifications covering physical and chemical properties are in order and match the identification of the material.

3. Correct quantities have been received.
4. Proper packaging and absence of shipping damage.
5. Verify all characteristics described by the procurement specifications when source inspection is not performed.

In-process inspection applies to the construction work and will be performed to guarantee that all construction meets specification requirements. In-process inspection is discussed in more detail in Section 5.3.

Acceptance for payment will be based upon a receiving report which shows that the correct quantity is received, and all purchasing specifications are satisfied.

5.3 INSTALLATION, CONSTRUCTION AND, CHECKOUT PLAN

5.3.1 Summary Schedule

The installation and construction phase of the program begins on 1 July 1979 and is completed by 1 January 1980. Checkout begins at the end of installation and continues during 1980. The first half of 1980 involves subsystem checkout. During the second half of 1980 the thermal storage system will be checked out, in operation with the other major subsystems.

5.3.2 Installation and Construction Phase

5.3.2.1 Summary

Construction will be separated into three discrete tasks. This will reduce calendar time by permitting heavier job manning with adequate supervision, and reduces markups by the general contractor. The site preparation task will provide drain pit and basic site preparation to be followed by foundations. The erection of the TSU is a discrete work task, which will be performed as a continuation of the job prefabrication phase. The remainder of the construction scope will be given to a mechanical contractor who will subcontract controls, electrical, insulation, paint, and other separable specialty tasks.

5.3.2.2 Installation Plan

The TSU will be erected as soon as shop fabrication and site foundations permit (refer to summary schedule). Major items will be installed so that piping and installation of minor equipment items can begin.

The ullage maintenance unit (UMU) will be field-fabricated as part of the system fabrication. No off-site work is planned or required. Test and checkout of the unit can be performed independent of the system.

The fluid maintenance unit (FMU) will be installed as part of the system piping and requires no off-site fabrication or special handling. The filter vessels will be procured as equipment items. The distillation subsystem will be assembled on site as part of the mechanical installation. The heat exchangers, including the thermal storage heater, steam generator, and DSH will be installed on foundations, by the mechanical contractor. These items are pacing items in the production plan since their installation is necessary before piping can be started.

5.3.3 <u>Checkout Phase</u>

The checkout phase covers the period 1 January 1980 until the beginning of the 2-yr development program on 1 January 1981.

5.3.3.1 Summary

The checkout phase covers the period 1 January 1980 to 1 January 1981 and is subdivided into two major tasks: TSS checkout, which occurs during the first half of 1980, and Complete Integrated System Checkout, during the second half of 1980. The TSS checkout phase is further divided into (1) preoperational and (2) operational checkout.

The preoperational checkout period consists of verification of proper connections in the electrical and pneumatic control and power circuits, and (where possible) interconnections to the Master Controller. Preoperational checkout will be accomplished during the same time that the thermal storage unit is being filled, and will not interfere with minor changes and adjustments that may carry over from the mechanical systems installation. Preoperational checkout will also include verification of calibration of all mechanical and electrical components that cannot be checked with in-line checkout operations.

The second part of the TSS checkout phase (operational checkout) will occur during the second quarter of 1980 and will involve operation and verification of the fluid and the water/steam networks. When thermal energy is available from the collector/receiver subsystems, it will be fed to the thermal storage subsystem to verify operation under operational conditions. At about the midpoint of 1980, the thermal storage subsystem will be fully operational and ready to interact with the receiver/collector/turbine.

The second half of 1980 will involve integrated control and monitoring operations with the Master Control. During the last quarter of 1980 the TSS will be operated in concert with the total plant, and performance will be fully documented over the complete range of operating variables. At the end of the checkout program, the thermal storage subsystem characteristics, charging and extraction flowrates, pressures, response times, and operation with both the subsystem controller and master control will be established for all major operating modes.

5.3.3.2 Checkout Flow and Requirements

The flow of activities during the checkout phase (subsystem and integrated system checkout) is shown in Figure 5-3, and is described in detail in the following sections.

Preoperational Checkout

Preoperational checkout will occur during the first quarter of 1980 and will provide verification that critical control and operational components have been installed correctly and have been calibrated to the necessary flow, pressure, and temperature requirements. A complete record will be kept of the state of calibration and verification of performance for all components. Those components that have had adequate certification will not be recheked in the laboratory. Components that have questionable certification or that have not been calibrated or certified at the source or prior to installation will be reverified with an engineering laboratory checkout.

On-site checkout will include verification (1) of all electrical and pneumatic circuit connections, (2) that all records and drawings showing connections are current, and (3) all terminal box connections are properly marked. System checkout will be facilitated by a portable checkout module which is

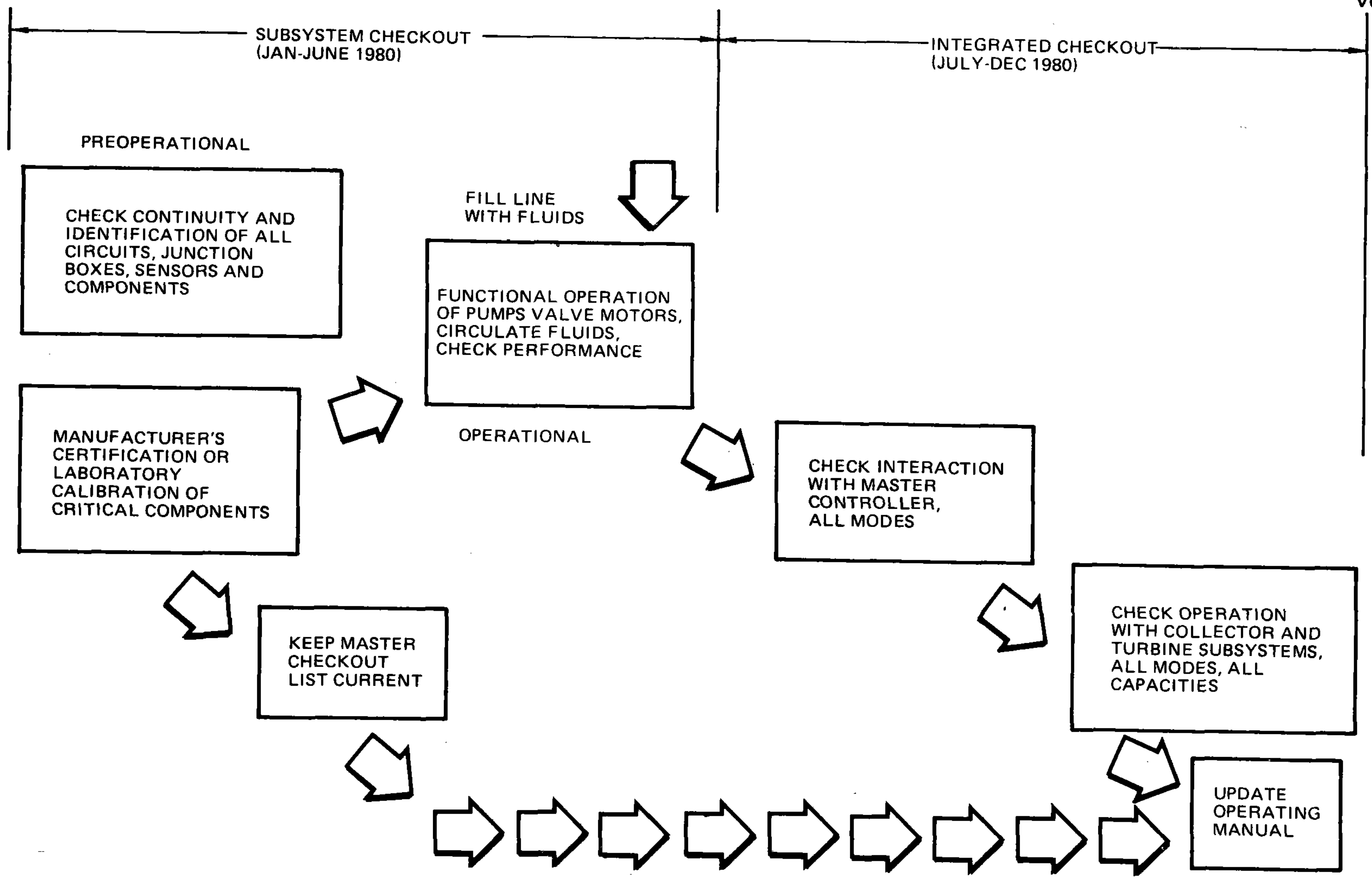


Figure 5-3. Thermal Storage Checkout Operations Sequence

described later. After circuit verification, power will be selectively applied to all circuits to confirm that the appropriate activation occurs. All pumps, solenoid valves, and pneumatic valves will be actuated. Direction of pump rotation and normal-open/normal-close position of valves will be verified. Checkout will include response verification at the operators subsystem console in the control center and with Master Control.

Operational Checkout

Thermal storage subsystem operational checkout will begin in the second quarter of 1980. Initial checkout will include activation and control of all fluid networks from ambient temperature up to whatever level is possible with the heat available from the collector/receiver portion of the plant.

Operational checkout will begin with verification of the operation of all manual and remote-controlled electropneumatic valves. Valves then will be set to the appropriate positions for the subsequent checkout procedures. During the initial portion of the fluid checkouts the UMU will be vented to the atmosphere and the fluid maintenance distillation unit will be disconnected from the operating circuits. The initial checkout activity will involve filling the lines with heat-transfer fluid from the TSU, including venting and bleeding to eliminate all air and gases. During this period all flanges and joints will be checked for possible leakage. When it is verified that the lines are filled, the charging loop will be activated in the warmup mode with one of the charging pumps in operation at slow speed.

All checkout in the charging loop will require both fluid maintenance unit in-line filters to be active. Continuous monitoring of the in-line filters will be necessary during the first 20 to 40 hr of initial checkout operation to avoid pump cavitation as the filters begin to remove solids coming from the TSU. Warmup will continue with operation of the No. 1 and No. 2 charging pumps in both low- and high-speed operation, in parallel, and individually while operating the flow controllers over their complete span. All instrumentation will be monitored at this time and readings will be checked against engineering estimates made during the design of the TSS. Where deviations from performance estimates are large, a diagnostic procedure will be developed to provide insight into the anomalies.

After completion of the warmup loop checkout, the charging loop will be activated with flow through the thermal storage unit. Again, both charging pumps will be operated over the full-flow range with performance being checked against original estimates and with resolution obtained of any anomalies between the measured performance, measured flows, pressures, and temperatures, and the original estimates.

Operational procedures during checkout of the heat-extraction loop will follow those of the charging loop. The steam generator systems units will be operated in parallel and individually over the complete range of flow and pressure conditions. Anomalies with the predicted estimated characteristics will be resolved and performance will be reported and established when operating at room temperature.

Ullage Maintenance Unit Checkout

Checkout of the UMU will be done during the second quarter of 1980. All circuit controls, valves, and sensors will be checked on the UMU prior to activation. Tank vents will be closed and nitrogen slowly supplied by the UMU to the TSU while simultaneously monitoring the pressure sensing and control system. The nominal set pressure in the TSU ullage space is 10.0 cm (4 in.) of water gage. Signal under- and over-pressure limits will be checked during the pressurization. As pressure increases, various check points such as low-pressure transducer readings and low-pressure switch activations will be monitored. As the pressure increases past the nominal set value, the high-pressure switches and controls will be monitored. When the UMU is proven to be operating according to specifications it will be further checked by venting the TSU through the manual vent system and pressurizing the UMU independently through the emergency pressurization system. Both the UMU system and the emergency ullage pressurization and venting system will be actuated several times to establish repeated readings and determine hysteresis.

Fluid Maintenance Distillation Unit

The fluid maintenance distillation unit will be checked during an opportune time in the checkout period. The fluid maintenance unit system checkout will consist of monitoring the actions of all valves, pressure regulators, and controllers, and operating the unit in the manner for which it was designed.

Auxiliary System

Auxiliary systems, such as automatic venting and bleeding, seal steam, and hot standby, will be checked out during the integrated system checkout phase. These units will be operated and monitored at an opportune time. Performance will be compared with the initial engineering estimates. The firex water system and fire detection system will be checked out prior to heating of the fluid past its flash point.

Integrated Subsystem Checkout

Initially, TSS checkout will interface with the Master Control, and provide signals on all circuits at the subsystem level to check response in the Master Control program. Similarly, signals generated by the Master Control will be monitored at the appropriate component to verify operation as desired. It is assumed that a checkout procedure will be programmed into the Master Control that will enable verification of the operation of all components as predicted. Checkout procedures will include simulated operation of the components with verification determined by audible, visual or signal detection through the portable checkout module. Complete system operation will be replicated with control transferred back and forth between Master Control and the plant operator. All emergency signals and anomalous statements will be checked and displayed with simulated signals provided by the portable checkout modules.

After checking out and verifying satisfactory operation of the TSS independently and in concert with the Master Control the subsystem will then be placed in operational standby; ready to receive heat at any time. TSU bed conditioning can begin at any time thermal energy is available. It is expected that this procedure will take approximately one to two weeks, and can be done as energy becomes available from the collector and receiver subsystems. During bed conditioning the SRE system test showed that a relatively large amount of water may be driven off, and this must occur slowly to avoid overpressurizing the TSU tank. It is anticipated that initial tests will be at partial power which would be an opportune time for bed conditioning purposes.

During the SRE thermal storage subsystem checkout, frequent cleanup of the in-line filters was needed during bed conditioning to purge the system of silt. These filters are provided in duplicate for the Pilot Plant as in the SRE tests so that conditioning need not be interrupted for silt removal. During this period silt removal will be recorded as to time and amount and correlated with the total amount of fluid that has traversed through the filters between cleanups. When it is evident that silt and water removal is complete, the system will be available for complete operation and can operate at maximum conditions as designed at any time.

As the energy level increases in the TSU, continuous monitoring of the temperature and pressure transducers will occur for verification of normal operation. In addition, daily monitoring will occur to detect possible fluid or water leaks in the system. A complete log will be kept of the operational time of the units to provide a prediction of scheduled maintenance. When the energy level in the thermal storage unit reaches operational range, the system will be checked out over the complete operational heat flow and fluid flow band.

As the total Pilot Plant begins to function as a completely integrated unit, all operational characteristics will be documented and tracked for reproducibility from day to day. Heat losses in the system will be monitored and operational characteristics will be established and included in the operations manual. During the last quarter of 1980 it is expected that the solar thermal power plant will become completely operational and demonstrate all operating modes. During this period the operations manual will be updated and will include all operational characteristics of the thermal storage subsystem. On 1 January 1981 the thermal storage subsystem will be completely activated and checked out and ready for a 24-hr/day operation and the 2-yr development program.

Portable Checkout Module

The TSS portable checkout module will contain those electrical sensing and power circuits necessary to check out subsystem circuit continuity, monitor signals and provide simulated signals, where necessary, to verify operation of all instrumentation and controls.

In addition, a second module may be included, depending upon the size, to provide and verify pneumatic signals. The thermal storage subsystem will be designed for rapid access to pneumatic and electrical transducers to facilitate initial checkout as well as subsequent periodic checkout during the life of the plant. The portable checkout module will (1) be capable of simulating signals from all instrumentation sensors to recorders or readout elements, (2) monitor commands by the operator or Master Control, and (3) provide initiation signals as required for operating control components.

Design Requirements

All sensors and checkout points will be uniquely identified. Where possible, the thermal storage subsystem will be designed with electrical and pneumatic circuits, sensors, relays, junction boxes grouped in such a manner that checkout will be rapid and junction points will be readily identifiable and accessable. The portable modules will be designed to be self-sufficient and will include safety features that will prevent over-ranging of initiated or received signals. Prior to checkout and periodically during the life of the plant the portable checkout module will be checked completely in the instrument calibration laboratory.

5.3.4 Quality Assurance

Quality assurance will consist of the following tasks:

1. Source inspection will be used on critical characteristics that become concealed prior to shipment. Some examples are:
    - Heat exchanger material certification
    - Heat exchanger tube to tube sheet attachment
    - Heat exchanger intercircuit test
    - TSU material (some inspection for laminations)
    - TSU prefabrication (welder qualifications and x-rays of weld deposits)

2. All purchasing requirements will be inspected prior to preparing a receiving report which is authorization for payment.
3. All system characteristics will be noted on the quality control set of the construction drawings or on inspection check sheets prepared by Quality Assurance. All variations will be identified. Acceptance of a variation will require approval of the design and using agencies.
4. Pressure- and leak-testing, motor rotation testing, electrical continuity testing, and item function testing will be performed by the mechanical contractor, but under the direction and monitoring of an inspector.
5. Examples of characteristics to be inspected and techniques to be used are as follows:

Characteristic	Inspection Methods
Welds-Full Penetration	Operator Qualification Test, ASME Section IX Procedure Qualification Test, ASME Section IX Appearance, ASME Section IX: Test, ANSI Section 31.3; X-Ray, A.P.I. 620 and ANSI Section 31.3
Welds-Fillet	Same as above, but substitute dye penetrant inspection for x-ray
Pipe Bends	Visual, section and ultrasonic
Concrete	Slump test
Soil Compacting	Testing in accordance with ASTM standards such as D1557

6. A Quality Assurance representative will record the results of all tests, maintain test records, and assure that all test conditions and results have been achieved. Acceptance of all tests and completion and inspection of the facility constitute acceptance of the contractor's work. The equipment is then ready for activation and checkout. At this point, the builder's participation is limited to warranty work and scope changes.

5.4 MAINTENANCE PLAN

This section provides the maintenance plan for the various items included in the 10-MWe Pilot Plant TSS.

The plan is based on experience gained by Rocketdyne from the thermal storage Subsystem Research Experiment (SRE) program conducted at the Rocketdyne Santa Susana Field Laboratory in 1976, from manufacturers' literature, and through experience with similar hardware in rocket engine systems Rocketdyne has provided over the past 25 yr.

The plan is based on rapid removal and/or repair in place which will be facilitated by using components that are readily repairable and are located with adequate access and clearance.

5.4.1 Summary Schedule

Table 5-1 is the summary schedule for periodic maintenance and inspections required to detect problems for unscheduled maintenance. It shows the suggested frequency with a reference code for the work to be performed on the listed items of the TSS. It is expected that this schedule will be modified and updated with experience gained during operation of the Pilot Plant. Maintenance experience with the Pilot Plant will be directly applicable to the Commercial Plant.

Table 5-2 is an explanation of the code items in Table 5-1. Table 5-3 includes failure modes, corrective action, and repair/replacement times. These do not necessarily represent unscheduled outage times since repair and replacement may be accomplished when the system is normally dormant or during other scheduled outage.

Components requiring repair or replacement and discovered by routine inspection or by observation of erratic or abnormal behavior will be reported to the maintenance manager or supervisor and logged for corrective work scheduling. Emergency situations involving health and safety will receive immediate attention with appropriate action (i.e., deactivating all or a portion of the TSS).

Table 5-1

TSS MAINTENANCE SUMMARY SCHEDULE*

Item	Maintenance Period and Maintenance Action					
	Daily	Weekly	Monthly	Semiannual	Annual	Other
1 Thermal Storage Unit (TSU)	A				B, C, D	
2 Ullage Maintenance Unit (UMU)	A	E, F	G		B, C, D	
3 Fluid Maintenance Unit (FMU)	A	E	G			
4 Pump Units (Including Motors)	A		G		H	
5 Heat Exchangers (TSH and SG)	A		A		H	
6 Desuperheater	A		A		H	
7 FMU Filters	A		J			
8 Valves						
• Manual	A	K			K	
• Control	A				L	
• Relief	A				L	
9 Instrumentation and Transducers						
• Flowmeters			A		L	
• Pressure and Delta-p Transducers			A	L		
• Temperature Transducers						
• Thermocouple			A			
• Resistance Bulbs			A	L		
10 Control Subsystem				M		
11 General Maintenance: Descaling, Cleaning, Painting, Etc.						Every 3 yr or as Required

*See Table 5-2 for Code Explanation

Table 5-2

TSS MAINTENANCE INSPECTION AND REPLACE/CALIBRATE CODE

A. Walk around visual and audible check for leaks, mechanical, and electrical abnormalities.

B. Check bolt torque.

C. Check tank settling or distortions.

D. Check for weathering and insulation spalling.

E. Draw off waste fluids and submit a sample for analysis.

F. Check oil levels for pumps, motors, compressor, etc. Replenish as required.

G. Lubricate bearings, shafts, etc. that require periodic lubrication.

H. Check for scaling and corrosion, primarily steam and/or water systems.

J. Inspect and clean FMU filters as required. (Requirement for cleaning will be as indicated by differential pressure readouts.)

K. Check manual valves for verification of open/close operation.

L. Service, calibration, and proof test changeouts (applies to valves and transducers).

M. Check for dust, sand, corrosion, connector integrity.

All major work, except where time does not permit, as in an emergency, will be coordinated and scheduled for corrective action with the Facility Operation Manager. A major criteria for proper maintenance will be continuous monitoring of instrumentation readouts-flows, pressures, and temperatures. Much of the maintenance work performed will be as a result of data trend analysis which will pinpoint incipient problems. It should be

Table 5-3

MAINTENANCE DETECTION AND CORRECTIVE ACTION GUIDELINES (Page 1 of 2)

Item	Expected Failure Mode	Corrective Action	Time to Repair/ Replace (Hr)*
TSU	Insulation, spalling, weather cover loosening, external leaks	Repair or replace as necessary	2 to 24
Controls (Controllers) Actuators	Check mechanical linkages, shaft seals, and hydraulic or pneumatic pressure source leaks	Repair linkages, seals, fix leaks, verify actuating fluid pressures, replace EP converter or motor actuator	3 to 12
Valves			
Large Shutoff (12 in.)	Shaft seal leaks	Adjust stem packing,	2
		replace packing and	4
	Seat seal leaks	readjust lap or replace seats	8
Small Shutoff,	Shaft seal leaks		
		Adjust stem packing	1
Gage, Transducer, Isolation	Seat seal leaks	Replace packing	2
		Replace valve	2
Remote-Operated and Position	Same as large	Same as large	Same as large
Shutoff + Actuator Service and Control System Components	Actuator control pressure leaks Sticking Stem Rupture diaphragm	Replace components Adjust valve control components	6 4
Pressure and Temperature Transducers	Internal leaks, open /short circuit, out of calibration	Remove and replace, recalibrate, send to repair depot	1 (in field)

*Not outage time, system may be all or partially active

Table 5-3

MAINTENANCE DETECTION AND CORRECTIVE ACTION GUIDELINES (Page 2 of 2)

Item	Expected Failure Mode	Corrective Action	Time to Repair/ Replace (Hr)*
UMU	Degradation of soft goods	Replace seats/seals	2
	Pressure out of calibration	Reset, in place	1
		Replace component	6
Pumps			
Large	Shafts seals	Adjust or replace seals	4
Charging and extraction	Bearings	Pump removal, critical component replacement	2 days
	Electric starters		
	Electric motor	Replace and align	8
Small	Same as large	Replace component or pump	4
Gauges			
Pressure	Tube assembly failure	Remove and replace	2
Temperature	Drive coupling loose	Tighten and recalibrate	2
Differential Pressure	Capillary tubing damaged		
	Bellows failure		
Flowmeters	Erratic readings, Restricted flow Out of calibration	Remove and clean, Recalibrate, Remove and replace	4 for all
FMU	Excessive pressure drop	Replace or clean filter basket. Check gauges, clear gage lines	3
Heat Exchangers	Internal circuit leaks, external leaks	Pull heads, repair, plug or replace tube, pull tube bundle, clean and repair, replace flange gasket	2 days to 4 weeks

*Not outage time, system may be all or partially active understood, while this is an operational function carried on by operating personnel, a prompt reporting to maintenance personnel will avoid many problems.

5.4.2 Maintenance Procedure and Component Procedure

Proper maintenance is essential to sustain the TSS in a continued state of operation, high efficiency, and safety over its 30-yr life. Maintenance includes all actions taken to retain an item in a specified condition by providing systematic inspecting, detecting, and servicing for the prevention of incipient failure and the action taken to restore an item to a specified operational condition. This includes fault isolation, item replacement, repair, and checkout. To accomplish this, two types of maintenance are considered. They are scheduled maintenance and corrective maintenance and are defined as follows:

1. Scheduled Maintenance—Actions performed to retain an item in an operable condition by systematic inspection, detection, prevention of incipient failures, replacement of life/cycle limited components, adjustment, calibration, cleaning, and lubrication. Scheduled preventive actions will be minimized during operating periods and emphasized during nonoperating periods of portions of the thermal storage subsystem to sustain equipment/system availability. Servicing activities will be minimized and conducted on a noninterference basis.
2. Corrective Maintenance—Actions performed to restore an item to a satisfactory condition by correction of known or suspected malfunctions or defects which have caused degradation of the item below the specified performance level. Corrective maintenance consists of repair, replacement, checkout, and verification of repaired equipment. It is performed as a result of condition monitoring or unexpected/unpredicted failure or malfunction.

The following definitions have been used in describing maintenance requirements:

Limited Life. Those items which are removed and replaced by a like item, at scheduled intervals, regardless of their condition, are classified as limited life replacement items.

Item. Any level of hardware assembly such as a system, segment of a system (element), a subsystem, equipment, component, or a part.

Repairable Item. An item in unserviceable condition which can economically be restored to serviceable condition.

Spare Part. An item capable of support and replacement which is required for the maintenance, overhaul, or repair of the article for which it is provisioned. Spare parts are classified as standard or peculiar parts. Standard and peculiar parts may be further classified by characteristics or usage as bulk or common items.

Scheduled maintenance (i.e., those portions dealing with systematic inspection, detection, cleaning, and lubrication) when possible, will be performed during the active period of the subsystem from 6 AM to midnight daily. The balance of scheduled maintenance, and the corrective maintenance, except for unexpected subsystem failure or malfunction, will be performed during the dormant period of the subsystem—midnight to about 6 AM daily and within scheduled shutdown periods where possible. Repairs resulting from the unexpected failure or malfunction will be performed on an expedited basis to minimize downtime.

Specific maintenance procedures for individual components and subsystems are described in the following sections. Detailed corrective procedures and instructions will be developed and included in a maintenance and repair manual that will be written during the Phase 2.

Procedures described herein are tasks that will be accomplished in coordination with, but without request of, the plant operating personnel. In addition to these scheduled procedures, plant operation will be continuously monitored for out-of-spec performance. When component or subsystem deviates from established norms, operations will notify maintenance and corrective procedures will be established that will provide rapid return to establish performance levels with a minimum of outage time.

5.4.2.1 Thermal Storage Unit (TSU)

The TSU is basically maintenance-free. A daily visual walkaround check will be performed to note existence of any small seepage or leaks through connections, bolt flanges, etc., that if left unattended could result in a potential fire hazard. Where necessary, seeping connections will be tightened and cleaned of heat-transfer fluid residue. Insulation will be replaced if damaged from excessive leakage or environmental effects. An annual walk-through will be conducted to check bolt torquing, possible tank settling, weathering, and insulation spalling.

5.4.2.2 Ullage Maintenance Unit

The UMU consists of nitrogen storage and supply and storage tanks, valves, a compressor heat exchanger, indicators, controllers, sensors, gages, vents, etc. As such, it will require all of the maintenance normally required for those items. Daily inspections will include examination for leaks on the compressor, flanges, valves and fittings, and check of fluid level in the compressor intake manifold sump and gas precipitate storage tank. Draining of precipitate will be done when at the upper operating level. It is expected that draining will be required once per week. Operational personnel will draw off the organic fluids and deliver to the chemical laboratory for analysis. On a weekly basis the lube oil level for the compressor should be checked and replenished as required. Annual maintenance will include replacement of drive belts and hoses plus cleaning of the sight gages on the compressor intake manifold sump and ullage gas storage tank.

5.4.2.3 Fluid Maintenance Unit (FMU)

The FMU consists of two subassemblies, a pair of full-flow in-line filters and a distillation subsystem.

Maintenance on the two filters (TFF-1 and TFF-2) in the charging loop will consist of removal and cleaning of the filter element when operations personnel notify maintenance that pressure drop limits have been exceeded. The charging loop filters have been sized to operate over extended periods of time without cleaning. SRE tests conducted during Phase 1 indicate that once the initial particle removal is completed, time between cleaning is on the order of once per month. During normal operation one filter is on-line and the second filter is clean and on standby. Switching filters will be done manually. SRE tests have shown that filter buildup occurs slowly and filter cleanout can be scheduled 24 hr in advance. This fact, together with the use of two full flow filters, will allow adequate cooldown time to a safe handling temperature before the filter is opened and cleaned. It is expected that 24 hr cooldown will result in a fluid and filter temperature below 43° (110° F) which will be adequate for safe handling.

The distillation unit consists of pumps, valves, a backflow regulator, evaporator, condenser, and instrumentation. As such, it will require all of the maintenance normally afforded to those items. In addition, it will require a weekly inspection for vacuum pump oil level, loose belts, abnormal noise, and emptying and cleaning of the fluid waste from the waste collector. Operating personnel will draw off samples of the fluid waste weekly for laboratory analysis. In addition, weekly inspection should be performed on the evaporator for proper oil level in the reservoir and for shaft seal condition. Bearings should be lubricated monthly. Expected life of shaft seals is 2 yr minimum, and bearing expected life is 5 yr minimum for the evaporator according to the manufacturer.

5.4.2.4 Fluid Circulation Pumps

The fluid circulation pumps consist of dual operating units mounted in parallel, each one capable of providing 50% of the maximum flow of the thermal storage charging and extraction network. In the charging network, the pumps are powered by two-speed electric motors operated together or independently providing control over a wide range of flow conditions to minimize parasitic power consumption. In both the charging and heat extraction loops, one or two pump/heat exchanger units are in operation depending upon the energy flow requirements. The pumps are arranged with the heat exchangers so that if one pump is in operation, either one of the heat exchangers can be used for transfer of thermal energy. This arrangement provides a high degree of reliability, because the plant can operate and perform a partial function with one pump/heater combination if the other is out of commission for repair. Pump inspection should include the fluid pump as well as the electric motor drive. Daily inspections will be required to check for leakage, proper lubrication, and proper flow of cooling water through the bearings and seals, as well as the presence of excessive noise or vibration. The pumps and motor drives should be relatively maintenance-free for a minimum of 5 yr after installation. Failure of major components such as bearings and stuffing box seals usually do not occur precipitously which allows scheduling for maintenance during major outage. Annual maintenance on pumps and related components will include checking of bolt torques, tightening connections, replacing gaskets and hoses as needed. Replacement of gaskets, seals, and bearings for the most part can occur with a minimum of downtime without removing the pump and the electric motor drive.

5.4.2.5 Thermal Storage Heat Exchanger

Heat exchangers in the TSS are dual units mounted in parallel and matching the circulation pumps. The heat exchangers are conventional shell and tube; no moving parts. Separate exchanger sets are used for charging and heat extraction. Daily maintenance procedures will involve a walk-around inspection observing the condition of the insulation as well as noting any fluid heat leaks or steam heat leaks from instrumentation ports and the flanged bolted joints. Tubes use the U-tube configuration which provides long life with the daily thermal cycling. It is expected that 5 to 10 yr of operation should occur before disassembly, cleaning, and replacement of any of the tubes is necessary. Two sources of possible maintenance exist for long-term operation. Tube corrosion on the water side is a possibility and may require that tubes in the heat exchanger be replaced periodically. A second source of maintenance is the possibility of hard carbonaceous deposits on the fluid side of the tubes which will inhibit heat transfer. Monitoring of cross flow leakage in the heat exchanger is essential to detect the extent detrimental to the operation of the rest of the system. Cross flow that is most detrimental to the remaining portions of the system involves leakage of the heat-transfer fluid into the water side of the water steam network. Heat-transfer fluid droplets in the steam would result in severe erosion of the turbine blades. A more natural way for leakage to occur is from the high-pressure water side into the low pressure transfer fluid. Although relatively large amounts of water will strain the capability of the UMU, water in itself is not considered detrimental to fluid life. Monitoring the vented gases in the UMU will provide information as to possible leakage in the heater exchangers. When leakage develops or whenever carbonaceous deposits seriously deteriorate the heat-transfer properties of the heat exchanger, the U-tube bundle will be removed for repair and mechanical cleaning of the inner and outer portions of the tubes. Because of the size and weight of the heat exchanger this will require from several weeks up to 1 mo. It is not expected that these conditions will develop precipitously so that maintenance may be scheduled and integrated with downtime scheduled for the remaining portion of the plants, (i.e., mirror cleaning/replacement, turbine adjustment, repair, receiver replacement or repair, etc.)

5.4.2.6 Desuperheater (DSH)

The DSH is basically a tank with a water spray nozzle in it to lower the incoming superheated steam temperature to the proper value (343°C) before entering the thermal storage heater. Scheduled maintenance will include daily inspection for leaks in the steam and high pressure water circuits. This unit is expected to be maintenance-free except for the possible clogging of the water spray nozzles due to chemical deposits or dirt in the water system. This condition will be detected by operating personnel through monitoring the desuperheater operating conditions. When the desuperheater is approaching out-of-spec performance the operating personnel will notify the maintenance personnel and arrangements will be established to clean the desuperheater during a normal outage period.

5.4.2.7 Control Valves

In the thermal charging and heat extraction flow loops, control valves are designed to permit inplace service during the dormant period. In the auxiliary fluid flow, loops valves are smaller and may be repaired in place or replaced with spare valves that will be stocked in the maintenance equipment laboratory. Expected failure modes are stem or seat leakage due to worn seats or worn or loose stem packing and seal seals or wear or damage from foreign material on the valve seats. The control valves are designed with removable bonnets so that when the complete subassembly has to be exchanged, rapid replacement will be possible. The complete bonnet assembly as well as seats, stem guides, stem packing, body seals, and flange gaskets will be stored to provide ready availability and minimum downtime.

5.4.2.8 Manual Valves

Manual valves should be inspected daily for stem and connecting flange leakages and actuated monthly for verification of close/open operation. When leakage becomes evident the appropriate parts may be replaced or a complete bonnet assembly may be installed from the spare parts inventory.

5.4.2.9 Relief Valves

Relief valves should be inspected daily for leakage. Monthly scheduled inspection will include operation of the relief valves and checking of pressure relief set points.

5.4.2.10 Controls Subsystem

The control system consists primarily of electronic and electrical functions between the master controller and thermal storage subsystem control panel in the control room and the specific units to be controlled, (i.e., valves and actuators). The control system basically is maintenance-free as long as the equipment is kept clean and dry. Normal daily operational procedures will be to monitor transducers, check redundant readings, to provide verification of control and sensor operation. When a malfunction is evident the operations personnel will contact maintenance personnel and arrange for replacement or repair of the unit at the earliest possible convenience. Scheduled maintenance will include a monthly inspection of all junction and terminal boxes to establish cleanliness standards and detect visual defects.

Repair of transducers may occur within the maintenance shop if adequately equipped or may be shipped back to the manufacturer for rebuilding, if possible. Experience at Rocketdyne indicates that critical sensors which are used to accurately control flows, pressures, and temperatures should be recalibrated every 6 mo to ensure proper functioning.

5.4.2.11 Instrumentation

Instrumentation can be best checked by operations personnel who may note abnormalities during daily operation. A 6-mo calibration period is scheduled for all critical control transducers.

Since the control system is essential to functional operation of the thermal storage subsystem a high proportion of spares will be stocked to provide rapid replacement and repair.

5.4.2.12 Equipment and Manpower Requirements Summary

Equipment and manpower skills and type for servicing the thermal storage subsystem are identical to what is required in a modern steam powerplant. It is expected that a well-equipped shop and instrument repair and calibration laboratory will be included on-site. The shop should contain conventional cutting, welding, machining and grinding equipment plus valve seat and plug lapping equipment for repair and servicing of valves, transducers, motors and pumps. A stock of bulk materials including seal compounds, electric wire, insulation, weather covering, and fastening hardware will be available. Most maintenance will be accomplished with small forklifts, pickup trucks, and auxiliary power equipment. Major equipment servicing such as heat exchanger tube bundle removal and large pump and motor removal will be at very infrequent intervals allowing the use of rented or borrowed equipment when needed for this purpose. Manpower requirements for the thermal storage subsystem are identical with respect to skills and crafts of a conventional fossil fuel fired electrical powered generation station.

Maintenance manpower requirements are summarized in Table 5-4 for three different levels of maintenance procedures. Level 1 is representative of the daily effort involving visual inspections for leakage, pump bearing failure, and attention to general cleanliness. Level 1 may also include minor adjustments, replacement of sight gages, and in-place short functional checks. It is estimated that Level 1 maintenance will take from 4 to 8 hr per shift and will be done during day or early evening hours when the system is functioning.

Level 2 involves the removal, repair, and replacement of pressure transducers, temperature transducers, flowmeter sensors, large valve components, complete small valve assemblies, line gaskets where leakage is apparent, small pumps, motors, and heat exchangers, and minor adjustment and replacements of insulation and insulation weather covering. It is expected

Table 5-4

TSS MAINTENANCE MANPOWER REQUIREMENTS

Level	Frequency	Men Per Shift*	Duration (Hr)	Walk	Maintenance Task (Typical)
1	Daily	1 (F)	4 to 8		Walk around visual and audible inspection, minor adjustment/replacement, unit in place functional check, general cleaning.
2	Monthly	2 (F, S)	4 to 16		Remove, repair and replace sensors, large valve components, small valves, line gaskets, small pumps, motors, heat exchangers, minor insulation.
3	Annual	4 (F, S)	2 days to 4 weeks		Remove, repair, and replace large pumps, motors, heat exchangers, line segments, major insulation repair or replacement.

F — Field personnel

S — Shop personnel

that level 2 effort will occur on the order of once per month and require two men 4 to 16 hr to complete. This time may not necessarily all occur during a single shift depending on the spare parts availability, the amount of shop work required, and the shop backlog between the removal and replacement. Level 3 tasks are those involving the removal, repair, and replacement of major equipment items such as large pumps, electric motor drives, large heat exchangers, large line segments of assemblies, and major insulation repair or replacement. Based on the type of equipment it is estimated that Level 3 effort can be expected to occur on an annual basis and will require four men per shift with an elapsed time in the order of 2 days to 4 weeks. Level 2 and Level 3 tasks are split between field personnel and shop personnel. Field personnel will remove and replace items of equipment in the thermal storage subsystem which is dormant between midnight and 6 AM. However, shop personnel will work on the equipment that has been removed during the normal day work shift.

5.4.2.13 Spares Provisioning

The required initial spares provisioning plan is summarized in Table 5-5. Stocking of these spares will allow rapid and economical repair with a minimum of outage time. Since all large valves will be welded in place, repair will be accomplished by replacement of major subassemblies (such as complete bonnet assemblies). Major control valve downtime will be minimized by tools that facilitate rapid removal and repair in place.

Rocketdyne experience has been that the most cost-effective method of providing a minimum amount of downtime is to provide a high percentage of transducer/sensor spares and to schedule periodic replacement and recalibration.

Spares for the main pumps and electric motor drives will consist of replacement bearings, seals, gaskets, and the necessary electrical hardware. During the 30-yr lifetime it is expected that the large pumps and electric motors can be repaired in place. Small motors and auxiliary equipment are usually most economical to replace completely.

Table 5-5

PILOT PLANT THERMAL STORAGE SUBSYSTEM INITIAL SPARES PROVISIONING

Item	Scheduled Maintenance Cycle	Probable Condition Requiring Servicing	Servicing	Required Spares[1]
Transducers/Sensors				
Pressure	6 mo	Out of calibration	Lab repair/calibration	50%[2]
Thermocouple	6 mo	Out of calibration	Lab repair/calibration	10%
Temperature Bulbs	12 mo	Out of calibration	Lab repair/calibration	50%
Electronic Controllers	12 mo	Dirt, bad connection	Clean and check in place	
Flowmeters	12 mo	Sticking, leaking, ball worn	Lab repair/calibration	1 each type and size
Sight Gages	12 mo	Sticking, leaking, bearings worn	Lab repair/calibration	1 each type and size
Heat Exchangers (change and extract)	5 yr or	Cleaning	Remove U-bundle and clean	Gaskets
	10-15 yr	Tube replacement	Replace tubes	Tubes, order when ready
Valves				
Control (large)	12 mo	Out of calibration sticking, leaking, or worn stem	Shop repair/calibration	1 bonnet assembly each type plus 10% trim, seats, seals, actuators and EP converters
Manual (large)	AR	Sticking, leaking	Shop repair, bonnet assembly	1 bonnet assembly each type and size
Small (≤ 2 in.)	AR	Sticking, leaking, worn	Shop repair, valve assembly	1 valve each type and size
Pumps/motors	AR	Worn bearings, seals, belts, gaskets	On-line or shop repair	1 each set plus bulk gasket material
Piping	AR	Leakage	Replace gasket and remount component if excessive thermal movement is causing leakage	1 gasket each flanged joint

(1) Integrated with other major subsystems

(2) Percent of thermal storage subsystem

AR - As Required

The large charge and extraction heat exchangers will be disassembled in place for cleanout and/or repair. The appropriate gaskets will be stocked and replaced as needed.

Piping gaskets and sealing compounds will be adequately stocked in all sizes to allow leaks and/or repair and replacement of line sections.

The spares provisioning list in Table 5-5 provides for the thermal storage subsystem independently. When detailed designs are completed during Phase 2, a master spares provisioning list encompassing all subsystems will be prepared. Where commonality exists, spares will not be duplicated. This will provide cost-saving, particularly in the areas of transducers, steam/water line gaskets, and small pumps and motors.

Section 6
SUBSYSTEM RESEARCH EXPERIMENTS

6.1 TECHNICAL ISSUES AND TEST OBJECTIVES

The subsystem research experiments (SRE) consisted of tests to provide data on system and components needed to establish a firm engineering basis for design of the Pilot and commercial plants.

SRE tests were chosen judiciously at the beginning of the present contract through the application of a verification analysis. The verification analysis provided an assessment of risk in critical technology areas that would impact performance, schedule, and/or cost. The need of SRE tests was determined from the availability of established engineering analysis or of demonstrated commercial products.

6.1.1 Verification Analysis

The verification analysis conducted at the beginning of this program considered all technology areas concerning all components and subsystems of the thermal storage subsystem. The SRE test program evolved from detailed consideration and evaluation of 37 key line items weighed against the possible risks of performance, cost, and schedule impact. These assessments indicated that the primary tests for the SRE should involve large-scale thermocline establishment and storage characteristics, high-temperature stability of candidate heat-transfer fluids in contact with rock and heated metal surfaces, and tank stresses induced by interaction with the rock bed. Table 6-1 summarizes the key issues and their size and scale relationship between the SRE, Pilot, and Commercial Plants. Details of the verification analysis are given in Reference 6-1.

The primary efficiency and economy of the subsystem is determined by the thermocline in the thermal storage unit. The performance of the rock bed, with respect to energy extraction efficiency, hold capability, fluid distribution over wide flow ranges, heat loss, and rock bed filling to achieve

Table 6-1

THERMAL STORAGE SUBSYSTEM KEY ISSUES

Issue	Technology Base at Start of Phase I	SRE Test Objectives	Pilot Plant Requirements	Commercial Plant Requirements
Efficiency:				
Thermocline	10% full-scale height Full-scale fluid velocity	100% full-scale height, * Full-scale velocity	Full-scale height, Full-scale velocity	Full-scale height, Full-scale velocity
Heat Loss	Laboratory-scale size	Full-scale heat-transfer coefficients and temperatures	2-3% Loss	1-2% Loss
Economy:				
Rock bed concept	Laboratory scale	3.2m (10.5 ft) dia, 13.3m (43.7 ft) h	15.2m (50 ft) dia, 13.4m (44 ft) h*	27.6m (90.5 ft) dia, 18.3m (60 ft) h*
Fluid type	Caloria HT43, Therminol 55 and Therminol 66	Caloria HT43 to 316°C	302°C Life	316°C Life
Fluid maintenance	Filter Technology	Operational filtering	Filtering and refining	Filtering and refining
Pilot Plant Design:				
Tank Stresses	Design analysis	Model test plug analysis (30-yr life substantiation)	30-yr life	30-yr life
Transients	Limited	All operating modes	All operating modes	All operating modes
Controls	Analysis	Operational simulation	Fully operational	Fully operational

* Design revised since program initiation so that SRE thermal storage unit is 100% of Pilot Plant unit height and 73% of Commercial Plant unit height.

low void fractions, needed to be established for the Pilot Plant preliminary design. For the most part, these characteristics were determined in an earlier Rocketdyne-sponsored program using small-scale hardware. However they needed to be verified on a larger scale, which was accomplished by the SRE subsystem test program. The SRE subsystem tests extended the thermocline verification to a unit having the actual velocities and temperatures, and approximately 100 and 80% of the height of the Pilot Plant and Commercial Plant units, respectively.

Another critical issue in the Pilot Plant design is to ensure acceptable working stress levels in the thermal storage tank. The thermal storage unit is basically a pebble-bed heater, from a structural standpoint. Pebble-bed heaters have been used for many years for heating a wide variety of gases. The primary area of uncertainty is the prediction of tank life with possible interaction occurring between the tank wall and rock bed as a result of thermal cycling. At the beginning of the program, satisfactory design basis was not available that would apply to the particular case (e. g., thermal cycling up to 317°C (600°F) in a large tank filled with rock). Analyses made in the course of establishing the Pilot Plant preliminary baseline design indicated that the stress that is produced at the bottom of the tank by repeated thermal cycling of the rock is open to substantial uncertainty, which required tests to resolve. Strain measurements were made on the SRE subsystem tank to correlate analysis and provide a basis for later extension to the 30-yr life requirement. If stress levels in the basic tank were high, then conservative designs would be used.

Another important issue identified in the verification analysis was the life of the heat-transfer fluid and the availability of the fluid at low cost. Investigation of commercial fluids available indicated that a natural petroleum-base heat-transfer fluid represented by Caloria HT43 is the best baseline candidate for achieving this goal. However, other fluids are available on the market.

The fluid must meet a life of 30-yr, with reasonable refurbishment and replenishment, to satisfy the Commercial Plant requirements.

Although the use of the rock bed reduces the fluid inventory cost by a factor of 3 to 4, it is still desirable to retain the fluid and refurbish it as needed to minimize overall operating cost. The FMU performs this function. To minimize program risk, laboratory tests and analysis were conducted to identify the type of fluid maintenance and procedures required to minimize costs for a 30-yr life.

All information available prior to contract inception (manufacturers' literature as well as short-duration tests) indicated that a 30-yr fluid life could be expected operating in the 302° to 316°C (575° to 600°F) range but neither life nor refurbishment requirements had been confirmed by controlled tests.

Preliminary design analysis to establish the baseline characteristics of the Pilot Plant indicated that thermal losses through the tank wall are a factor in establishing tank size. Because of the natural loss of heat at the tank wall, the outer bed temperatures adjacent to the wall drop below acceptable levels for required energy extraction. Tests of a scalable TSU would be needed to establish heat-loss characteristics. Transient operation of the SRE model subsystem is important for adequate design and analysis. Control of the thermal storage subsystem was considered to be well within the state-of-the-art; however, transient response of the TSU rock-bed combination was an unknown quantity.

The UMU using gaseous nitrogen to control the inert atmosphere without excessive nitrogen gas consumption, could be designed using established engineering procedures. However, the SRE subsystem tests were to substantiate the gas flow requirements.

In assessing the Pilot Plant design requirements in other areas, it was determined that, for most subsystem elements, the use of proven commercial hardware, together with established engineering design procedures, would result in a conservative and predictable Pilot Plant design approach.

Heaters, pumps, valves, controls, piping, connecting joints, and insulation as well as heat exchangers have been in use for many years at high temperatures using the proposed heat-transfer and equivalent fluids. Completely automatic steam generator systems using heat-transfer fluids operate continuously unattended in hospitals, apartment houses, and in a wide variety of commercial applications. These installations are highly regarded for their demonstrated safety, low maintenance, and high reliability of operation. However, more experience was needed to establish a 30-yr life for the fluid when operating near limiting peak temperatures. Maintenance of the fluid will be required to retain the necessary physical and chemical properties.

6.1.2 SRE Requirements

Requirements for the SRE tests are summarized in Table 6-2. The SRE program consists of the prequalification tests and the subsystem tests. The prequalification tests consist of fluid thermal stability and compatibility tests. Subsystem tests were designed to use a scalable (4 MWHt) TSU and complete thermal storage fluid charging and heat-extraction flow systems, with appropriate instrumentation to obtain extensive data for resolving the key issues.

Details of these tests are given in Sections 6.2 and 6.3.

6.2 FLUID TESTS

Laboratory tests were conducted (and some tests are continuing) to evaluate the high temperature (288° to 343°C (550° to 650°F)) thermal stability, material compatibility, and surface fouling of selected commercial heat-transfer fluids for extended periods of time. The tests have provided information on the rate of fluid replenishment required, the change of viscosity, the percent of high boiling material and the rate of fouling of heat-transfer surfaces as a function of temperature and time. Determinations have been made of the effect of the presence of materials likely to be used in the energy storage subsystem (rocks, stainless steel, and carbon steel) on these properties.

6.2.1 Past Data Base

6.2.1.1 Fluid Degradation Considerations

Degradation of a hydrocarbon heat transfer fluid can occur over time by two principal processes: pyrolysis (including thermal cracking and polymerization)

Table 6-2

SRE TEST REQUIREMENTS

Key Issue	Test Requirement
Heat-Transfer Fluid Thermal Stability	Long-term tests at maximum operating temperatures, 288° to 316°C (550° to 600°F) with most promising fluids.
Heat-Transfer Fluid Compatibility	Long-term tests at maximum operating temperature, 288° to 316°C (550° to 600°F) with most promising fluids exposed to primary materials (rock, sand, carbon steel, and stainless steel).
Heat-Exchanger Surface Fouling	Long-term tests at maximum operating temperatures, with heat-transfer fluid exposed to hot metal surfaces at 316° to 343°C (600° to 650°F)
Rock Bed/Tank Interactions during Thermal Cycling	Careful strain measurements during repeated cycle SRE tests with subsystem tank
TSU and Subsystem Operating Characteristics in all Basic Operating Modes; Thermal Storage Unit Heat Loss Characteristics	Test TSU with associated charging and extraction networks at rates representative of full-scale operating conditions

and oxidation (primarily from contact with air). The rate of pyrolysis depends upon the structure of the hydrocarbon; for an acceptable heat-transfer fluid, these reactions must be very slow in the desired temperature range. Catalysts for pyrolysis reactions, which include a variety of compounds, especially copper alloys, must be avoided completely in the system design. The air oxidation reaction rate of the hydrocarbons increases sharply with temperature; the rate is so rapid at 316°C (600°F) that a fluid in extensive contact with air would last only a few days. One of the types of products of air oxidation, peroxides, are effective polymerization catalysts. Some heat-transfer fluids contain antioxidant additives to inhibit the action of dissolved oxygen when the fluid is initially heated.

These pyrolysis and oxidation mechanisms are illustrated in Figure 6-1. Very low molecular weight compounds, either formed through cracking or initially present in the fluid, present no problems in moderate quantities. They merely cause a moderate increase in vapor pressure (which is normally only 25.5 KPa (3.7 psia) at 316°C (600°F) for "weathered" Caloria HT43 and 77.9 KPa (11.3 psia) for fresh fluid) and a decrease in viscosity. However, if the amount of volatile material becomes excessive, it would be necessary to withdraw some of it through the normal operation of the ullage maintenance unit.

The other products of thermal cracking are olefins, which can polymerize. Normally, the amount of thermal polymerization would be very small at the temperatures involved; however, contact with air and formation of even small amounts of polymerization catalysts could result in significant polymerization, which is potentially harmful. As the fluid degrades, the polymers may increase both in molecular weight and quantity. Unless the fluid is severely degraded, the polymers remain in solution and cause moderate increases in viscosity. Experience with Therminol 66 indicates that polymers remain in solution up to about 40-wt. percent and systems are operated routinely with about 30-wt. percent polymers (Ref. 6-2). However, the polymers are considered to be the precursors of fouling of the heat exchanger tubes by carbonaceous deposits (coking). Coking tendency in the heat exchanger is not accelerated until the fluid begins to degrade considerably. Severe fluid degradation could result in high-molecular-weight insoluble polymers that separate from the fluid as resins or deposits on heat exchanger tubes.

6.2.1.2 Caloria HT43

Some physical property data are available on Caloria HT43 from Ref. 6-3, but only limited information is available on its thermal stability and nothing is available on its fouling or coking properties. It is known that copper compounds can behave as cracking catalysts for Caloria HT43 and cause a rapid degradation in fluid properties. Fluid life prediction information related to polymer formation in Caloria HT43 has been given (Ref. 6-4 and 6-5) under its previous name, Humbletherm 500. However, Caloria HT43 contains an antioxidation additive which was not present in Humbletherm 500, so the polymerization rate of Humbletherm 500 could have been substantially higher than for Caloria HT43.

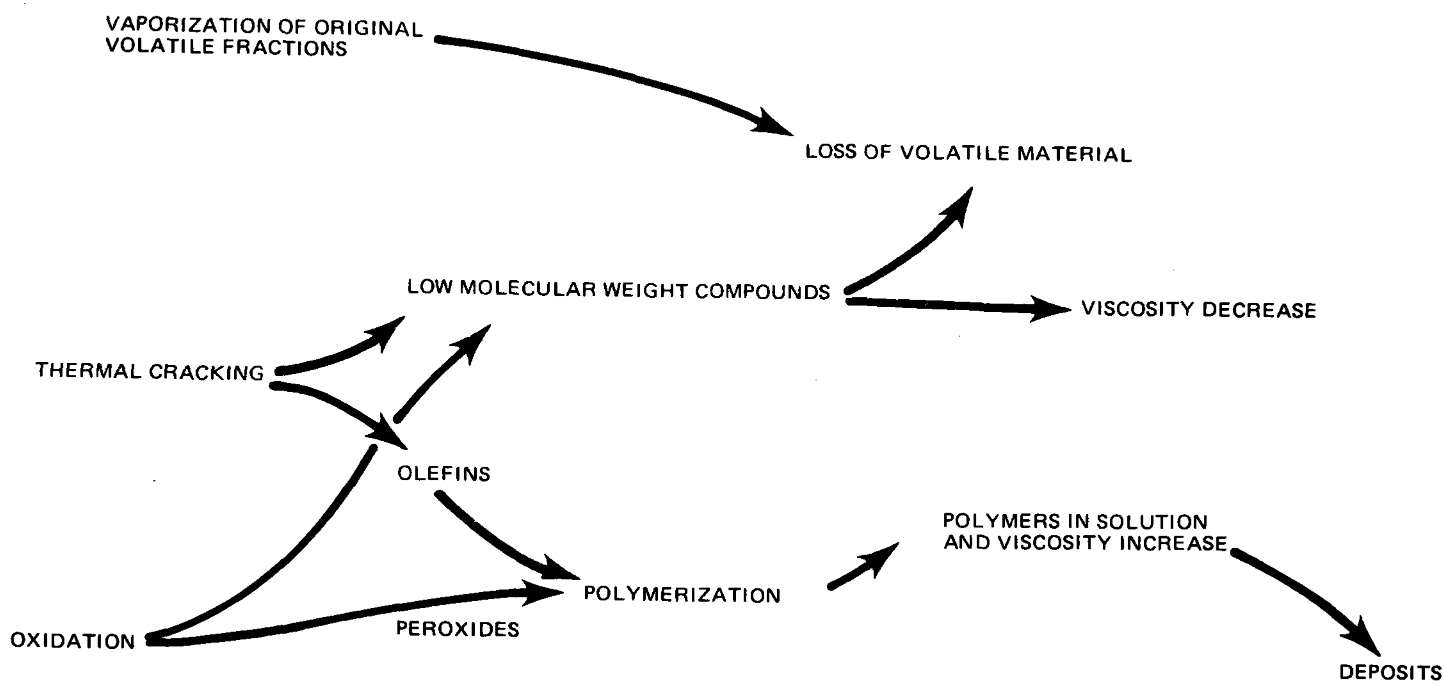


Figure 6-1. Fluid Maintenance Considerations

A number of heat-transfer loops have been operating with Caloria HT43 for extended periods of time. One application in Louisiana containing about 760,000 liters (200,000 gal), has been operating at 305°C (580°F) for over 6 yr without fluid treatment of any kind (Ref. 6-6). Continuing efforts are being made to obtain additional details on this and other commercial operations.

6.2.1.3 Therminol 55

Some data on the stability of Monsanto's Therminol 55 are available in the literature (Ref. 6-4 and 6-5). The experimental work (Ref. 6-4) deals primarily with the time required to accumulate 10% high boilers at temperatures ranging from 330° to 357°C (625° to 675°F).

6.2.1.4 Therminol 66

Data on polymerization of Monsanto's Therminol 66 are also available in Ref. 6-4 and 6-5. A great amount of test data has been accumulated on Therminol 66 as a result of an experimental program conducted by the Atomic Energy of Canada Limited (AECL) to develop an organic cooled and moderated reactor. Therminol 66 was designated HB-40 and OB-85 for the AECL work. The test data made available by contact with the AECL include:

A. Information on physical properties (Ref. 6-7)
B. Methods of analysis of the chemical and physical properties (Ref. 6-8 and 6-9).
C. Impurities and coolant quality tests (Ref. 6-8 and 6-9).
D. Analyses of thermal decomposition rates (Ref. 6-10, 6-11, and 6-12).
E. Fouling of heat-transfer surfaces (Ref. 6-2 and 6-13 to 16).

Since the decomposition rate data taken by the AECL were obtained in an irradiated environment, both pyrolysis and radiolysis occurred together. The claim is made, however (Ref. 6-10 and 6-11) that the pyrolytic and radiolytic contributions can be separated out and their interdependence is insignificant until high temperatures are reached. Pyrolytic decomposition was assumed to be a first-order reaction dependent only upon temperature and the fraction of nonhigh boilers. The data were obtained on a fluid containing 25% high boilers. The rate constant giving the best fit of the data between 300° and 400°C is:

$$K = \exp[25.80 - 43.255/(0.001987\ T)] \quad \text{(Ref. 6-10)} \qquad (6\text{-}1)$$

and the rate of decomposition of a fluid with a high boiler concentration (HB) is given by:

$$\text{Rate } (\text{hr}^{-1}) = K\ (100-HB)/75 \text{ (Ref. 6-10)} \qquad 6\text{-}2$$

where K and the rate are in hr^{-1} and T is in °K.

Information is also given for determining the distribution of decomposition products as high boilers, volatiles (generally C_6 and C_{12} hydrocarbons) and gases (generally H_2 and C_1 to C_5 hydrocarbons) (Ref. 6-10).

Specifications to reduce fouling from Therminol 66 (i.e., HB-40, OS-84) are reproduced from Reference 6-2:

A. Low concentration of particulate material at operating conditions.
B. Low chlorine content.
C. Exclusion of oxygen.
D. Maintenance of adequate water concentration (approximately 150 to 200 ppm).
E. Elimination of dissolved and particulate iron.

A distinction is made between coking and fouling. Coking is defined as a formation of massive carbonaceous deposits and is caused by decomposition of the fluid and precipitation of the high molecular weight insoluble decomposition products in regions of stagnant or low-flow velocity and high temperature. Fouling is the formation of thermally resistive films on heat-transfer surfaces and is dependent upon the nature and concentration of impurities in the coolant.

Two main types of fouling have been identified (Ref. 6-2 and 6-12 to 15) as (1) mass-transfer fouling, which involved inorganic deposits, and (2) particulate fouling, which involved carbonaceous deposits. Particulate fouling rates increase with a decrease in velocity while an increase in velocity reportedly increases the rate of mass transfer (or inorganic deposit) fouling (Ref. 6-2 and 6-14).

Inorganic deposits occurring in mass-transfer fouling are believed to be caused by the reaction of soluble impurities with iron from the piping system to produce an iron complex which decomposes at the heat-transfer surface to form Fe_3O_4, αFe or Fe_3C. The most important impurity in this process was chlorine (Ref. 6-2 and 6-12 to 15). A brief mechanism is cited for the iron-chlorine interaction (Ref. 6-12). The presence of small amounts of water will reduce fouling. Some theories on the role of water are reviewed by Smee et. al. (Ref. 6-12). It is surmised that water hydrolyzes a Lewis acid catalyst that can cause polymerization of benzene, biphenyl, and terphenyl.

Methods of fouling detection are discussed and reviewed (Ref. 6-2 and 6-12). Several methods are discussed such as pressure drop measurements, thermocouple measurements in a standard reactor fuel element with mounted thermocouples (Ref. 6-2 and 6-13), measurements of the particulate content, electrical conductivity measurements, and concentration of collodial species measurements (membrane stain test and tetrahydrofuran test), but the most successful technique for determining the fouling potential of an organic coolant was the small probe fouling test (SPFT) (Ref. 6-2, 6-7, and 6-12).

In the SPFT, a small flow of the heat-transfer fluid was passed over an electrically heated stainless steel probe for approximately 24-hr after which time the weight of film deposit per unit area was measured. A relationship was developed empirically between the deposited weight per unit area on the SPFT and the fouling potential of the fluid.

6.2.2 Thermal Stability/Compatibility Tests

Thermal stability and compatibility tests were conducted on three commercially available fluids which were candidates for meeting the requirements of the thermal storage system: Exxon Caloria HT43, Monsanto Therminol 55, and Monsanto Therminol 66. The objectives of the tests were: (1) to determine the ability of the heat-transfer fluids to function at 288° to 316°C (550° to 600°F) for extended periods of time, and (2) to assess the high-temperature, long-term compatibility of these heat-transfer fluids with rock and materials of construction (stainless steel, carbon steel) which will be in contact with fluid in thermal storage unit.

6.2.2.1 Test Equipment

The tests conducted on the heat-transfer fluids are designed to obtain data under conditions which simulate those that will be encountered in both the Pilot Plant and commercial plants.

The Pilot Plant TSS will be operated over long periods of time with the heat-transfer fluid cycling between 218° and 302°C (425° and 575°F). In the Pilot Plant and commercial plants, the pressure over the fluid in the thermal storage unit will be slightly over atmospheric, to prevent air leakage into the tank. Some volatile components resulting from thermal decomposition of the fluid will be removed from the ullage space through the UMU. The concentration of high boiling polymers will be regulated by a vacuum distillation process in the FMU.

Long-term thermal stability tests of the heat-transfer fluids were conducted at 288°, 302°, and 316°C (550°, 575°, and 600°F) in the presence and absence of rock, silica sand, stainless steel, and carbon steel. Two additional tests were conducted with Therminol 66 at 329° and 343°C (625° and 650°F) using electric heating mantles. All tests were carried out at atmospheric pressure under a nitrogen gas atmosphere.

The thermal stability tests were started using electric heating mantles at 288°, 302°, and 316°C (see Figure 6-2). The initial tests were interrupted after approximately 2,000 hr when three constant-temperature, molten-salt baths were set up to operate at these temperatures. The samples of Caloria HT43 at 288° and 302°C and a Therminol 66 sample at 316°C were transferred to the constant-temperature molten salt baths. A number of additional samples of fluids were prepared with and without solids. In all, 16 new fluid samples were prepared and put under test.

The rocks, metal samples, and fluid placed in each flask were individually weighed. Two kinds of rock were used: silica sand (in the range 16 mesh) and granite (from 1/2 to 3/4-in.). The rocks were rinsed in water, acetone, methanol, and distilled water. The 321 stainless steel samples are short pieces of 0.6-cm tubing that had been washed in an oxalic acid solution. Carbon steel samples consisted of turnings from a metal lathe that had been

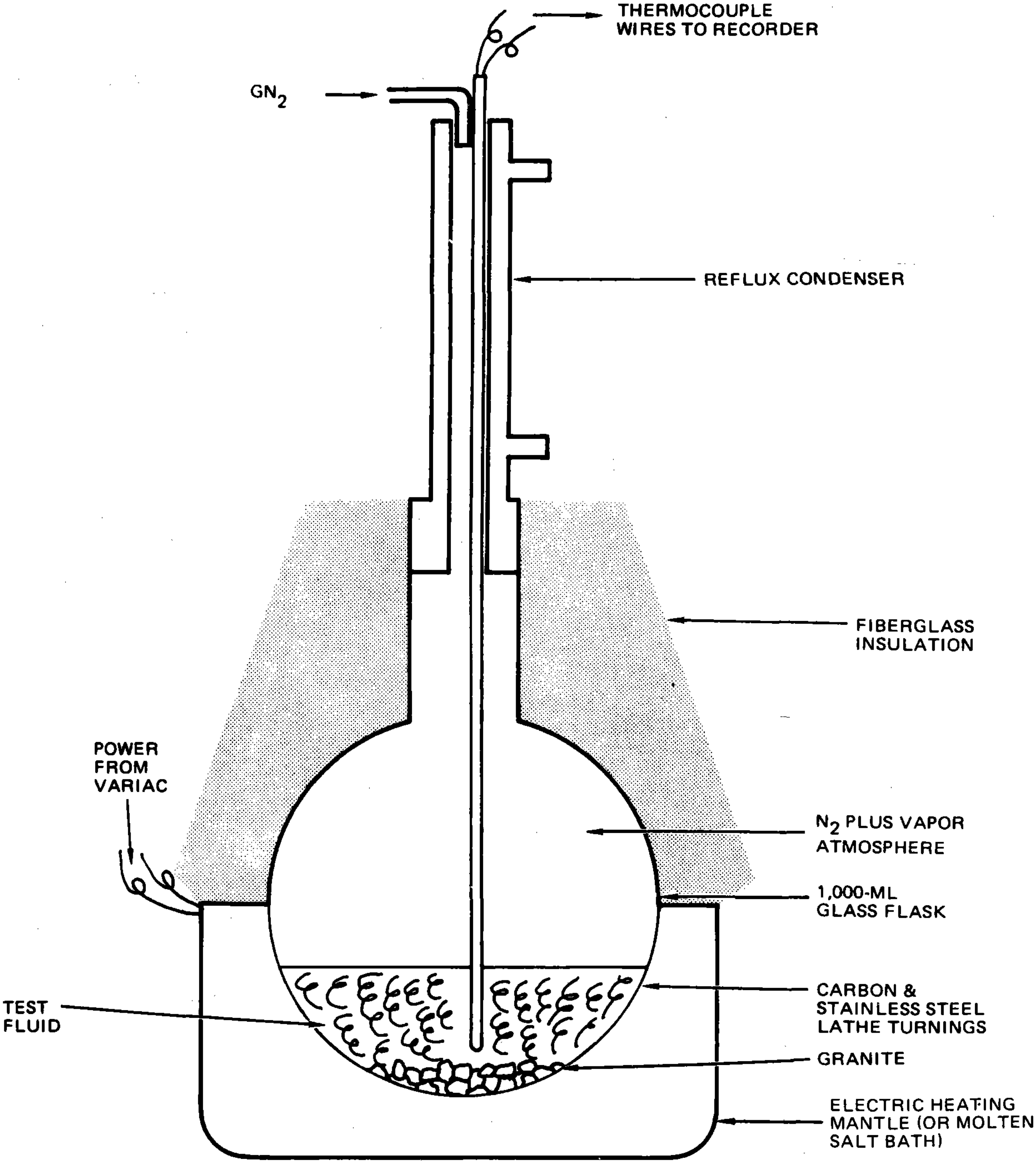


Figure 6-2. Fluid Thermal Stability and Compatibility Setup

degreased and rinsed in acetone, methanol, an oxalic acid solution, and distilled water. All of the materials were oven-dried before being added to the fluid.

All of the fluid thermal-stability tests were conducted in round-bottom flasks fitted with an air-cooled condenser. The top of the condenser is open to the atmosphere. A nitrogen atmosphere was maintained over the fluid and oxygen was prevented from entering the flask by a small nitrogen bleed at the top of the condenser. The temperature of the molten salt bath was controlled with Thermatrol regulators. The temperature of fluid samples in heating mantles was measured by Pyrex-enclosed thermocouples inserted down the condenser into the fluid; temperature was manually regulated by adjusting the power to the heating mantles.

At intervals of usually 1,000 hr, the flasks of fluid were removed from the constant temperature environment for testing. The flask and contents were weighed to determine the percent weight loss and the kinematic viscosity of the fluid was determined at 38°C with a Cannon-Ubbelohde viscosimeter. In addition, a fluid sample of approximately 10 ml was withdrawn at each 1,000-hr interval for further fluid testing. A small portion of these fluid samples was used by Sandia/Livermore for gel permeation chromatographic (GPC) and infrared (IR) spectroscopic analyses. The GPC tests are used to reveal the distribution of molecular size in the sample fluids. The distribution of molecular weights can be obtained after calibration of GPC columns by chromatographing pure components of known molecular weight and size. The IR data indicate the presence of functional groups, i.e., double or triple bonds, OH, -COOH, ring compounds, etc.

6.2.2.2 Results

Data on weight loss and viscosity change for the 19 tests are summarized in Table 6-3. The loss of fluid from each of the flasks over the heating time was determined from weighings before and after each heating internal. The percent weight loss was determined based on the original weight of fluid and corrected for the amount of sample material removed after each heating interval. Kinematic viscosity of the fluids was measured at 38°C with a Cannon-Ubbelohde viscosimeter to further assess the effect of heating on the fluid.

Table 6-3 (Page 1 of 3)

THERMAL STABILITY AND COMPATIBILITY TEST RESULTS

Fluid	Temperature °C (°F)	Rock and Metals	Time (hr)	Weight Loss (%)	Kinematic Viscosity Change (%)
Caloria HT43	316 (600)	Yes	230	1.4	--
			500	2.8	-27
			1,299	13.5	-34.3
			2,000	31.9	-35.3
	316 (600)	Yes	1,020	4.4	-35.7
			2,027	38.2	-12.0
	316 (600)	No	500	4.9	-34.4
			1,189	19.4	-36.0
			1,890	28.4	-42.2
	316 (600)	No	1,171	9.3	-33.0
			2,178	33.0	-22.5
	* 316 (600)	Yes	1,008	4.6	-34.6
	* 316 (600)	No	1,008	6.3	-29.4
	302 (575)	Yes	512	1.3	-6.4
			1,299	2.1	-13.8
			2,000	3.0	-17.1
			3,028	6.3	-24.1
			4,035	22.3	-11.3
			5,159	26.5	-20.0
			1,024	1.6	-16.2
			2,024	6.4	-23.1
			3,148	9.7	-29.5
			4,157	11.4	-47.2
	302 (575)	No	1,241	8.8	-8.2
			2,248	20.6	-9.2
	288 (550)	Yes	512	1.1	-5.8
			1,287	1.4	-6.1
			1,987	1.6	-7.1

* Tests Continuing

Table 6-3 (Page 2 of 3)

THERMAL STABILITY AND COMPATIBILITY TEST RESULTS

Fluid	Temperature °C (°F)	Rock and Metals	Time (hr)	Weight Loss (%)	Kinematic Viscosity Change (%)
Caloria HT43	288 (550)	Yes	2,971	11.8	14.7
			3,978	12.3	14.7
			5,102	12.3	11.0
			6,136	15.0	-17.9
	288 (550)	Yes	976	2.3	-4.8
			1,983	5.9	-1.8
			3,107	--	-4.6
			4,141	9.8	-27.9
	288 (550)	No	1,196	2.8	-3.5
			2,203	9.8	2.1
			3,327	--	0.3
			4,361	13.6	-24.0
Therminol 66	343 (650)	Yes	1,004	55.3	91.0
			1,984	66.1	--
	* 343 (650)	No	1,004	3.12	-13
			1,988	12.7	-17.1
			3,020	17.6	-23.2
			4,068	25.1	-20.6
			5,760	30.3	-26.1
			6,942	31.7	--
	* 329 (625)	Yes	1,013	3.2	-16.1
			2,061	5.2	-20.6
			3,753	7.3	-24.5
			4,935	9.0	--
	316 (600)	Yes	405	1.7	---
			842	2.8	-13.3
			1,303	3.9	-15.0
			2,010	12.1	-3.2
			3,017	24.7	31.3

* Tests Continuing

Table 6-3 (Page 3 of 3)

THERMAL STABILITY AND COMPATIBILITY TEST RESULTS

Fluid	Temperature °C (°F)	Rock and Metals	Time (hr)	Weight Loss (%)	Kinematic Viscosity Change (%)
Therminol 66	* 316 (600)	No	1,195	0.51	-0.6
			2,202	2.26	0
			3,210	2.85	-0.7
	* 302 (575)	Yes	1,024	6.74	20.8
			2,031	43.5	119
			3,155	45.5	120
			4,164	49.9	125
	* 302 (575)	No	1,266	4.2	0.23
			2,273	25.3	8.9
			3,397	25.4	7.8
			4,406	26.0	--
	288 (550)	Yes	956	3.1	-7.2
			1,963	18.8	47
			3,087	18.8	46
			4,121	19.8	58.8
Therminol 55	316 (600)	Yes	500	19.0	-54.7
			1,189	38.6	-52.2
			1,076	38.9	-56.8
			2,083	68.2	--
	316 (600)	No	1,100	13.4	-55.3
			2,107	52.8	-38.6
	302 (575)	Yes	1,146	17.1	-45
			2,153	68.1	--
	288 (550)	Yes	976	10.4	-56.0
			1,983	34.9	-24.2

* Tests Continuing

Caloria HT43

The curves representing the percent change in kinematic viscosity of Caloria HT43 as a function of time are plotted in Figure 6-3. While the data reveal some lack of reproducibility, certain trends are apparent. All of the Caloria HT43 tests show an initial decrease in kinematic viscosity. Higher fluid temperatures resulted in greater initial reductions in kinematic viscosity. After the initial decrease, the kinematic viscosity of at least one experiment at each of the three temperature levels, was observed to begin increasing with time. At 288°C (550°F) and 302°C (575°F) the increase in kinematic viscosity occurred over the same time interval as a somewhat abrupt weight loss of fluid was detected (Figures 6-4 and 6-5). At 316°C (600°F), those samples displaying an increase in kinematic viscosity (after approximately 1,000 hr) did incur a relatively large loss of fluid in the same time interval but the curves of percent weight loss with time (Figure 6-6) show no sudden changes or deviations. The data accumulated for Caloria HT43 at 316°C extend only to about 2,000 hr; more data are required at this temperature.

The initial decrease in kinematic viscosity of Caloria HT43 is probably caused by the accumulation of smaller molecules (less viscous components) as some of the original fluid molecules undergo thermal cracking. When the cracking products are boiled off from the fluid sample, as indicated by a large weight loss, the remaining fluid increases in kinematic viscosity. One could postulate a type of multistep process involving the thermal cracking of some of the original fluid species to form lower molecular weight compounds that accumulate and cause a decrease in the kinematic viscosity. With time, these initial cracking products undergo further reactions resulting in fragments capable of being devolatilized from the laboratory apparatus. The percent weight loss data at 288°C and 302°C given in Figures 6-4 and 6-5, display a low initial loss rate that increases with time then levels off again to some low weight-loss rate. The data obtained at 316°C (Figure 6-6), as stated before, cover only about 2,000 hr and only a sample without solids displays any tendency toward a lower rate of weight loss with time. It is believed that at longer times all of the samples will show a decrease in the weight loss rate.

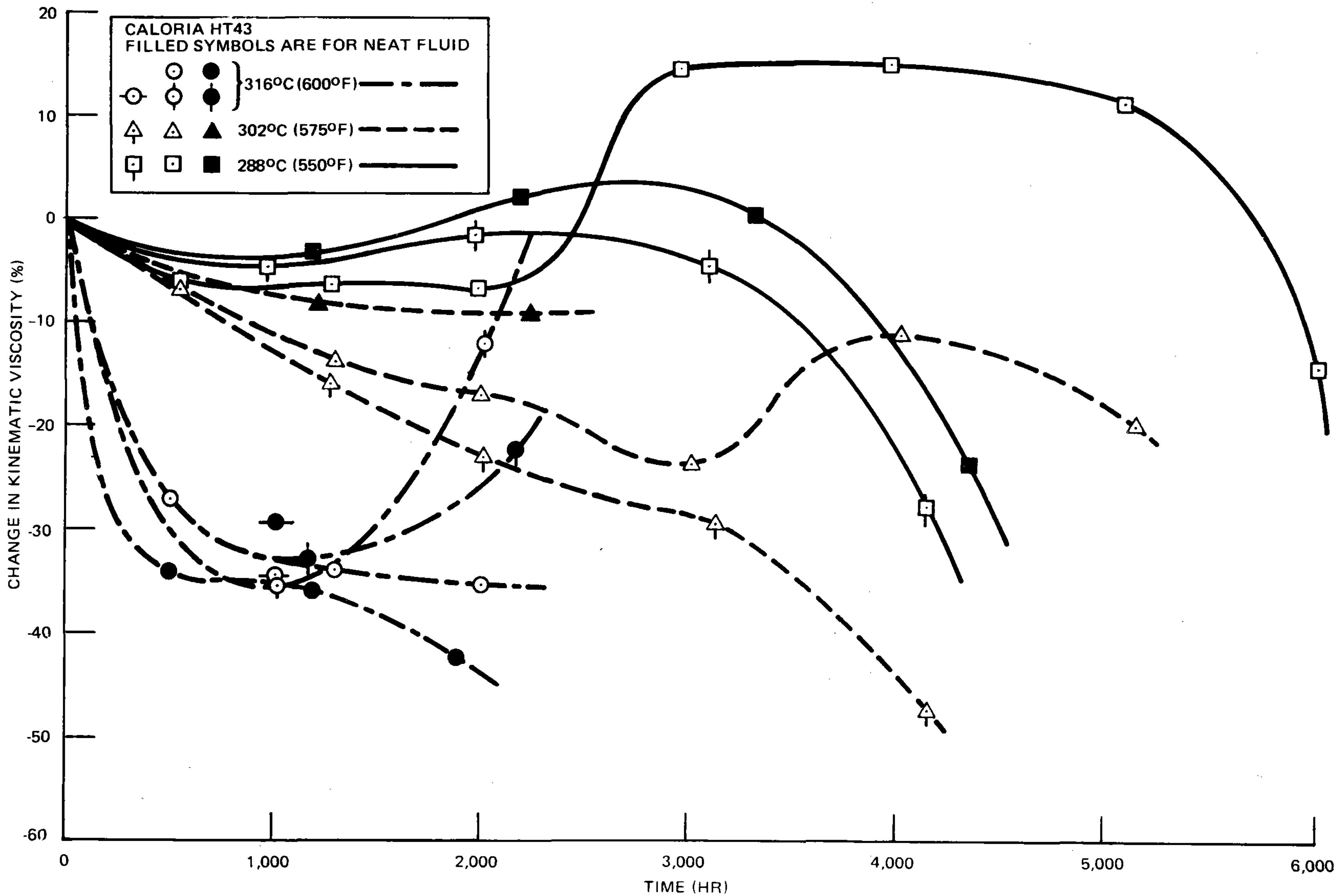


Figure 6-3. Caloria HT43, Percent Change in Kinematic Viscosity with Heating and Temperatures

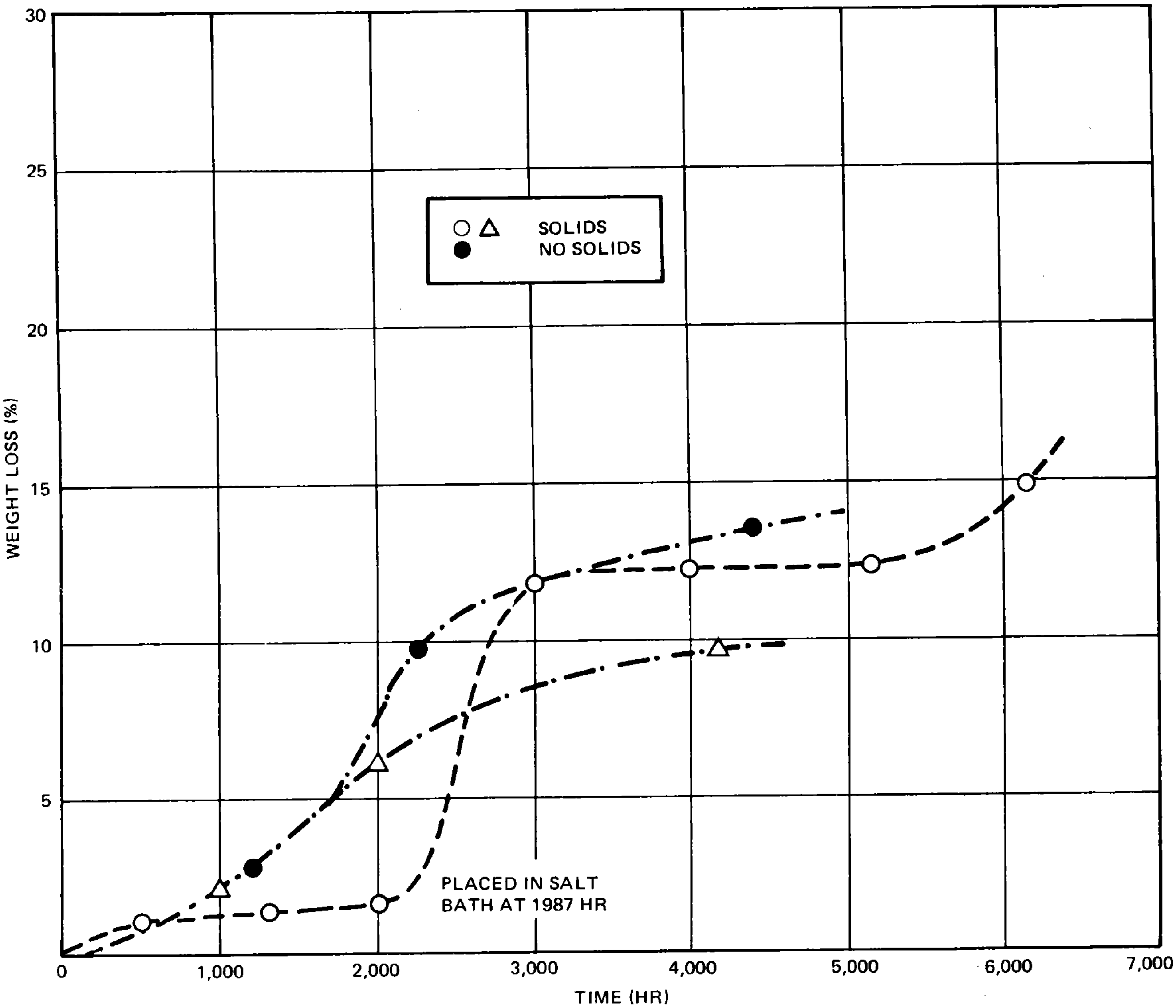


Figure 6-4. Caloria HT43, Percent Loss with Heating Time at 288°C (550°F)

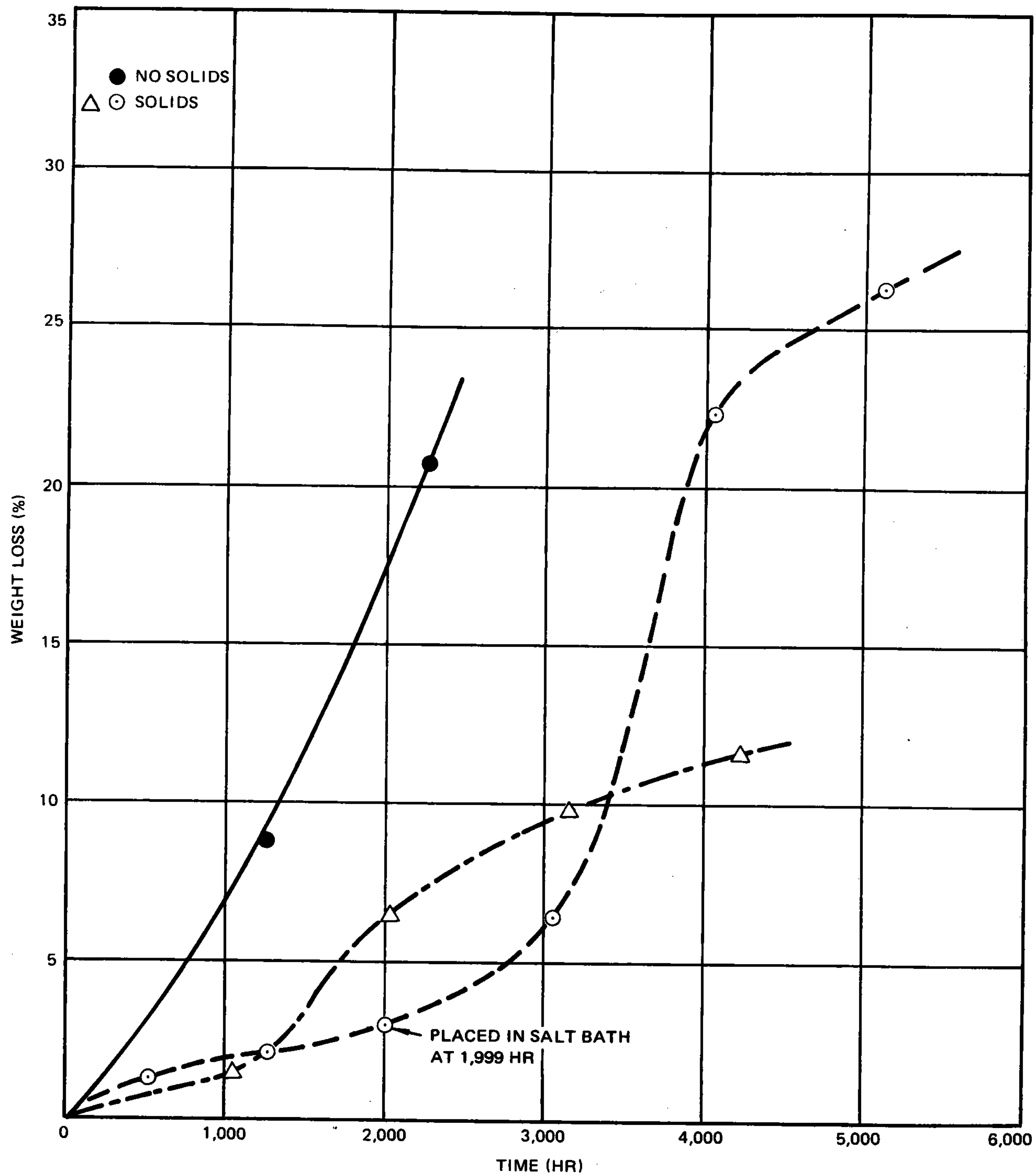


Figure 6-5. Caloria HT43, Percent Weight Loss with Heating Time at 302°C (575°F)

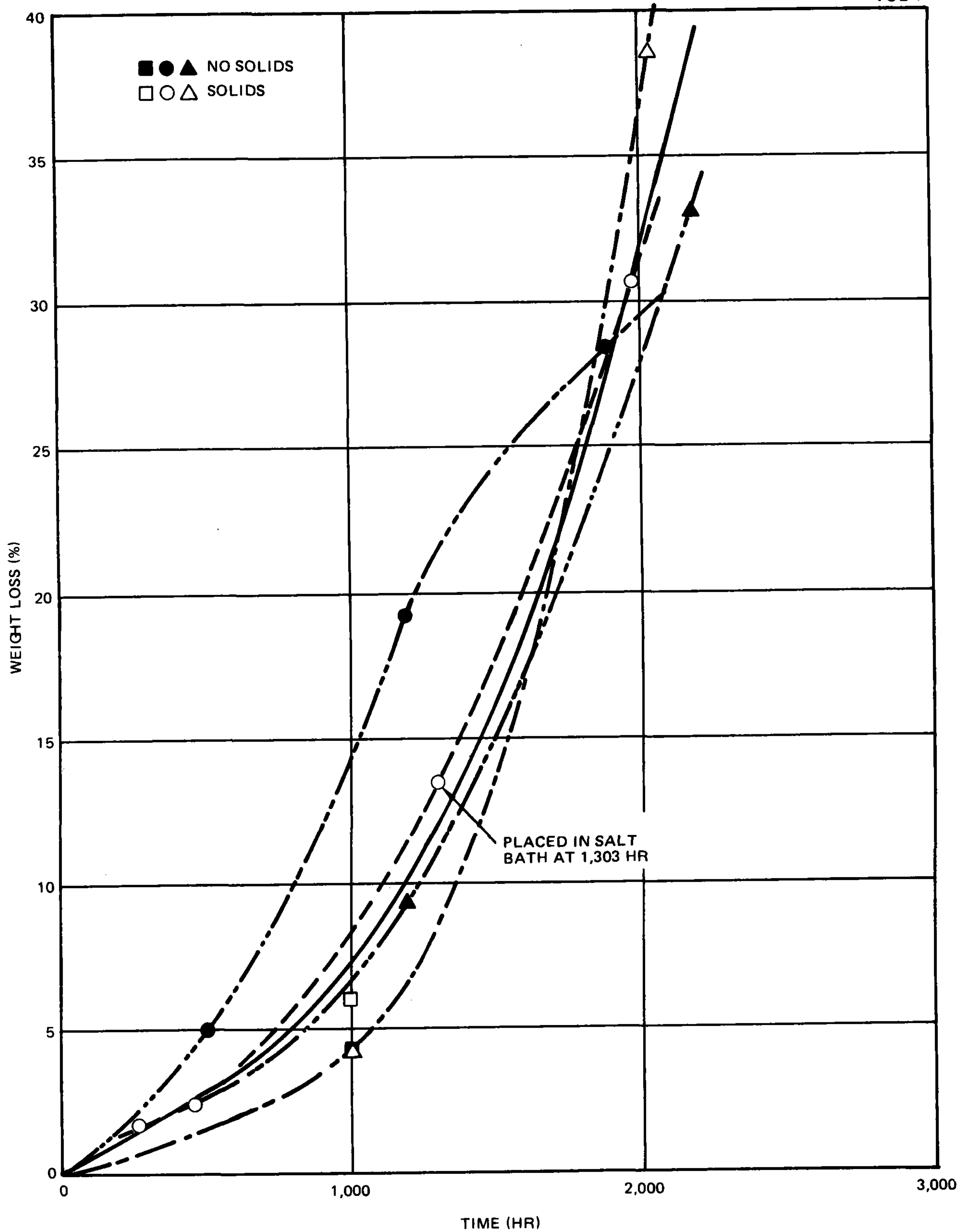


Figure 6-6. Caloria HT43, Percent Weight Loss with Heating Time, T = 316°C (600°F)

The weight loss and kinematic viscosity data, when viewed in terms of a multi-step process, would suggest that the second step leading to devolatilization occurs rapidly after the accumulation of a sufficient amount of the decomposition product from the first step.

The thermal decomposition of Caloria HT43 appears to be unaffected by the presence of the rock, silica sand, carbon steel, and stainless steel. If anything, the results of several experiments indicate that the solids may reduce the rate of fluid decomposition.

Gel permeation chromatography (GPC) and infra-red (IR) spectroscopic analyses were run on the samples of heat-transfer fluid taken at roughly 1,000-hr intervals. The GPC and IR tests were performed at Sandia/Livermore, and are still being interpreted (Reference 6-17). Tentative results from these tests indicate no evidence of any oxidation or of a significant amount of polymerization in the bulk oil. The results do not preclude reactions between low molecular weight species to form larger molecules that are still smaller than the average molecular size of the original fluid. The analyses are also interpreted as indicating that a significant amount of dehydrogenation (as opposed to unsaturation created by thermal cracking) occurred throughout the entire molecular size of the oil. The GPC columns have not yet been calibrated for the hydrocarbons likely to be found in Caloria HT43, which is necessary to obtain a molecular weight distribution.

Therminol 66

Data on thermal stability and compatibility for Therminol 66 are summarized in Figures 6-7 through 6-11. All of the Therminol 66 samples heated in the salt baths eventually showed an increase in kinematic viscosity above that of the fresh fluid (Figure 6-7). As with the Caloria HT43 data, rapid increases in kinematic viscosity accompany periods of high weight loss. Most of the data taken on Therminol 66 show that in the presence of the rock, silica sand, carbon steel, and stainless steel, the Therminol 66 undergoes greater weight losses and greater changes in kinematic viscosity than it does without solids, i.e., the added solids appear to speed up the thermal decomposition of Therminol 66.

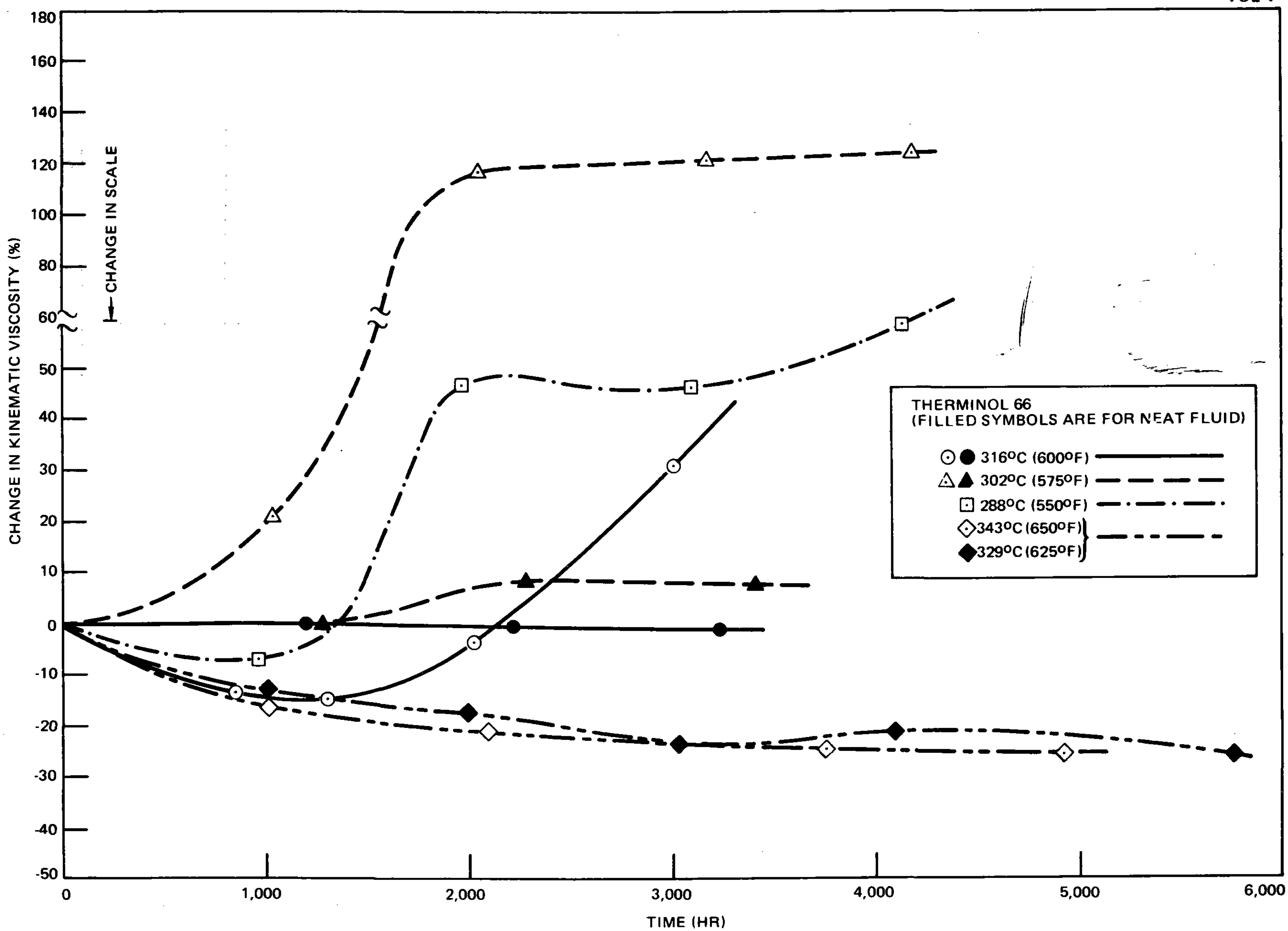


Figure 6-7. Therminol 66, Percent Change in Kinematic Viscosity with Heating Time and Temperature

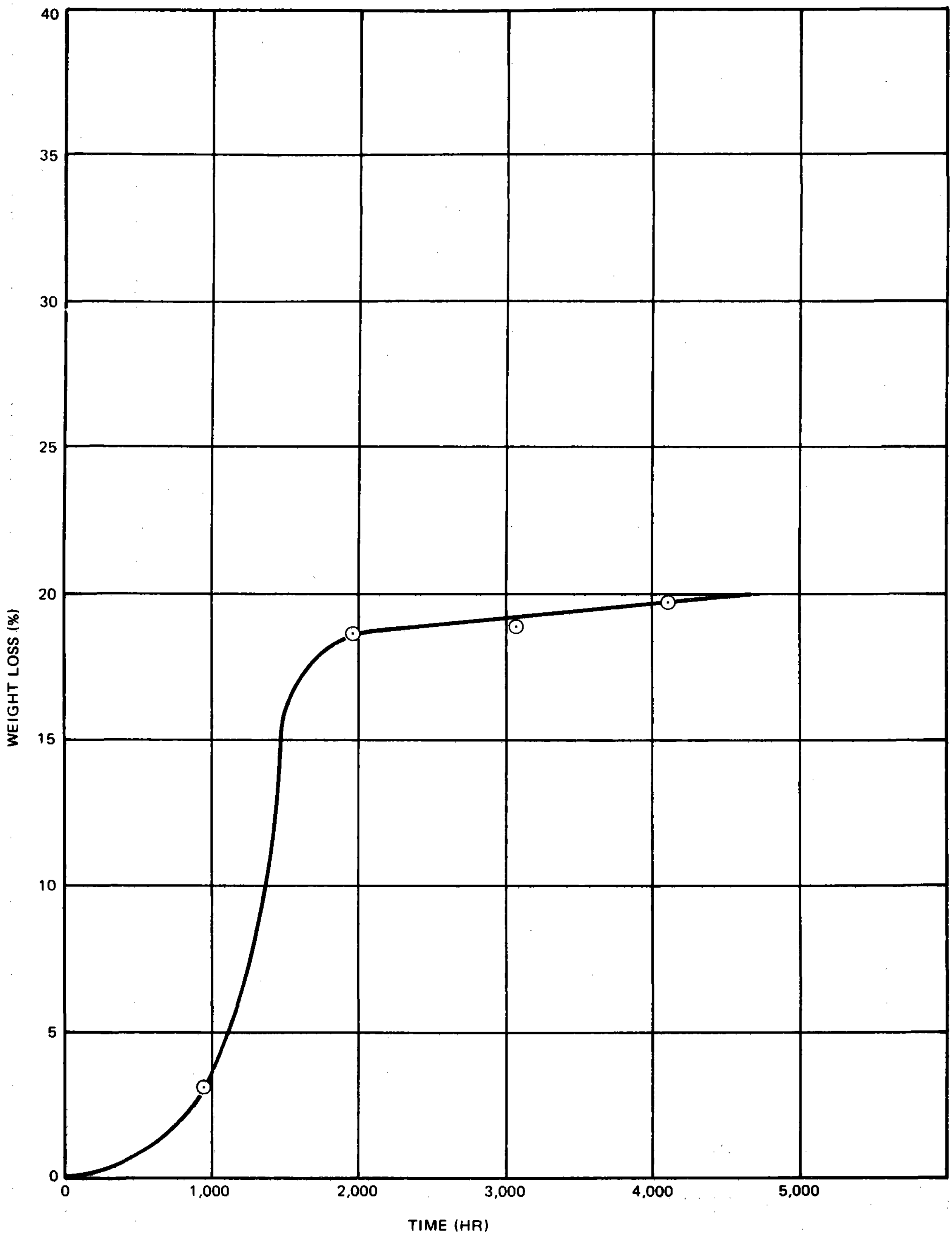


Figure 6-8. Therminol 66 with Solids, Percent Weight Loss with Heating Time at 288°C (550°F)

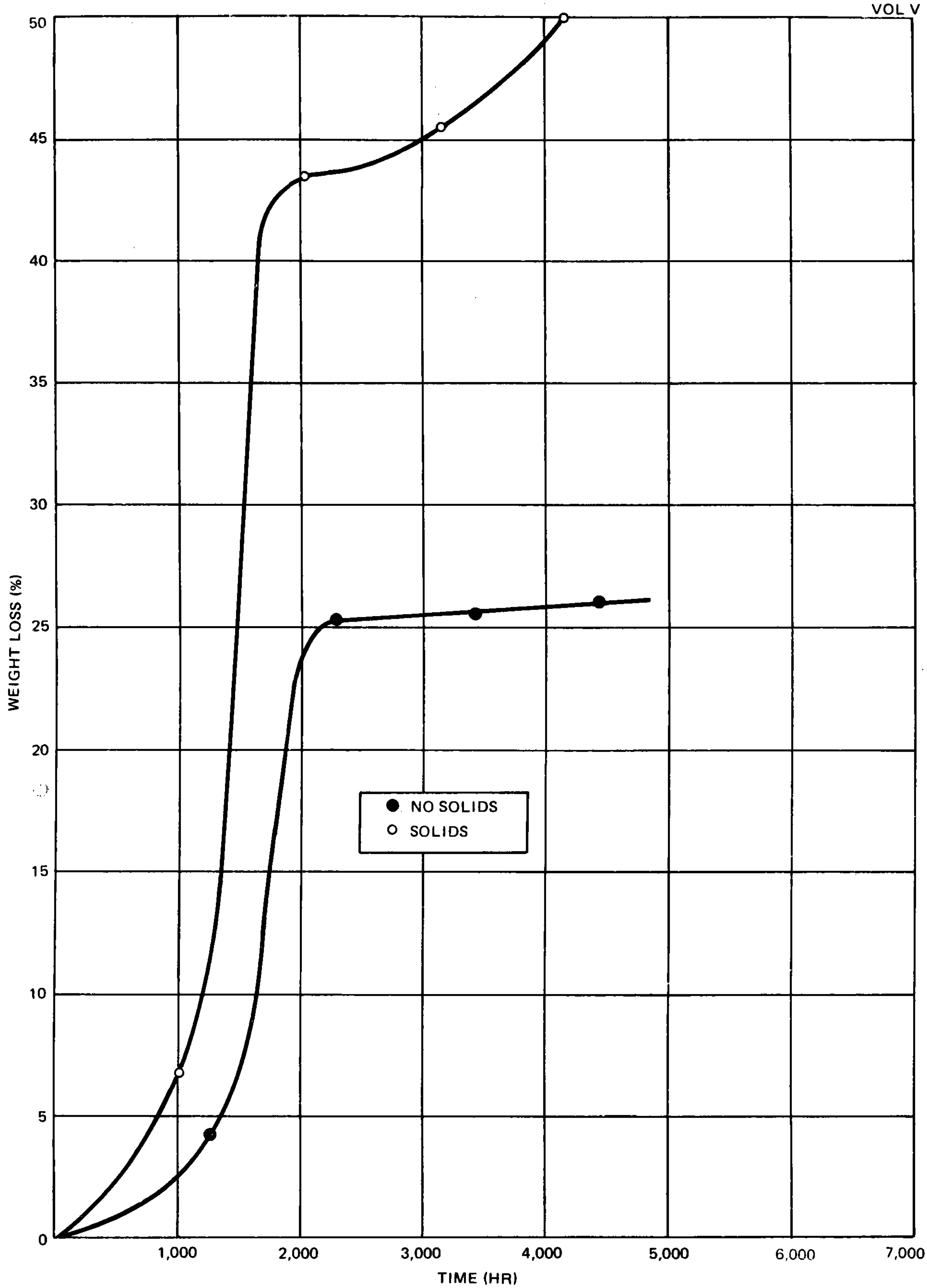


Figure 6-9. Therminol 66, Percent Weight Loss with Heating Time at 302°C (575°F)

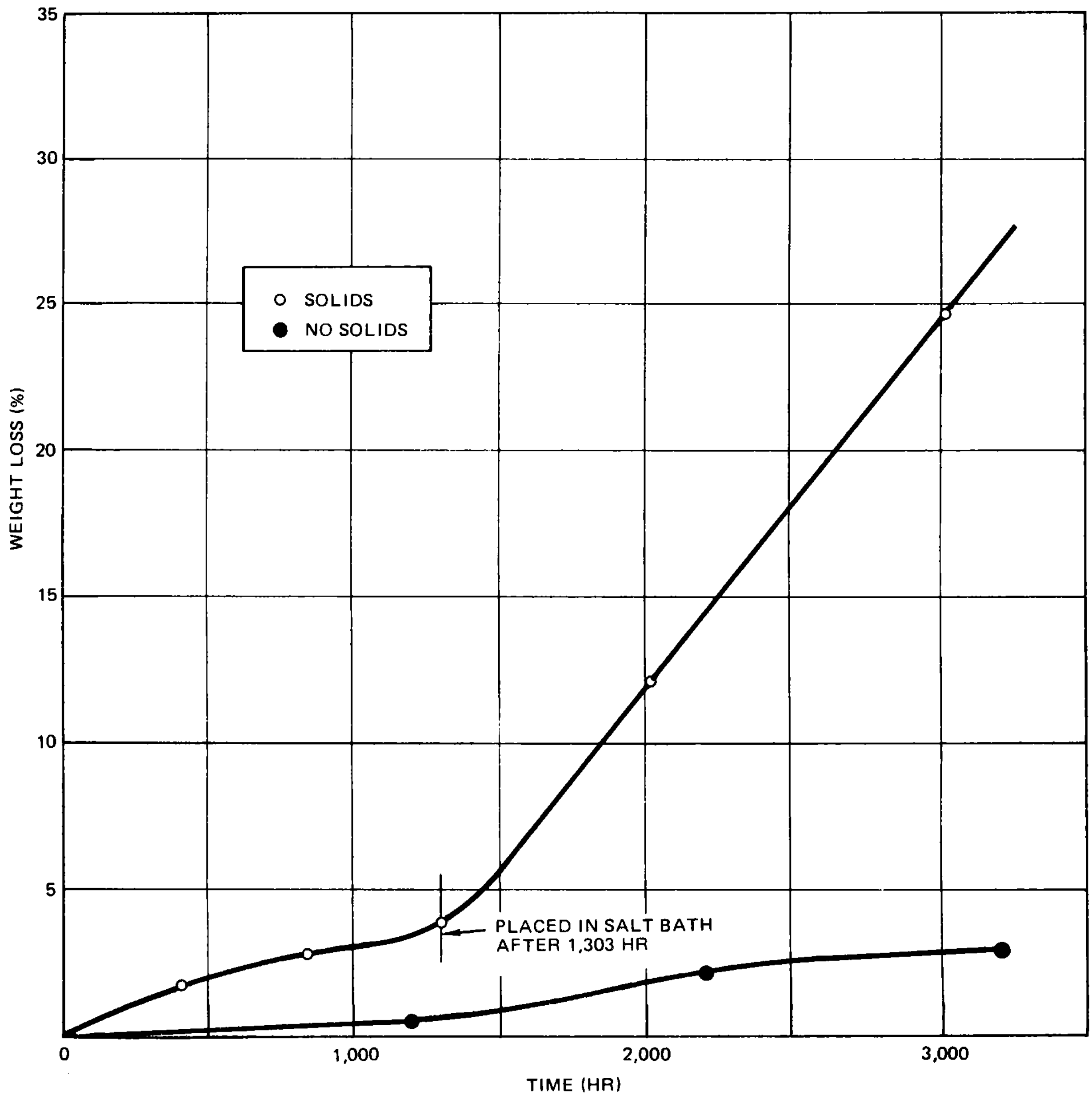


Figure 6-10. Therminol 66, Percent Weight Loss with Heating Time at 315°C (600°F)

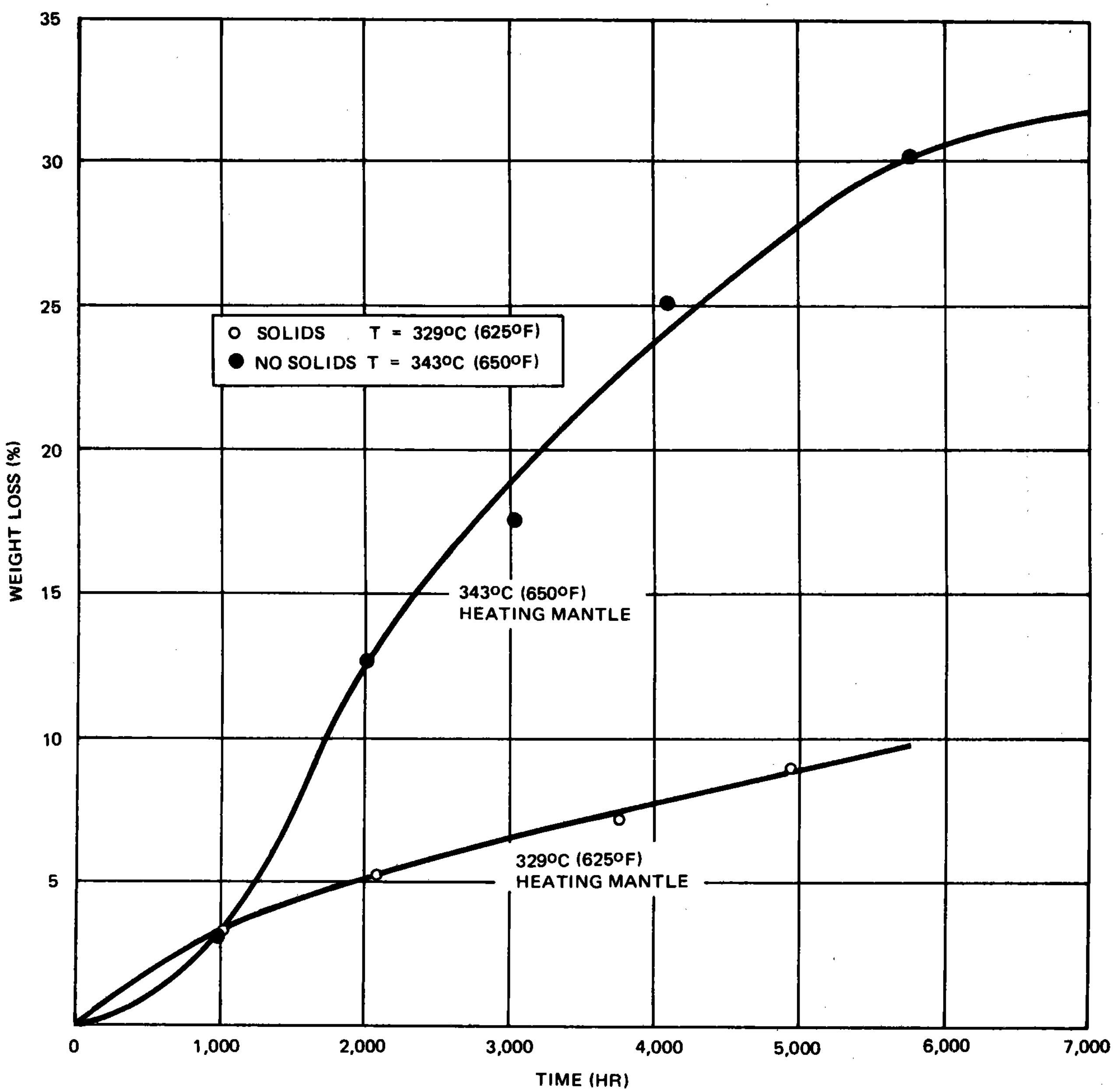


Figure 6-11. Therminol 66, Percent Weight Loss with Heating Time at 343°C (650°F) and 329°C (625°F)

Data obtained from samples of Therminol 66 heated to 329°C (625°F) and 343° C (650°F) with electric heating mantles, are not in general agreement with the rest of the data taken from fluid samples heated to 288°C (550°F), 302°C (575°F), and 316°C (600°F) in constant temperature molten salt baths. The percent-weight losses from the fluid at 329°C and 343°C (Figure 6-11) are subsequently lower than the losses measured at 316°C and 302°C (Figures 6-10 and 6-9). In addition, the kinematic viscosity curves given in Figure 6-7 for 329°C and 343°C show a relatively steady value. In an experiment conducted with Therminol 66 and solids at 316°C, after the first 1,303 hr the heat source was changed from an electric heating mantle to a molten salt bath. Both weight loss and kinematic viscosity curves given in Figures 6-7 and 6-10 show evidence that more has occurred than just a simple change in the heat source. Fluid samples heated in salt baths show greater weight losses and changes in kinematic viscosity than those heated with electric heating mantles.

Therminol 55

Samples of Therminol 55 were subjected to temperatures of 288°, 302°, and 316°C (550°, 575°, and 600°F) (Table 6-3). The earliest test, conducted using a heating mantle, resulted in an excessive weight loss (38.6 weight-percent after 1,189 hr). The test was discontinued. Duplicate tests were performed using Therminol 55 from another batch of the fluid (to eliminate the possibility that the original Therminol 55 sample had some unusual defect). As shown in Table 6-3, tests conducted with the second batch of Therminol 55 corroborate the earlier results. All testing of Therminol 55 was discontinued after about 2,100 hr, when large weight losses and changes in viscosity indicated this fluid could not be used economically.

Heat Source Effect

Thermal stability experiments indicate that the type of heat source used can somehow influence the rate of fluid weight loss. Data plotted in Figures 6-4, 6-5, and 6-10 show the change in weight loss rate accompanying the transfer of a flask from a heating mantle to a salt bath. Flasks of fluid, with or without solids, appear to lose mass more rapidly when heated with a molten salt bath than when heated with an electric mantle. The fluid temperature of flasks placed in a heating mantle, were monitored by a thermocouple probe inserted in the fluid, or if solids were present, in the granular solids layer. The fluid temperature was controlled by manually regulating the power to the heating mantle with a Variac. In the molten salt bath, on the other hand, the salt temperature was measured and controlled by thermoregulators.

The apparent effect of heat source on the fluid weight loss can be ascribed to differences in the surface area of the flask that is heated. Flasks placed in the salt bath were immersed to a depth sufficient to cover the entire round portion of the flask. In the heating mantle, however, only the bottom half of the flask is heated. Since the flasks are usually less than half-full of fluid, the vapor contents of a flask immersed in a salt bath are subjected to a heated surface that may enhance the decomposition rate. Then too, the unheated top of a flask placed in a heating mantle serves as a condensing surface. This added condensing surface could result in a lower loss of fluid weight for samples heated by an electric heating mantle.

Weight Loss Rate

The determination of the fluid replacement rate of Caloria HT43 and Therminol 66 from the long-term thermal stability tests, is complicated by the fact that the data on the percent weight loss vs time (Figures 6-4 to 6-6 and 6-8 to 6-10) cannot be correlated by straight lines. For the first 1,000 or 2,000 hr, the rate of fluid loss is seen to accelerate. After this acceleration period, the fluid weight-loss curves appear to level off and approach a constant slope. Experiments will be extended to longer times to ascertain whether the rate of fluid loss will remain constant after the initial acceleration. For Caloria HT43 at 316°C, where the data extend only to about 2,000 hours, only one experiment appears to be leveling off (this curve was used to determine the fluid weight loss rate at 316°C). The slopes of the curves (weight loss rates), measured after the acceleration period for Caloria HT43 and Therminol 66, are given in Table 6-4.

Table 6-4

WEIGHT LOSS RATE MEASUREMENTS

T °C (°F)	Caloria HT43 Rate (%/hr)	Therminol 66 Rate (%/hr)
288 (550)	1.06×10^{-3}	4.85×10^{-4}
302 (575)	2.81×10^{-3}	2.74×10^{-3}
316 (600)	4.3×10^{-3}	1.26×10^{-2}

These rates are plotted vs $1/T$ on semilog coordinates in Figure 6-12. The reaction rate equations (with R in wt %/hr and T in °K) obtained from Figure 6-12 are:

$$R = 5.38 \times 10^{10} \exp[-17650/T] \text{ (for Caloria HT43)} \qquad 6\text{-}3$$

$$R = 1.93 \times 10^{27} \exp[-39580/T] \text{ (for Therminol 66)} \qquad 6\text{-}4$$

The rate data indicate that above 302°C (575°F), Therminol 66 with solids will degrade faster than Caloria HT43. The more extensive data obtained by AECL on the decomposition of Therminol 66 as a function of temperature and percent high boiler (Ref. 6-9) indicate a much lower rate. At 316°C (600°F) the weight loss rate obtained from this study, Eq. 6-4 is about 500 times the rate calculated using the AECL rate expression, Eq. 6-1 and 6-2. Catalysis by the added solids is believed to be responsible for the higher weight loss rates of Therminol 66 measured in this study.

Fluid Replacement Rate

We have obtained data on the fluid loss vs time at 288°, 302°, and 316°C (550°, 575°, and 600°F) in batch systems. In the TSU, the makeup fluid required to replace losses due to thermal decomposition and subsequent devolatilization or perhaps polymerization and filtration, will be supplied continuously to the fluid inventory. With the passage of time, the fluid inventory attains a constant average age and a constant fluid makeup rate. If it is assumed that the decomposition rate (or rate of fluid loss by devolatilization) of fluid subjected to high temperatures for a certain time (and hence having a certain age), is unaffected by the age of fluid it is mixed with, i. e., the fluid decomposition rate is a function only of its age and is independent of the ages of other fluid it may be mixed with, the problem can be formulated as follows. If $W(y)$ is the amount of fluid added to the TSU at time y to $y + dy$, $W(t, y)$ represents the amount of fluid added at y to $y + dy$ remaining at time t, $L(t-y)$ is the equation relating the fractional weight loss of the fluid to residence time $(t-y)$ at temperature T, and $dL(t-y)/dt$ is the weight loss rate of the fluid after a residence time $t-y$, then the rate of addition of makeup fluid caused by losses from $W(y)$ at time t is

$$\frac{dW(t, y)}{dt} = W(y)\,[1 - L(t-y)]\,\frac{dL\,(t-y)}{dt} \qquad 6\text{-}5$$

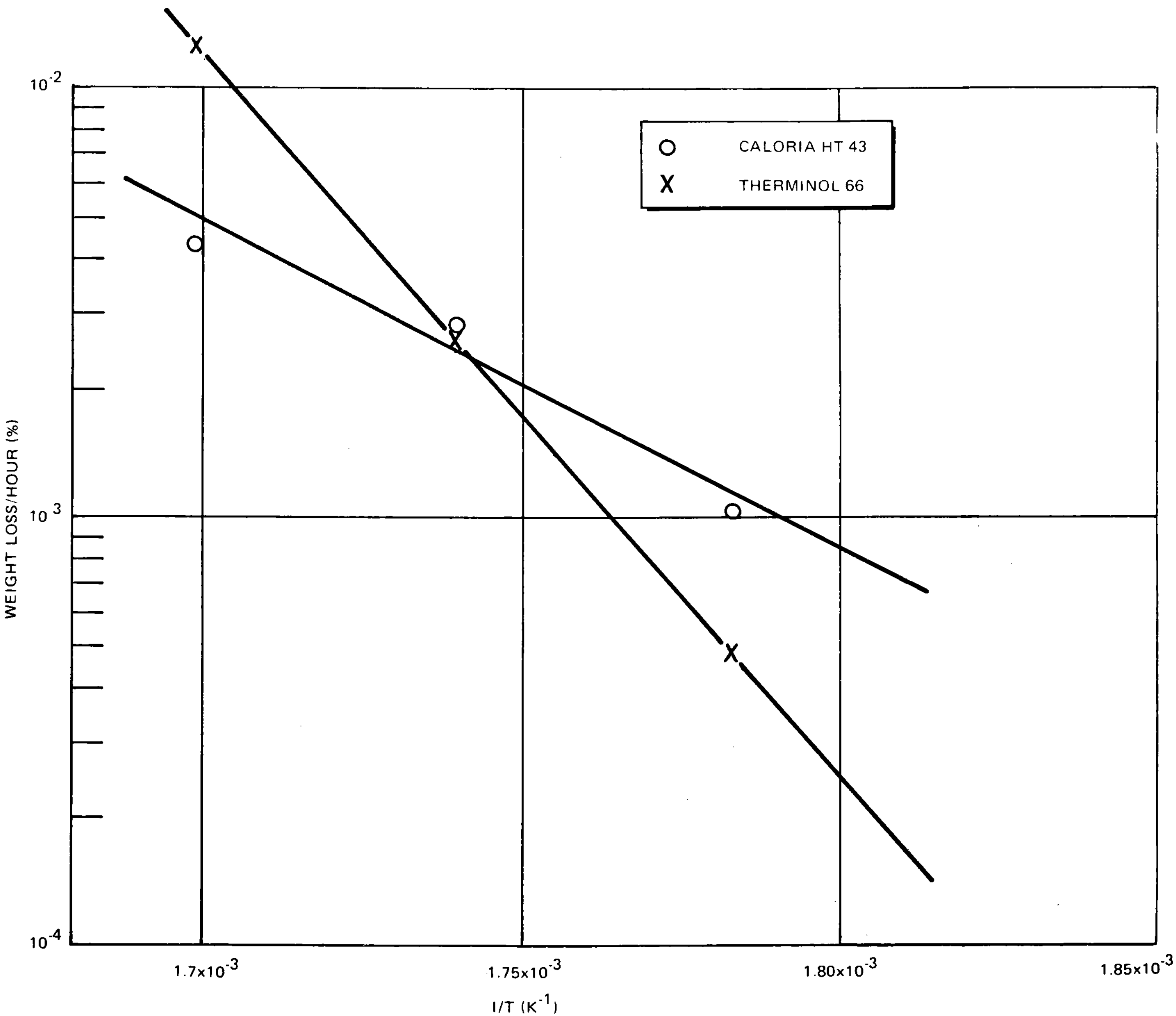


Figure 6-12. Weight Loss Rate of Caloria HT43 and Therminol 66

and the total fluid makeup required at time t, W(t), is

$$W(t) = \int_{y=0}^{y=t} \frac{dW(t,y)}{dt}\, dy = \int_{y=0}^{y=t} W(y)\,[1 - L(t-y)]\, \frac{dL(t-y)}{dt}\, dy \qquad 6\text{-}6$$

The function L(t-y) representing fluid loss rate would be obtained from a curve fit of weight loss vs time data such as Figures 6-4 to 6-6, as appropriate.

If there is an interaction between fluid of various ages due, for example, to some intermediate present in older fluid that may catalyze or stabilize the decomposition of fresh fluid, then Equation 6-6 is invalid. The fluid replacement rate would then have to be determined from experiments in which fresh makeup fluid was continually added to the system to replace the volatilized fluid.

Equation 6-6 has not been solved for W(t) in closed form. The solution, however, would indicate that with the passage of time the fluid would attain a steady-state composition and W(t) would therefore approach a constant value. It is believed that this steady-state value can be closely approximated by using the fluid weight loss rate Eq. 6-3 for Caloria HT43 and Eq. 6-4 for Therminol 66, determined for the constant weight-loss regime of fluid batches.

Application of the fluid weight-loss equations to the TSU requires knowledge of the TSU cycling time and the percent of the time the fluid spends at each temperature. Experiments with the TSU subsystem have shown that the thermoclines in the thermal storage tank are steep. Thus, the percentage of fluid present at temperatures between the top and bottom temperature of the TSU is rather small and can be neglected.

A typical 24-hr operating cycle of the TSU will consist of 10 hr with the entire tank at the bottom temperature, 1 hr with the entire thermal storage tank at the top temperature, and the remaining time being divided between heating up and cooling down of the storage tank (with the thermocline moving through the fluid-rock bed). It is assumed that the charging and extraction of energy to and from the tank will be done with the thermocline moving at a constant rate through the tank. A plot of the percent of total fluid at the top and bottom temperature versus time (Figure 6-13) reveals that for one 24-hr cycle the entire fluid inventory can be considered to be at the highest temperature for 7.5 hr and at the bottom temperature for 16.5 hr.

For the Pilot Plant cycle where the top temperature is 302°C (575°F) and the bottom temperature is 218°C (425°F), the loss rate per day can be computed for Caloria HT43 via Equation 6-3 and for Therminol 66 using Equation 6-4. Similar computations can be made for the Commercial Plant where the top and bottom temperatures are 316°C (600°F) and 232.2°C (450°F). The results are presented in Table 6-5 as %/day and %/yr calculated using the temperature-time cycle shown in Figure 6-13, with 1 yr being defined as 330 cycles.

Figure 6-14 shows the dependence of the calculated fluid replenishment rate upon the length of the TSU charging and energy-extraction portion of the 24-hr cycle.

6.2.3 Heated Surface Fouling Tests

Surface fouling from degradation of the heat-transfer fluid is most likely to occur at the points of highest surface temperatures and lowest velocity.

Examination of the fluid flow loops shows that the highest temperatures occur in the thermal storage heater. The potential impact of fouling, if it should occur, would be greater for the heater than for many other portions of the system; e.g., fouling of the piping or heat storage media (rocks) would have little impact. Continuing attention will be given to evaluate the possibility of fouling in other portions of the fluid-circulation loops and, if such fouling seems possible, to evaluate whether it would have significant impact on the subsystem operation.

Tests have been performed to determine the extent and rate of fouling of electric heaters immersed in the heat-transfer fluids as a function of the surface temperature of the heater, for two fluids, Caloria HT43 and Therminol 66.

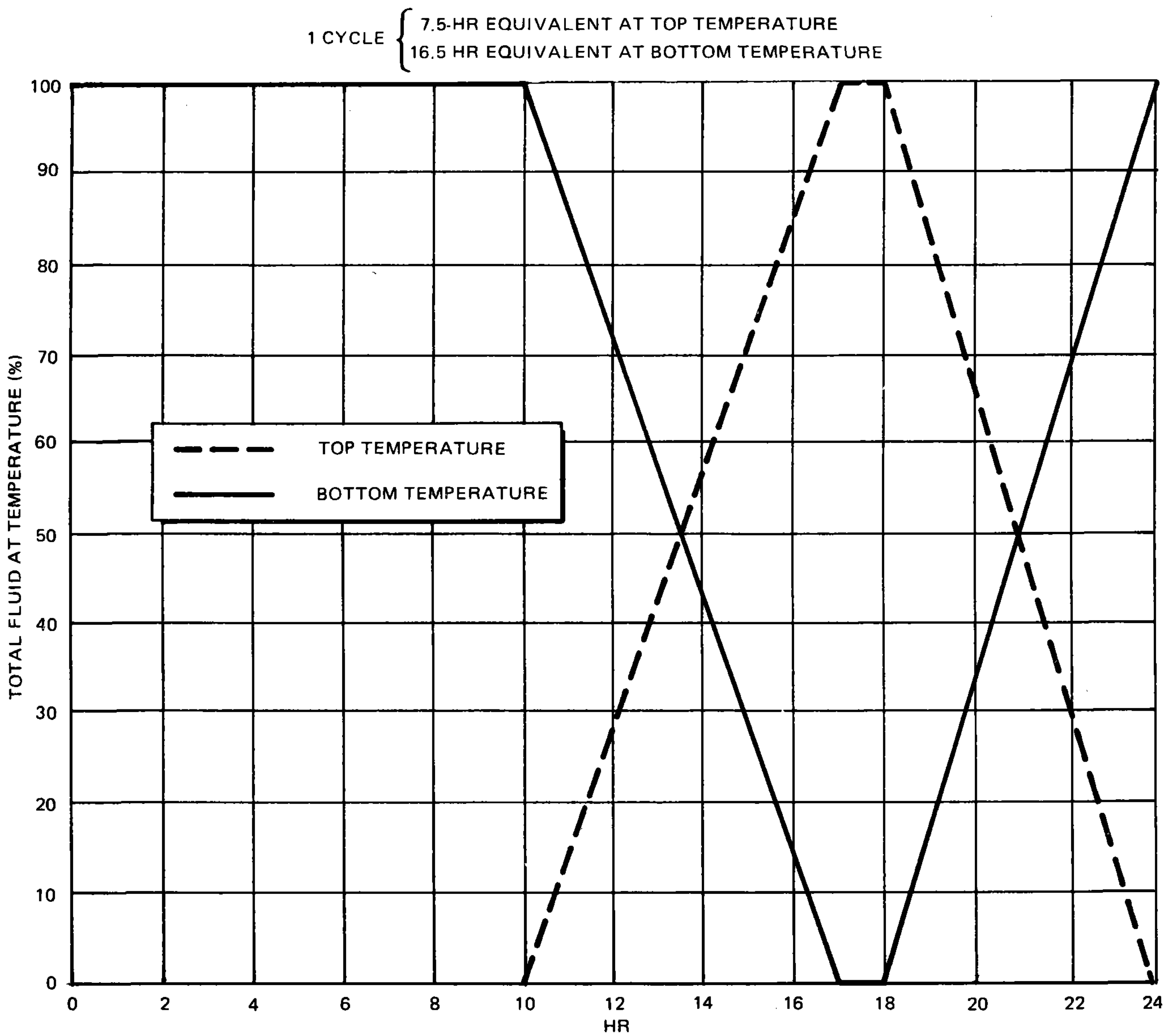


Figure 6-13. 24-Hour Operating Cycle for Fluid Loss Calculations (Steep Thermocline Is Assumed; Fluid Is Either at Top or Bottom Temperature)

Table 6-5

CALCULATED FLUID REPLENISHMENT RATE

Plant	Temperature °C (°F)	Caloria HT43 %/day	Caloria HT43 %/yr	Therminol 66 %/day	Therminol 66 %/yr
Pilot	302(575) 218(425)	0.0212	7.00	0.0184	6.07
Commercial	316(600) 232(450)	0.0391	12.9	0.0936	30.9

6.2.3.1 Test Equipment

Each apparatus for the surface fouling tests (see Figure 6-15) consists of a 180W electric heating element, sheathed with 304 stainless steel, immersed in a pool of the heat-transfer fluid contained in a 10-cm (4-in.) diameter Pyrex glass pipe cap bolted to a stainless steel plate. Heat transfer from the electric heater to the fluid occurs by natural convection which should, because of the low fluid velocity over the heater, represent the worst possible situation. Thermocouples spot-welded to the heater surface are used to monitor the surface temperature, which is maintained constant by manual adjustment of the heater voltage. The surface temperatures were continuously monitored by a multipoint recorder. The ullage space at top of the Pyrex cap contains nitrogen. A 1/4-in. SS tube extends into the ullage space to prevent a pressure buildup when the apparatus is initially brought up to temperature and to vent any gaseous decomposition products produced during the course of the experiment. The Pyrex pipe cap was wrapped with fiberglass insulation.

Six fouling tests were conducted simultaneously. At the start of the fouling tests, three of the test setups were filled with Caloria HT43 and three with Therminol 66. The surface temperatures of the electrical heaters were controlled at 316°, 329°, and 343°C (600°, 625°, and 650°F). The manufacturer's recommended maximum film temperatures are 360°C (680°F) for Caloria HT43 and 374°C (705°F) for Therminol 66.

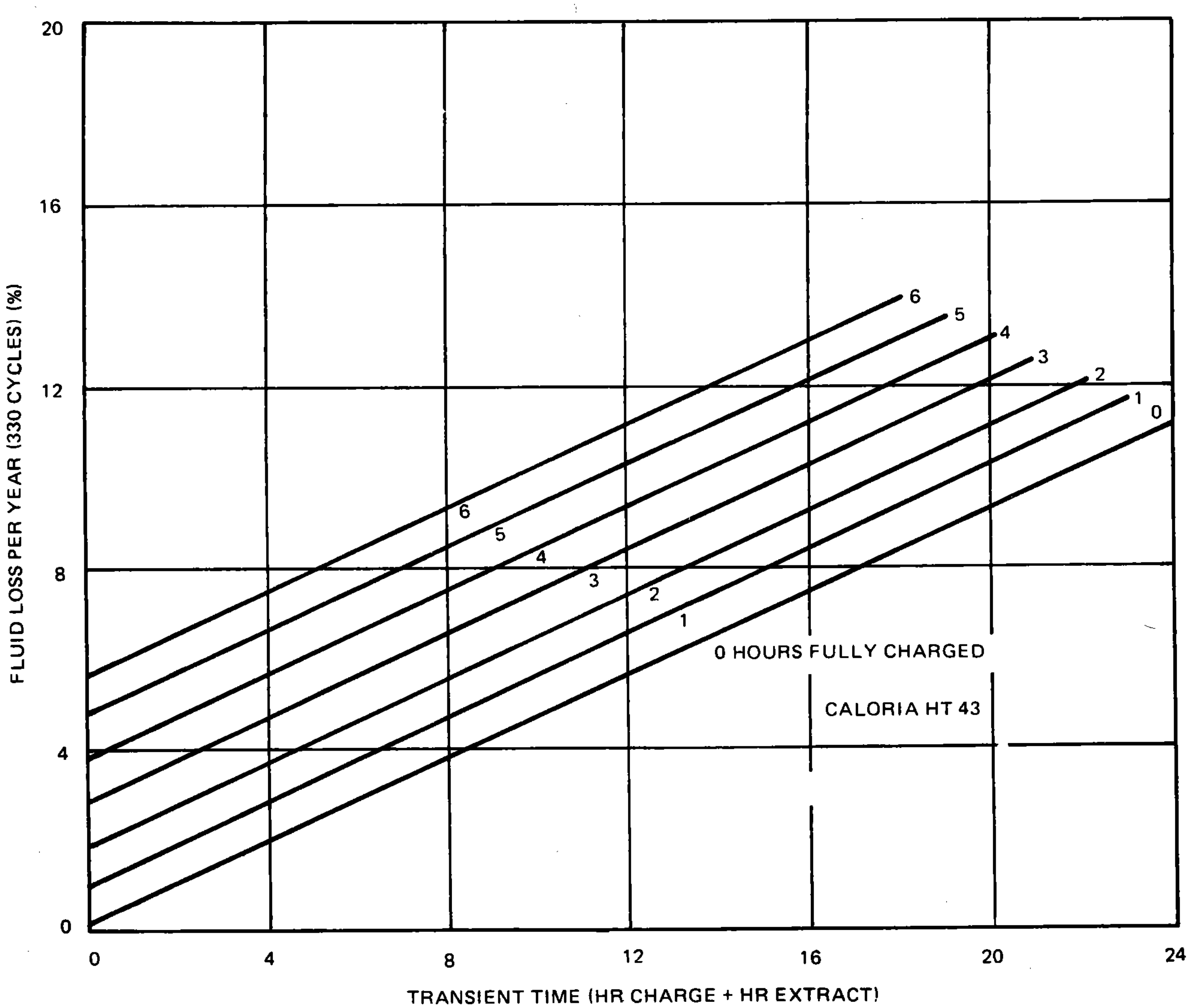


Figure 6-14. Fluid Loss (Percent Per Year) from the Pilot Plant Operating with Caloria HT43

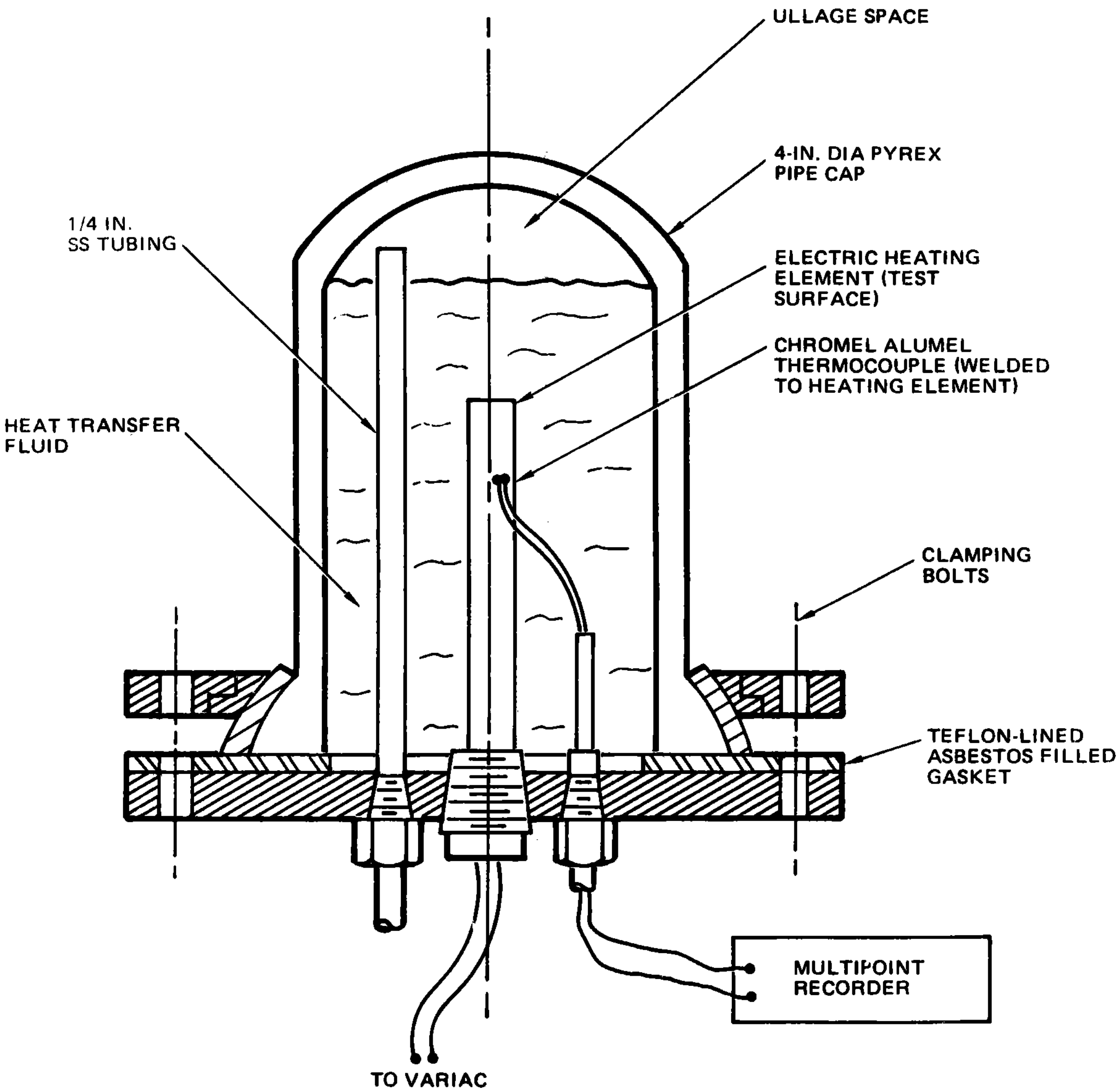


Figure 6-15. Surface Fouling Test Setup

Visual examination, still and motion photography, and measurement of the change -- with time in power supplied to the heater to maintain the surface temperature at a constant value -- were used to detect the presence of a surface film. The weight per unit area of deposits was determined in one case by removing the deposit from a known area and weighing it.

6.2.3.2 Results

The fouling experiments were run continuously. Several times a day the heater surface temperature was checked and the heater voltage adjusted accordingly. From the outset, bubbles appeared on the heater surface of all three Caloria HT43 tests. Some slow bubbling was also observed at the surface of the heaters in the Therminol 66 tests. Convective flow patterns near the heater surface were visible because of the differences in the refractive index of the fluid caused by the temperature gradients in the fluid. At the lowest heater wall temperature, 316°C (600°F), the flow near the heater wall was laminar as shown in Figure 6-16, while at 329°C (625°F) a transition from laminar to turbulent could be seen near the top of the heater (Figure 6-17). At 343°C (650°F) the transition zone had moved some distance down to the base of the heater.

The results from all fouling tests on the two fluids indicate no problems in practice due to fouling of heated surfaces. There was, however, a temperature abnormality in the lowest temperature test with Caloria HT43. After about 200 hr of testing, the Caloria HT43 heater at 316°C (600°F) was discovered to have extensive patches of gummy material that had a bubbly or blistered appearance. However, there was no significant change in thermal resistance due to the deposits. The deposits are believed to have formed around vapor bubbles that nucleated on the heater surface and were not rapidly swept away by the natural convective flow. A larger resistance to heat transfer in the vicinity of the vapor bubble, and the slower moving fluid, may have resulted in greater local heating and hence in the formation of surface deposits.

Although more bubbling should occur at the higher wall temperatures of 329°C (625°F) and 343°C (650°F), the faster moving natural convective flow near the heater wall (especially where the flow near the heater wall appeared

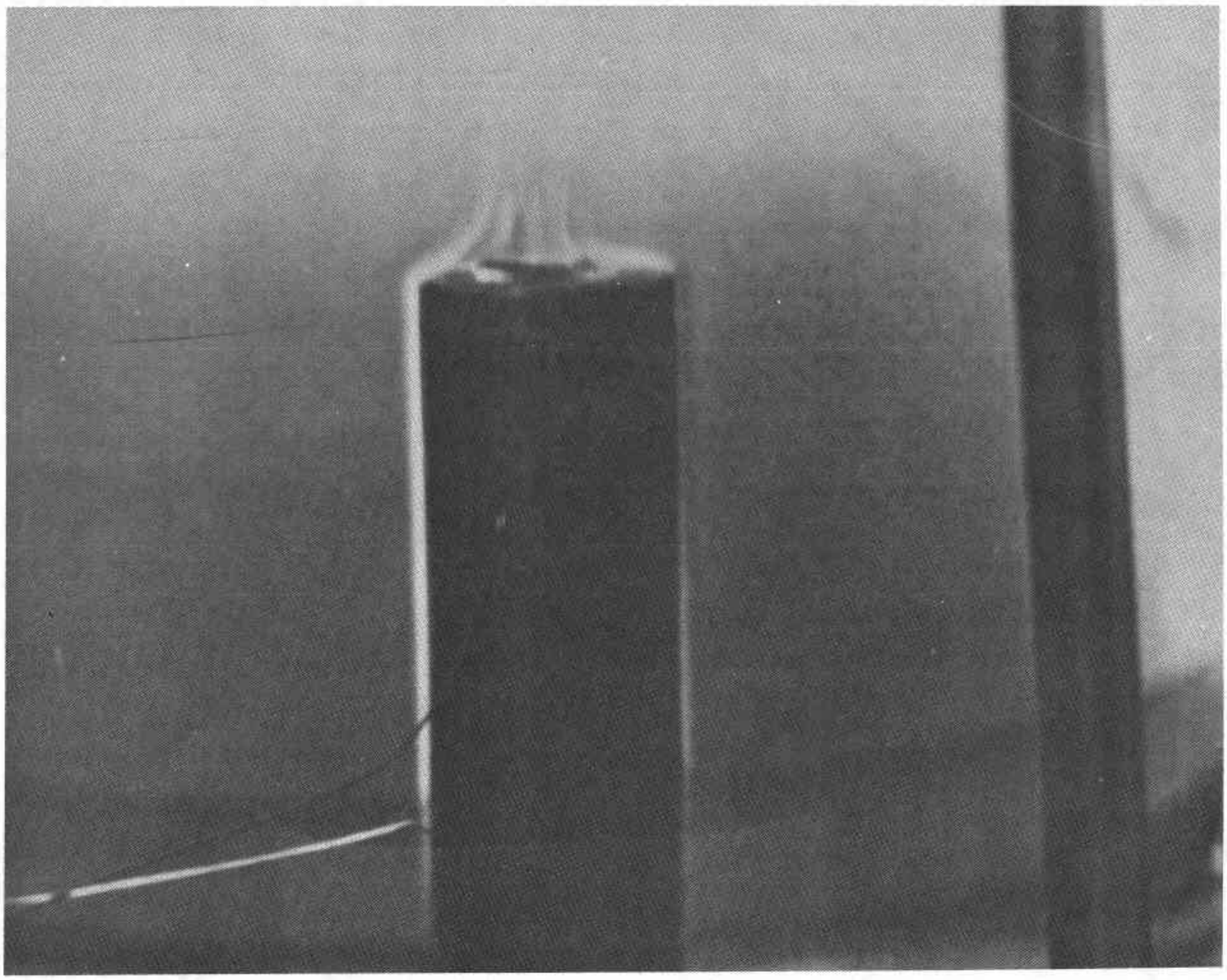


Figure 6-16. Heated Surface Fouling Test at 316°C (600°F); Note Laminar Convective Layer

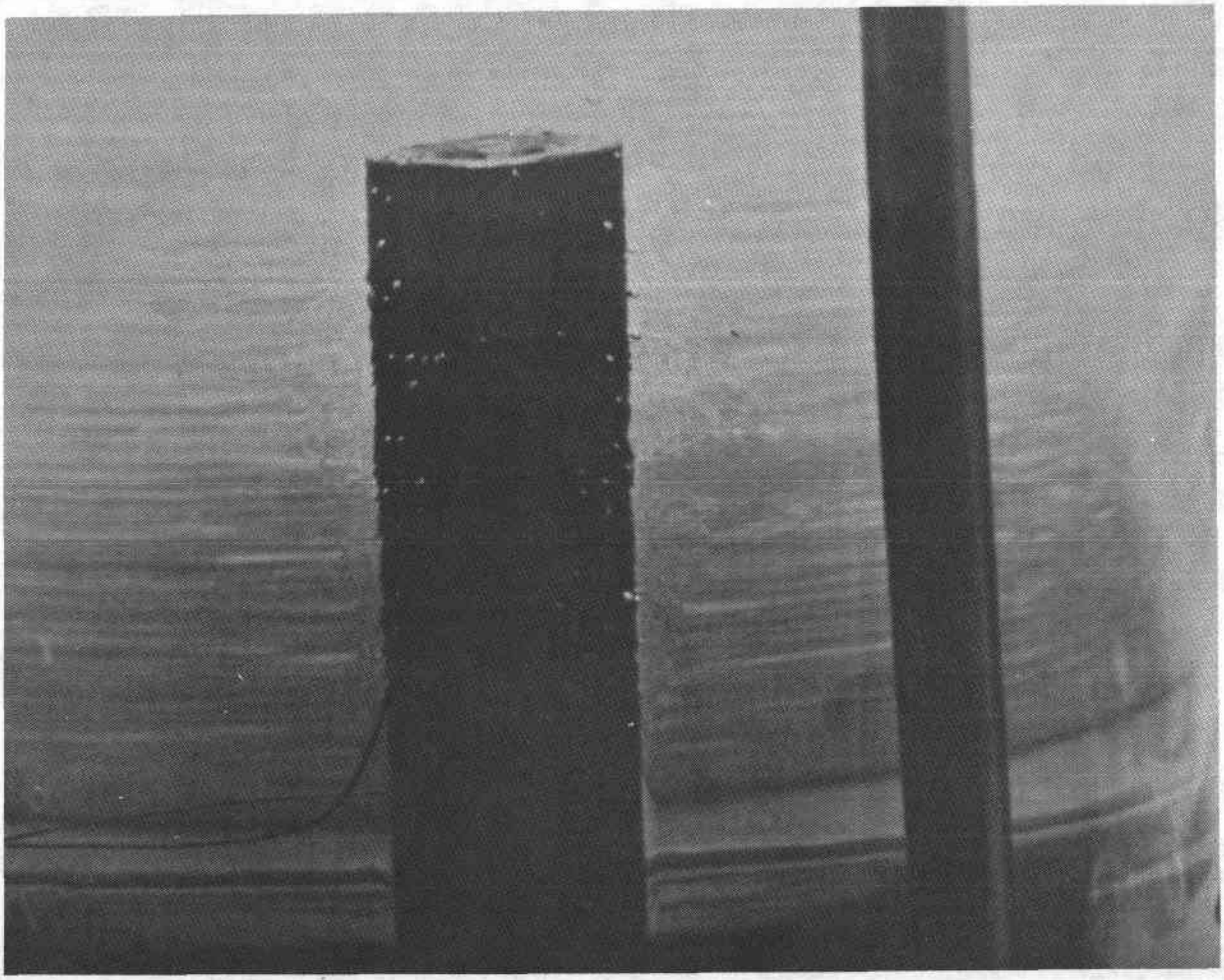


Figure 6-17. Heat Surface Fouling Test at 329°C (625°F); Note Transition from Laminar to Turbulent Flow in the Natural Convection Flow

turbulent) tended to sweep away the bubbles. Hence, very little deposition occurred at the higher wall temperatures. Even for the 316°C test, the early surface deposits did not continue to accumulate but steadily decreased to the point where, after about 2,000 to 2,500 hr almost no material remained. The fouling test for a heater wall temperature of 316°C was later repeated using fresh Caloria HT43 from a different batch than that used in the initial fouling test. The results were the same.

About every 800 hr the Chromel-Alumel thermocouples used to monitor the heater surface temperature for the Caloria HT43 tests would fail at the alumel-heater spot-weld. Similar thermocouple wires used in the Therminol 66 tests were trouble-free. After 2,000 to 2,500 hr, the Chromel-Alumel thermocouples used in the Caloria HT43 tests were replaced with Iron-Constantan.

When the Caloria HT43 fouling tests were temporarily interrupted, the 316°C (600°F) heater had accumulated more fouling deposits than the other two heaters. The patchy fouling deposits were confined mainly to the lower portion of the heater where the convection velocities were lowest. The top of the heaters was clear of deposits but appeared discolored or stained. The data from these tube fouling tests are summarized in Table 6-6.

After several thousand hours the heaters immersed in Therminol 66 appeared somewhat discolored. Patchy deposits sometimes observed near the bottom of Caloria HT43 heaters, were not present with Therminol 66. At the end of 3,745 hr of testing, the deposit on the 316°C (600°F) heater immersed in Therminol 66 was scraped from a known area and weighed. The deposit was found to be 0.0015 kg/m^2, which is negligible.

The Therminol 66 used in the tube fouling tests is still light yellow in color after more than 10,000 hr of heating and is only slightly darker than the fresh fluid. After 6,500 and 7,900 hr of exposure to wall temperatures of 329°C (625°F) and 343°C (650°F), Caloria HT43 had darkened considerably, but appears to be fine for continued use.

Table 6-6

HEATED SURFACE FOULING TEST SUMMARY

Test No.	Fluid	Wall Temp °C (F)	Test Time (hr)	Visual Observations
1	Caloria HT43	316 (600)	6,524	Deposits Near Bottom of Heater
2	Caloria HT43	329 (625)	7,928	Slight Deposit Near Bottom of Heater
3	Caloria HT43	343 (650)	4,804	Slight, Patchy Deposit Near Bottom of Heater
7	Caloria HT43	316 (600)	360	
4	Therminol 66	316 (600)	3,745	No Visible Fouling
5	Therminol 66	329 (625)	10,063	No Visible Fouling
6	Therminol 66	343 (650)	10,231	No Visible Fouling

Conclusions:

1. No Trace of Fouling With Therminol 66
2. No Significant Fouling With Caloria HT43

The power inputs required to maintain the heater surface temperatures at a fixed value were measured periodically. The buildup of a fouling film would cause the power requirement to decrease with time to keep the wall temperature constant. In practice, the use of power measurements in these fouling tests yielded some ambiguous results. However, the data indicate that no significant increase in thermal resistance occurred during tests which correspond to about 4 yr of Pilot Plant operation, for the Caloria HT43 tests.

6.2.4 Current Continuing Tests

The thermal stability tests being continued are marked with an asterisk in Table 6-3. They include three tests with Caloria HT43 and five tests with

Therminol 66. Additional thermal stability and solids compatibility studies will be performed with Therminol 66 and Caloria HT43. In several of these tests, samples of the volatilized species produced by thermal cracking will be collected for lab analysis. These supplementary tests will cover the same temperature range as used in prior experiments, that is, 288°, 302°, and 316°C (550°, 575°, and 600°F).

Thermal stability and surface fouling tests soon will be conducted with another heat-transfer fluid, Mobiltherm 600, which has just been put back into production after 3 yr. Additional test setups are being prepared for a full complement of tests with this fluid.

Six heated surface fouling tests (Table 6-6) are being continued with Caloria HT43 and Therminol 66. Similar tests will be started soon with Mobiltherm 600.

A laboratory-scale thermal storage system is under construction. This flow loop will contain a rock bed thermal storage unit with a charging loop heated by electrical immersion heaters and an energy extraction loop. The lab-scale system is expected to provide valuable information on tube fouling, measurement of fouling capability, fluid pyrolysis rate (both thermal cracking and polymerization) and sand migration.

6.2.5 Conclusions

The long-term thermal stability and material compatibility tests performed with Caloria HT43, Therminol 66, and Therminol 55 indicate that:

1. Therminol 55 was found to pyrolyze too rapidly, over the temperature range of the tests, to be considered for use in the TSU.
2. Caloria HT43 has satisfactory stability and compatibility for use in the Pilot Plant and Commercial Plants.
3. Therminol 66 has satisfactory stability and compatibility for use in the Pilot Plant, but questionable compatibility with rock for use at the higher temperatures of Commercial Plants.
4. Based on the weight loss rate and cost per gallon, Caloria HT43 is much more economical than Therminol 66 for use in the TSU.

5. Based on a given temperature-time cycle for the TSU, the Caloria HT43 weight loss rate by volatilization was calculated to be 7.0 %/yr for the Pilot Plant operating over the range 302° to 218°C (575-425°F) and 12.9 %/yr for the Commercial Plant operating from 316° to 232°C (600° to 450°F).
6. The data on the rate of fluid weight loss for Caloria HT43 was correlated by the equation

$$R = 5.38 \times 10^{10} \exp[-17650/T]$$

and for Therminol 66 by the equation

$$R = 1.93 \times 10^{27} \exp[-39580/T]$$

where R is weight loss in %/hr, and T is temperature in °K.
7. The high experimental weight loss rate found for Therminol 66 may indicate an incompatibility of the fluid with either the rock, carbon steel, or stainless steel added to some of the fluid samples.
8. There was no indication of any significant amount of polymerization in any of the Caloria HT43 tests.

The heated surface fouling tests conducted with Caloria HT43 and Therminol 66 indicate that:

1. There are no significant problems with heat exchanger fouling with either Caloria HT43 or Therminol 66 for heater wall temperatures up to 343°C (650°F).
2. Caloria HT43 can form patchy fouling deposits on regions of the tube heater where the fluid velocities is low in natural convective flow.

Overall, the various fluid tests performed already under this program (with test times equivalent to over 4 yr in Pilot Plant operation) indicate that Caloria HT43 is an excellent choice for the preliminary designs, and that it will meet all requirements for the solar power application under consideration.

6.3 SUBSYSTEM TESTS

6.3.1 Purpose

The test program was conducted to provide data for the preliminary design of the TSS portion of the Solar Thermal Central Receiver program. The program was designed primarily to establish characteristics of TSU that would be scalable to the 10-MWe Pilot Plant (subsystem schematic shown in Figure 6-18). Additional system characteristics such as control and start stop transients were to be investigated where feasible. Specific design test objectives were:

A. Evaluate charging and extraction of a scalable TSU.

B. Obtain performance of the TSU for equivalent Pilot Plant conditions.

C. Demonstrate stable operation for high, low, intermittent, and no insolation.

D. Demonstrate mode changeover and emergency operation.

E. Evaluate fluid under operational conditions.

F. Determine the strain from rock/wall interaction.

6.3.2 Test Hardware and Installation

The complete thermal storage model subsystem tested consists of a thermal storage unit, two fluid circulation pumps, ullage maintenance unit, fluid maintenance unit, fluid heater, steam generator, interconnecting fluid lines, and control valves and sensors. A schematic of the system is shown in Figure 6-19.

6.3.2.1 Test Installation

The subsystem was assembled at Rocketdyne by installing the TSU and associated components in an existing steam generation system at Rocketdyne's Santa Susana Field Laboratory (SSFL), Figure 6-20. The existing steam generation system contained a fluid heater and steam generator and was designed to use a high-temperature heat-transfer fluid up to 316°C (600°F) for transferring heat identical in principle to the central receiver pilot and Commercial Plant thermal storage systems.

Characteristics of the various components are listed in Table 6-7. Details of other components are omitted from the preliminary report, but are published in Ref. 6-18, 6-19, and 6-20.

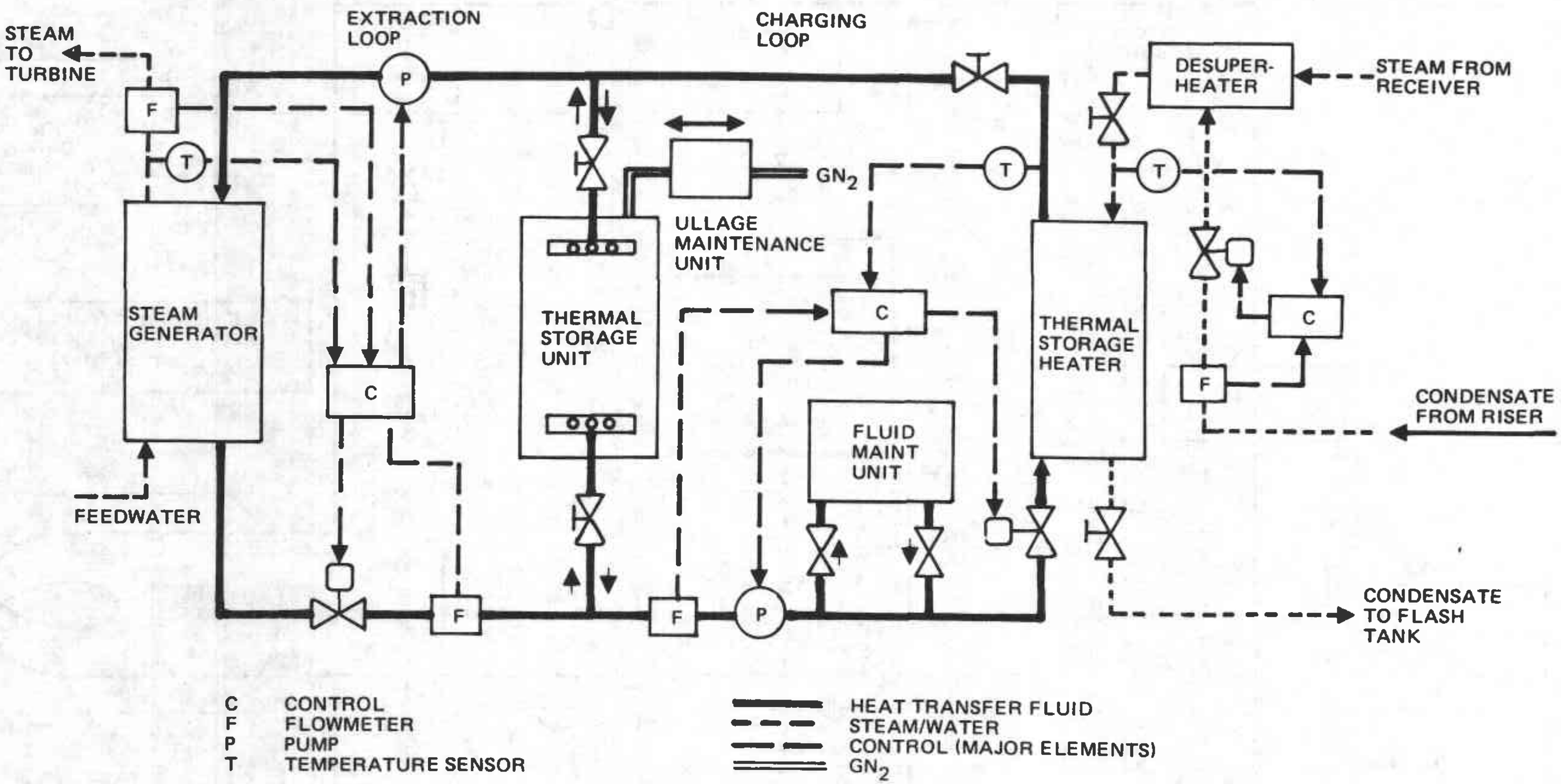


Figure 6-18. 10-MWe Pilot Plant Thermal Storage Subsystem Schematic

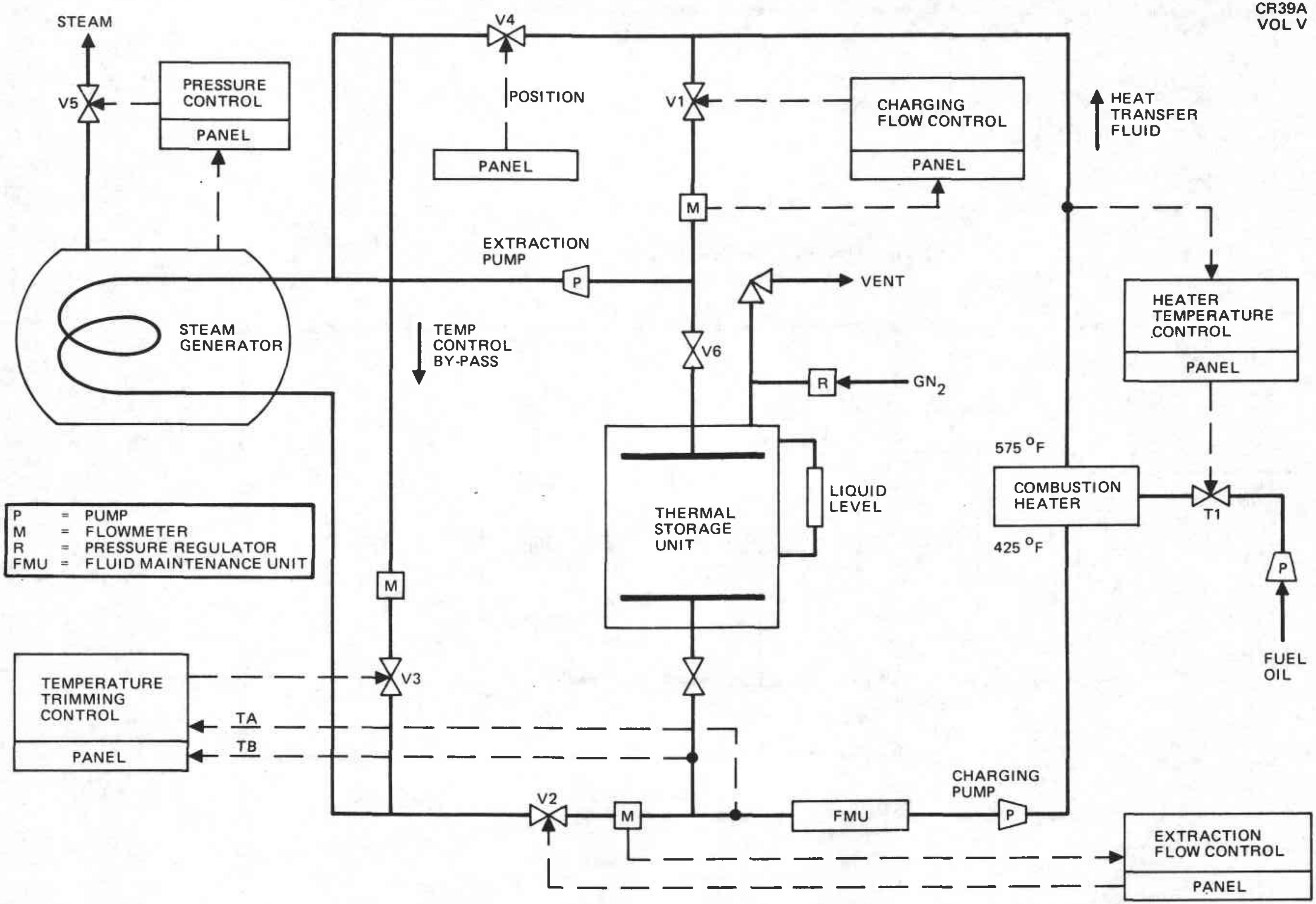


Figure 6-19. Subsystem Research Experiment (SRE) Schematic

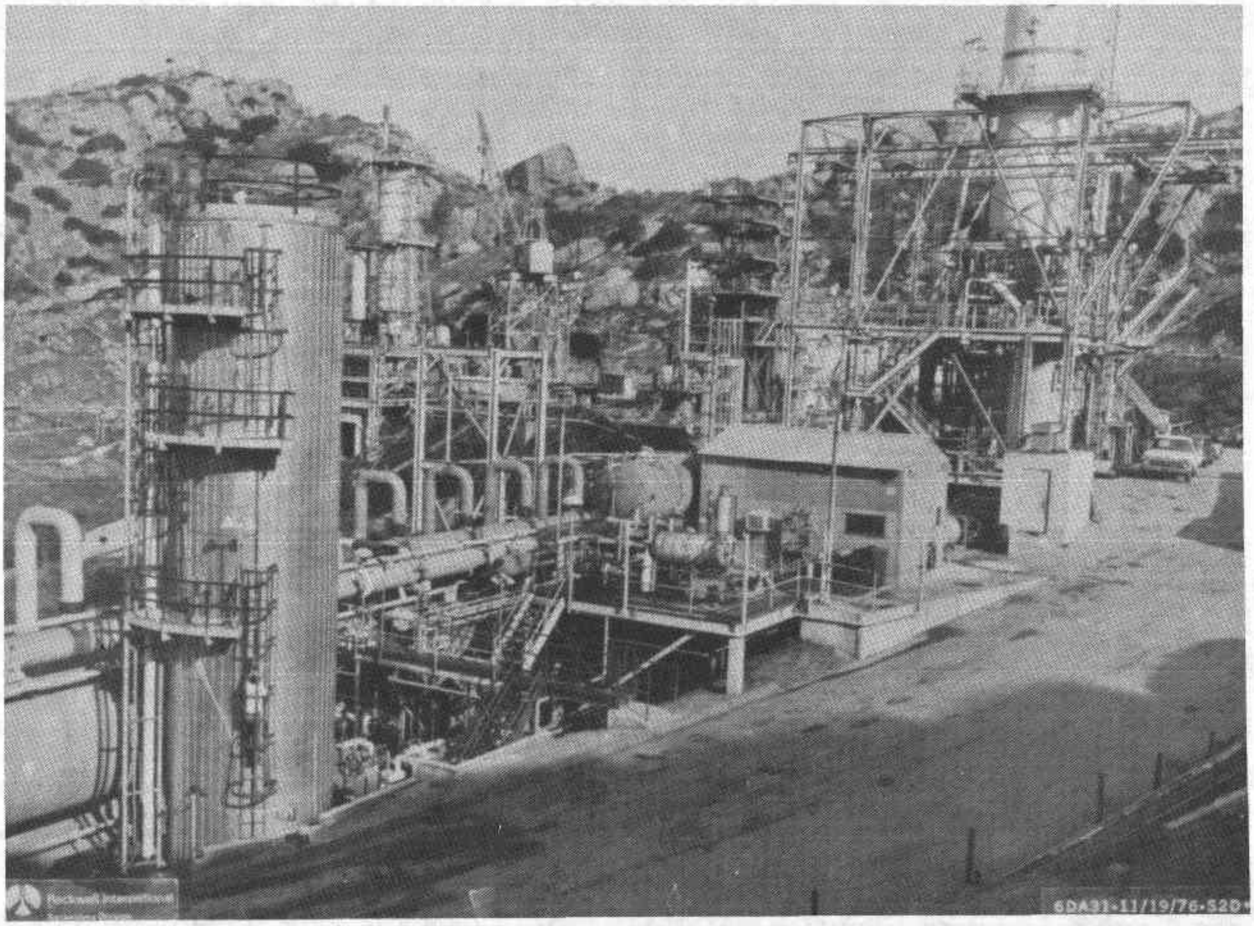


Figure 6-20. Rocketdyne Santa Susana Field Laboratory Bowl Area

Table 6-7

THERMAL STORAGE MODEL SUBSYSTEM
PRINCIPAL COMPONENTS

*Thermal Storage Heater (1)	Combustion, 1-MWth Capacity
Steam Generator (1)	2-MWth Steam Generation Capacity
Thermal Storage Unit (1)	4-MWhth Design Capacity, 0.1 MWth to 5-MWth Rate
Fluid Circulation Pumps (2)	100 gpm, 330-ft head, Dean Bros., High-temp, Centrifugal
Data Logger (1)	Multiplex, Print, Record, Read, 100 Channels
Ullage Maintenance Unit (1)	GN_2 Supply and Vent to Atmosphere
Heat Transfer Fluid	Caloria HT43, 7,000 gal
Fluid Maintenance Unit (2)	Strainer and Filter (Replaceable Cartridge)
Valves and Controls	
V1	Charging flow control (fluid)
V2	Extraction flow control (fluid)
V3	Temperature trimming control (fluid)
V4	Charging by pass control (fluid)
*H1	Charging heat control

*Existing Facility

6.3.2.2 Thermal Storage Unit (TSU)

The TSU is a welded steel storage tank with overall interior dimensions of 3.2m (10.5 ft) diameter by 13.3m (43.7 ft) high (Figure 6-21, Table 6-8). The vessel is filled to a height of approximately 12.5m (41.0 ft) with granite rock with diameter of approximately 25 mm (1 in.) and sand approximately 1.5 mm (0.06 in.). The tank top is removable for installation of the rock and the fluid distribution manifolds (Figure 6-22) which are located near the top and at the bottom of the tank and provide uniform flow through the rock during the charging and heat-extraction cycles. The exterior of the vessel is covered with insulating material to minimize heat losses during tests and to provide maximum energy recovery during heat extraction. Figure 6-23 shows the TSU in place at the steam generation facility.

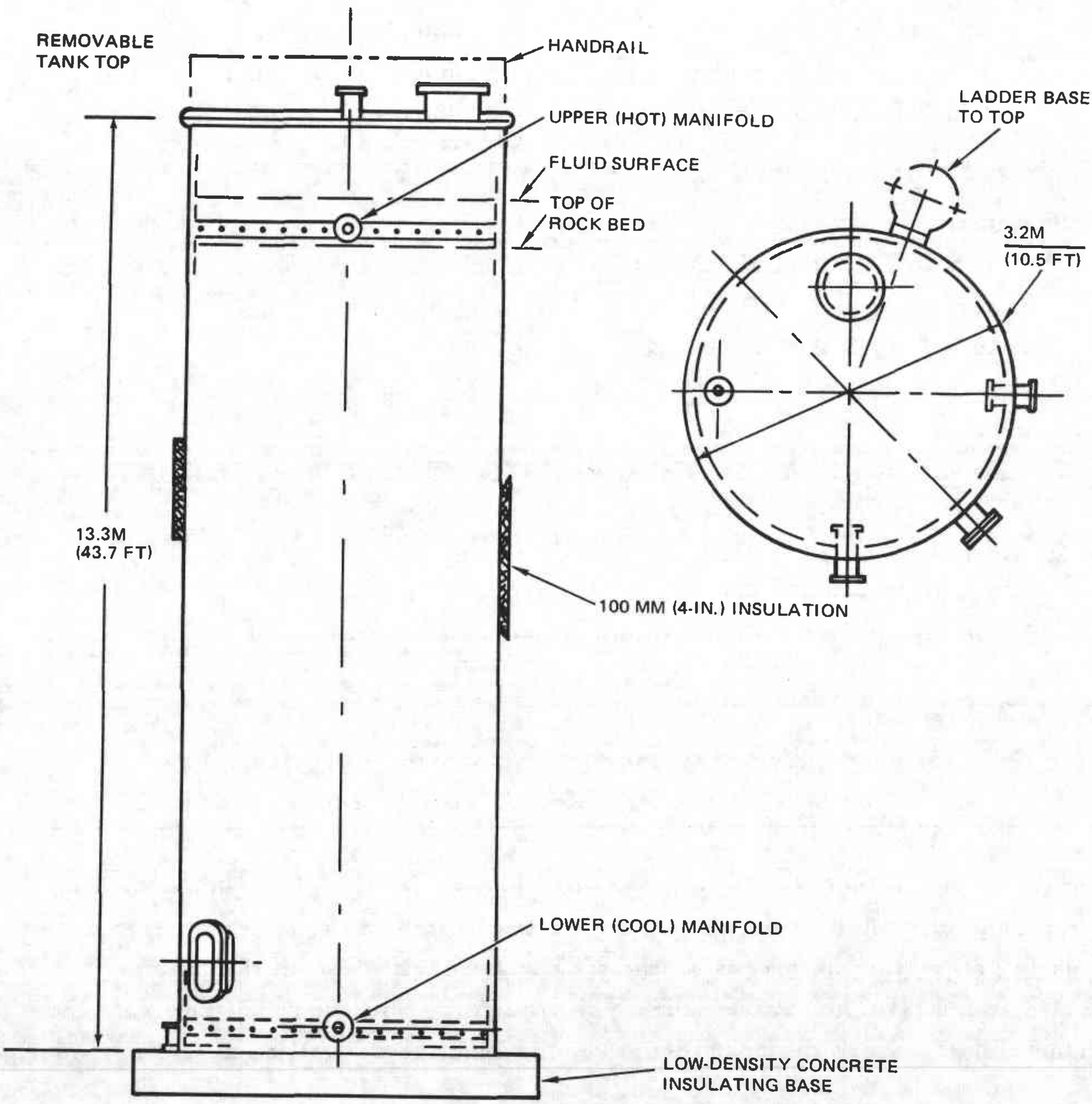


Figure 6-21. SRE Thermal Storage Unit (TSU)

Table 6-8

SRE THERMAL STORAGE UNIT DESCRIPTION

- Scalable Size

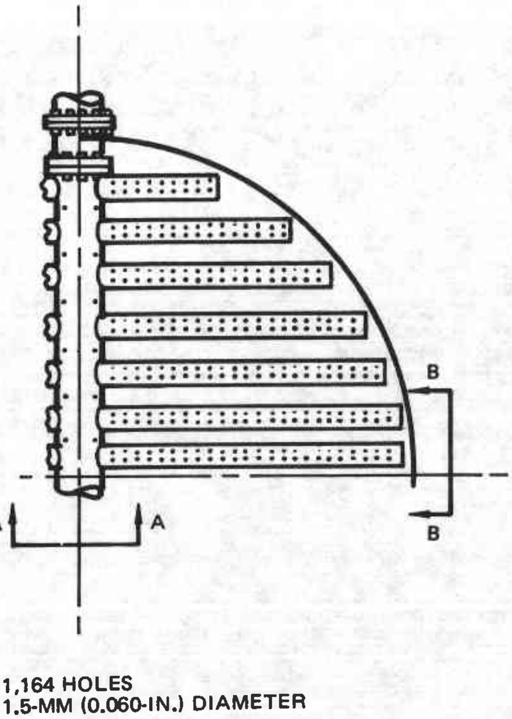
5-MWHt Capacity	107,000 liters (28,300 gal) volume
1-MWt Normal Operating Rate	209,000 kg (231 ton) rock/sand
5-MWt Maximum Design Rate	26,500 liters (7,000 gal) caloria Caloria HT43

- Structural
  - ASTM A537 Class 2 Steel
  - Fabricated to ASME Section VIII Div 2 Standards
  - Basic Design API Code
- Foundation
  - Low-density Insulating Concrete, (70% reduction in conductivity)
- Insulation
  - Owens-Corning Intermediate Hardboard (Fiberglas)
  - 0.38 mm (0.015 in.) Aluminum Weather Cover
- Instrumentation
  - 70 Thermocouples for Measuring Performance and Heat Loss
  - 20 Strain Gages for Measuring Rock/Wall Interaction Stress

Instrumentation includes (1) temperature probes located throughout the bed for thermocline definition, (2) strain gages located in the lower portion of the tank to determine the stress on the tank wall resulting from the thermal cycling, and (3) temperature surveys through the rock bed and tank wall and foundation to determine heat losses during inactive as well as active periods.

6.3.2.3 Instrumentation

Instrumentation for determination of heating rates, heat stored, heat leakage, bed and manifold hydraulic characteristics, and thermocline longitudinal and lateral configuration and tank wall stress is listed in Table 6-9.



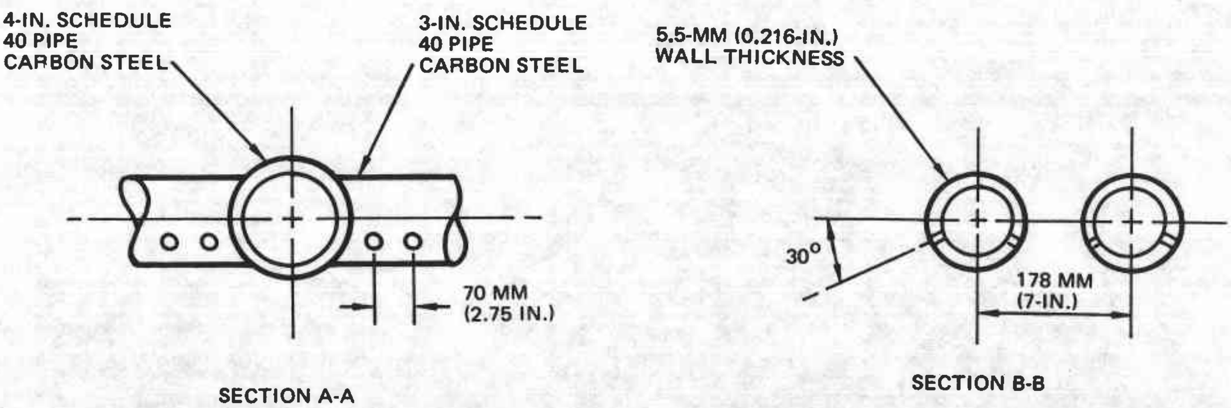


Figure 6-22. TSU Fluid Distribution Manifolds, Bottom and Top Identical

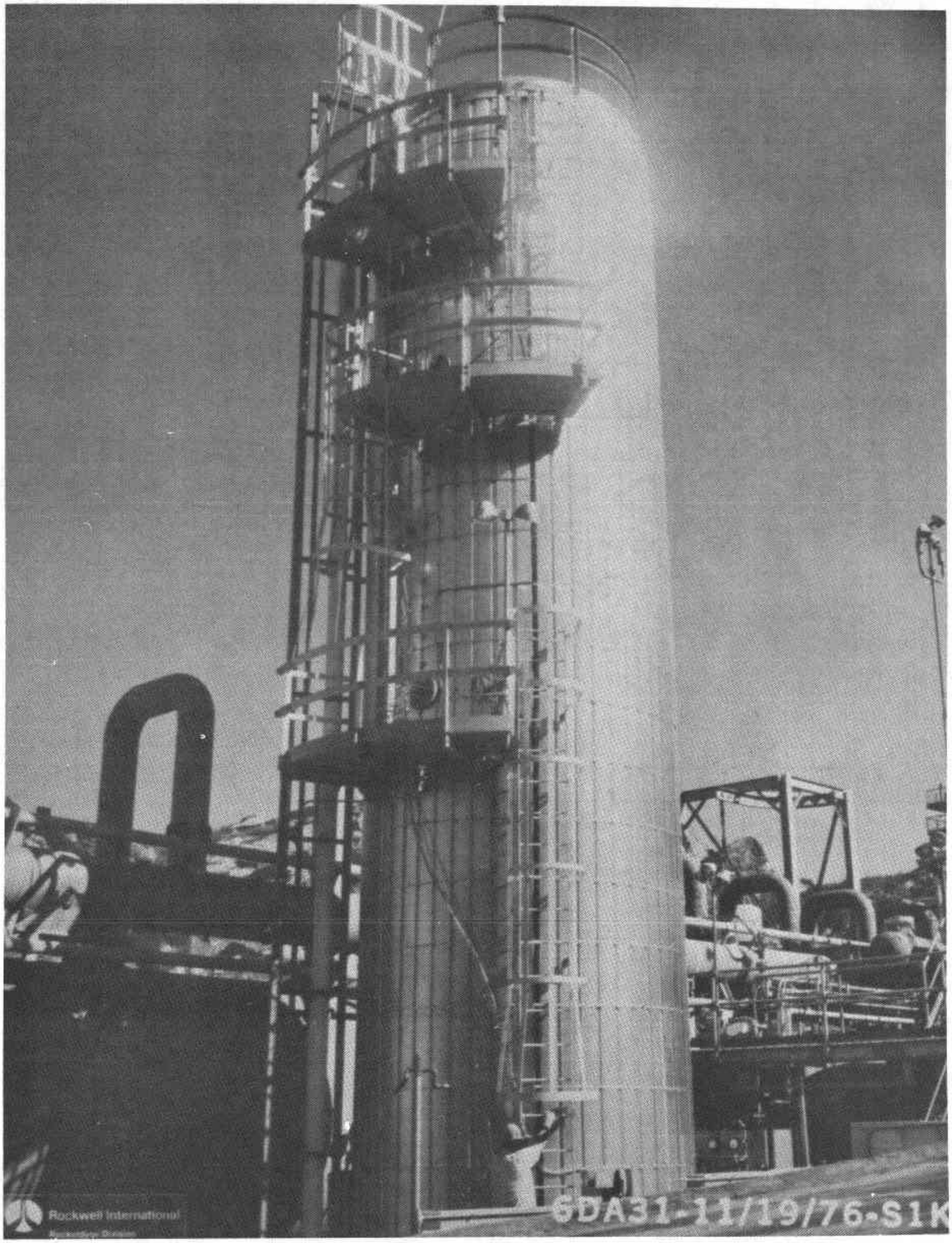


Figure 6-23. Thermal Storage Unit Installed at SSFL Test Site

Table 6-9

THERMAL STORAGE SRE INSTRUMENTATION RECORDED

Measurement	Purpose	Type Sensor	No.
Rock Bed Temperature	Thermocline Characteristics	Thermocouple Probe	32
Axial	Thermocline Characteristics	Thermocouple Probe	27
Transverse (3 Planes)	Thermocline Characteristics	Thermocouple Probe	2
TSU in-out (Fluid)	TSU Performance		
Tank Wall Temperatures	Stress Correlation and Heat Loss	Thermocouple-Weld	7
Insulation Temperature	Heat Loss (Tank)	Thermocouple-Fasten	9
Foundation Temperature	Heat Loss (Foundation)	Thermocouple-Inbeded	3
Tank Wall Strain	Rock and Thermal Stress	Strain Gage-Welded	20
Fluid Flow	Control/Heating Rate	Turbine	3
TSU Manifold Pressure	Characteristic	Transducer	1
Rock Bed ΔP	Characteristic	Transducer	1
TSU Ullage Pressure	Control	Transducer	1
Pump Outlet Pressure	Control	Transducer	2
Steam Generator Steam Pressure	Control	Transducer	1
Steam Generator Fluid Pressure	Control	Transducer	1
Heater Fluid Temperature	Control/Heating Rate	Thermocouple	1

Thermocouple locations for determining heat loss, heat capacity, and thermocline characteristics in the TSU are shown in Figure 6-24. Thirty-two sensors located on the TSU centerline provide the longitudinal temperature profile of the thermocline over the complete range from zero to fully charged. Twenty-seven sensors in three locations provide lateral characteristics of the thermocline when it passes these stations as well as continuous radial temperature profiles in the bed. Thermocouples fastened to the tank wall and insulation weather cover establish heat loss through the wall. These, together with thermocouples located at the top of the TSU and embedded in the foundation, provide a measurement of heat leakage. Thermocouples are Iron-Constantan, NBS type B, Circular 561.

Tank/rock stress interaction is established by 20 strain gages concentrated primarily in the lower portions of the tank (Figure 6-25). The strain gages use wire alloyed to duplicate the coefficient of expansion of steel. Further temperature compensation is accomplished by use of inactive branches of the bridge that are mounted adjacent to the active branch. Strain gages are BLH welded, type FNW36-50-12.

The remaining instrumentation serves to measure heat and flowrates and provide control information to operate the systems. Pressure transducers are standard bonded construction with a nominal full-scale output of 20 mV. Flow measurement is conducted with turbine type transducers. Output is DC after conversion by an Anadex Model 700 signal converter.

Data recording was done manually and automatically. Manual data recording involved slow transients (fluid level, weather, etc.), together with visual gage verification of pressure readings.

Automatic data recording was accomplished with a Doric Model 220, 100-channel data logger. Sweep frequency varied from 20 sec to 40 sec per sweep depending upon the selection of recording mode. Records with punch tape only allowed 20-sec sweeps. Recordings using the printed tape required a 40-sec sweep. Both manual and punched tape data were translated to magnetic tape which facilitated data analysis. The magnetic tape provides

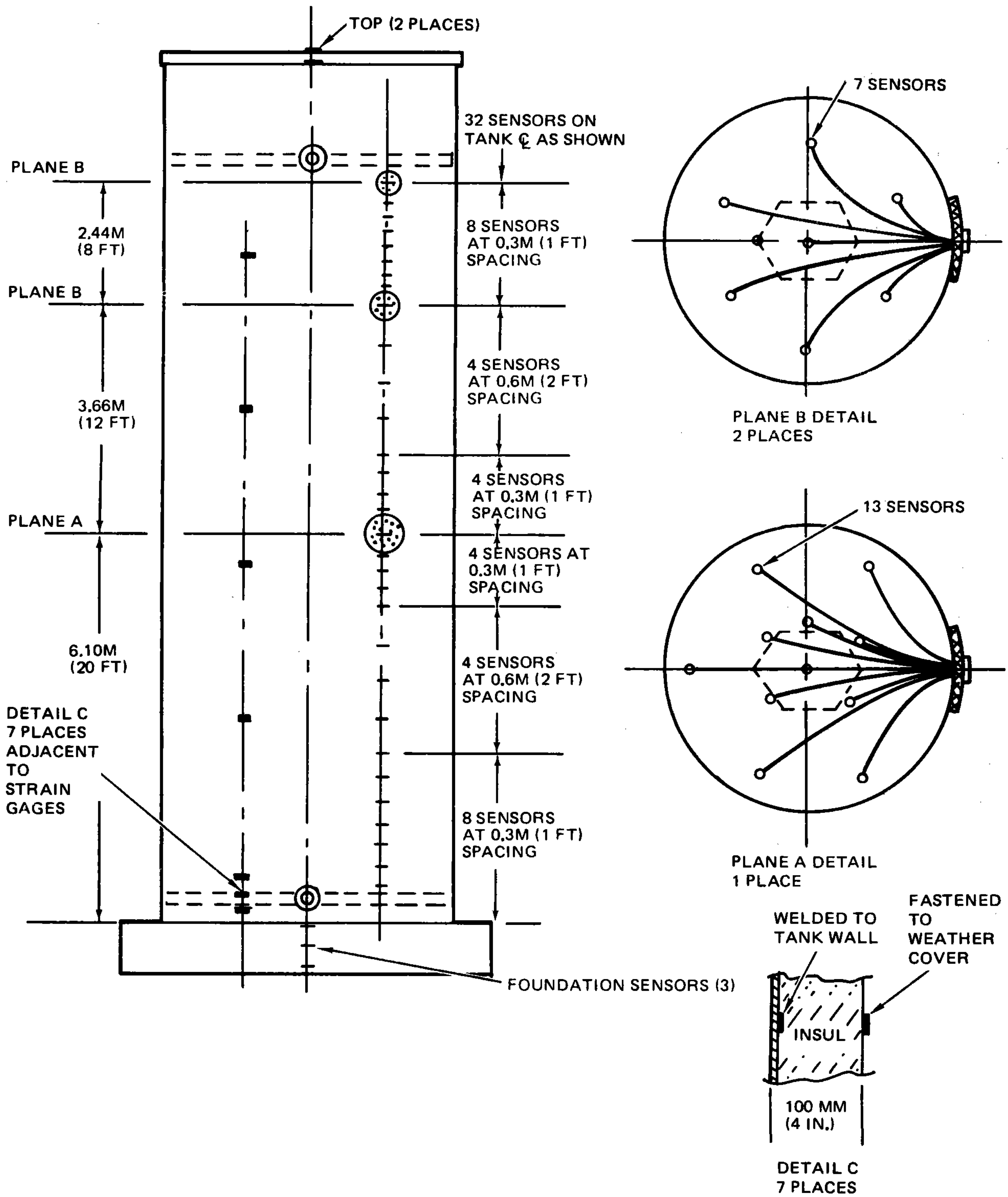
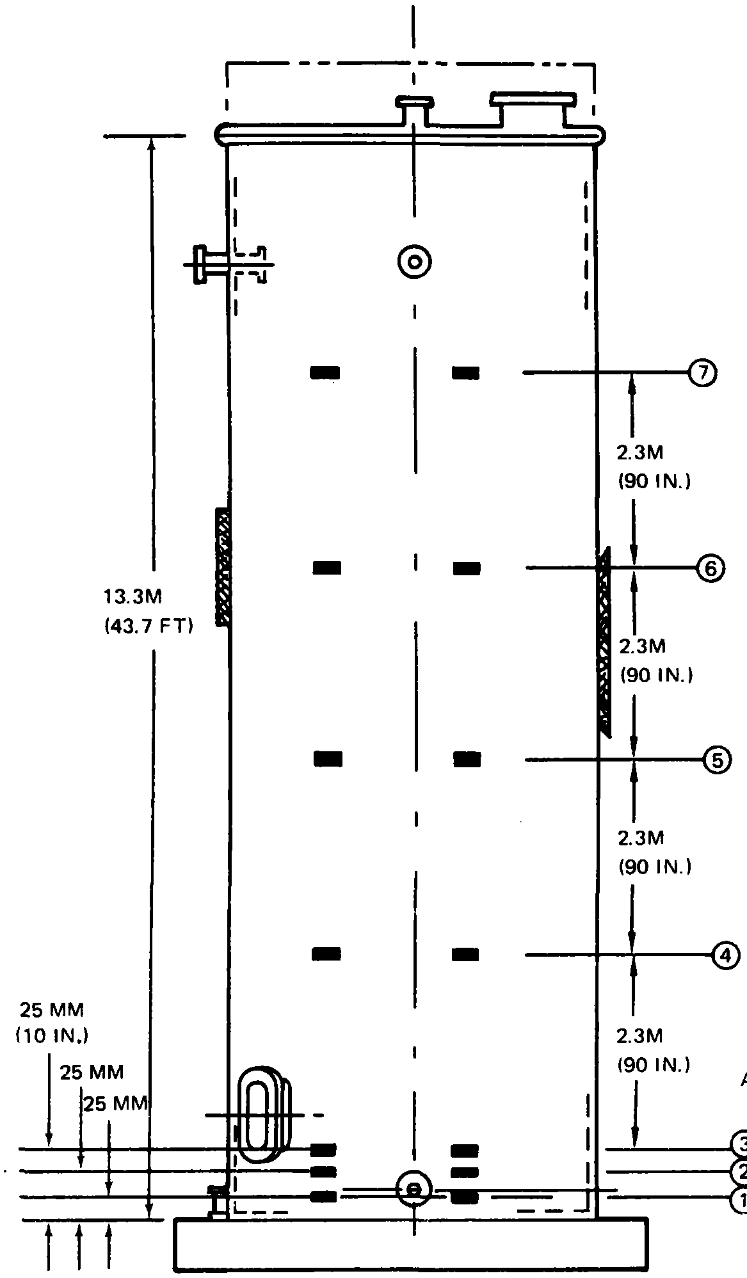
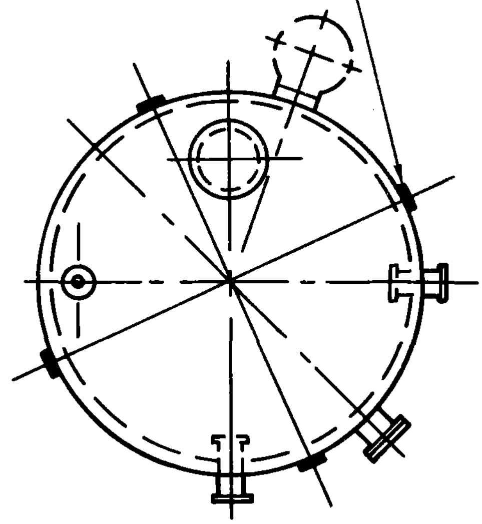


Figure 6-24. Thermal Storage Unit Temperature Sensors, All Sensors Iron-Constantan





STATION	NUMBER OF GAGES (SEPARATED BY 90 DEG)
1	4
2	4
3	4
4	2
5	2
6	2
7	2

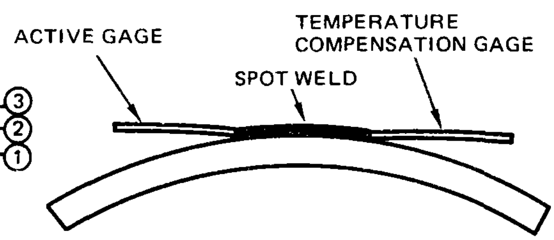


Figure 6-25. Strain Gage Locations, Thermal Storage Unit

listed as well as cathode ray tube (CRT) displays of all data with Julian day, hours, minutes, and seconds identification of each data slice.

6.3.3 Test Program

The test program was structured to provide a maximum amount of information for final design of the Pilot Plant TSS. Testing covered a wide range of flow rates and hold periods to characterize the system in the following areas:

- TSU performance map (steady state operation)
- Start/stop and changeover characteristics
    - Time to energy availability
    - Energy output for various hold times
    - Control error band and thermal lag
    - Emergency shutdown
    - Warmup
- Response to rapid energy changes (nonsteady state)
- Steady-state temperature control
- Heat losses
- Fluid stability and particulate measurements
- Tank/bed structural loads

6.3.3.1 Test Description

Testing covered the period from mid-October to mid-December 1976 and included 56 test data gathering periods. Testing was divided between active and inactive periods. Active testing involved charging or extracting heat from the TSU. Inactive periods were defined as "holds" between charging and extracting. Table 6-10 summarizes all tests.

Tests 01 through 10 covered a period of 1-wk and provided bed conditioning (warmup and water removal). During this period a relatively large amount of water was removed from the bed by vaporization and discharge through the ullage regular and emergency overflow vent lines.

Tests 12 through 50 provided overall performance and thermocline characteristics for a range of charging and extraction rates and hold periods. The tests were primarily conducted for Pilot Plant design conditions of 306°C

Table 6-10 (Page 1 of 2)
SRE SUBSYSTEM TEST SUMMARY

Test No.	Operating Mode	Temperature (°F)	Energy Rate (MWth)	Duration (hr)
1	Bed Conditioning	70 to 110	0.8	0.3
2	↓	70 to 110	0.8	0.5
3	↓	70 to 225	0.8	0.4
4	↓	70 to 225	1.0	5.4
5	Hold	225	-	19.0
6	Bed Conditioning	325	0.6 to 1.2	3.7
7	Hold	325	-	15.3
8	Bed Conditioning	350	0.7	3.6
9	Hold	350	-	47.5
10	Bed Conditioning	350 to 400	1.0	2.5
11	Hold	350 to 400	-	20.0
12	Charge	425	1.0	4.8
13	Hold	425	-	18.8
14	Charge	450	1.0	1.9
15	Hold	425	-	20.7
16	Hold	425	-	122.5
17	Charge	450	0.3 to 1.1	2.9
18	Hold/Charge	425 - 475	-	48.0/1.5
19	Hold	425	-	18.3
20	Charge	575	1.0	3.0
21	Hold	425 - 575	-	99.1
22	Charge	575	1.1	4.6
23	Hold/Charge	575	-	19.5
24	Charge	575	0.5 to 1.2	9.4
25	Hold	575	-	143.9

Table 6-10 (Page 2 of 2)

SRE SUBSYSTEM TEST SUMMARY

Test No.	Operating Mode	Temperature (°F)	Energy Rate (MWth)	Duration (hr)
26	Charge	575	0.8 to 1.2	9.6
27	Hold	575	-	110.3
28	Charge	575	0.8 to 1.2	9.1
29	Extract	575	0.4	3.7
30	Hold	425 - 575	-	34.5
31	Charge	575	1.0 to 1.2	7.4
32	Extract	575	1.3	4.7
33	Hold	425	-	93.9
34	Hold	425	-	-
35	Charge	575	0.6	7.2
36	Extract	575	0.6	5.2
37	Hold	425 - 575	-	17.3
38	Charge	575	0.8	4.2
39	Hold	575	-	34.1
40	Charge	575	1.1 to 0.1	10.0
41	Extract	575	0.1 to 1.1	4.0
42	Hold	425 - 575	-	58.0
43	Extract	575	1.8	3.5
44	Hold	425 - 575	-	26.5
45	Charge	575	0.7	3.1
46	Hold	575	-	17.6
47	Charge/Extract	575/575	0.7 to 1.7	3.5/3.5
48	Hold	425 - 575	-	7.6
49	Charge	575	0.1	1.0
50	Hold	425 - 575	-	25.0
51	Duty Cycle	575/600	0.1 to 2.0	18
52	Hold	575	-	87.3
53	Extract	575	2.0	3
54	Hold	400	-	13.5
55	Extract	350	2.0	5
56	Hold	300	-	(6 wks)

(575°F) maximum fluid charging temperature and fluid return temperature of 218°C (425°F).

Test 51 simulated a typical duty cycle with a variety of charging rates, holds, and extraction rates for a typical 18 hour day (6 AM to 12 M).

Tests 52 through 55 represent holds and extractions at lower temperatures. Test 56 was the final cooldown to ambient temperature.

6.3.3.2 Operating Procedures

Testing involved the three primary modes (charge, extract, hold). System operation during the two active modes is described below. During charging and extraction data logger scans were varied from maximum frequency (20 to 40 sec) to a period varying from 2 to 5 min depending upon rate of change of system operating conditions. (During hold periods, the data logger was set to scan automatically from 1 hr to 12 hr and operated unattended.) Warming the lines external to the TSU generally took approximately 10 to 15 min after initial heater startup. The data logger was not activated during the system warmup period. After warmup, the fluid was diverted through the TSU with heat addition (charging) or removal (extraction) as desired. Operating procedures are described below.

Thermal Charging

During charging (Figure 6-26) the heat-transfer fluid exits the TSU through the lower manifold; passes through the FMU, charging pump, combustion heater, Valve V1, flowmeter M; and reenters the TSU through the upper manifold. The flowrate of hot fluid entering the TSU is controlled by V1, a tapered plug globe valve. A panel control allows the operator to select a range of flows from maximum, 4.3 kg/sec (9.5 lb/sec), to approximately 10% of maximum. Because of limitation of the combustion heater, a second loop through V4, V3, and the steam generator is needed to accurately control the return fluid temperature, T3A. Table 6-11 lists the operational sequence for warmup and charging.

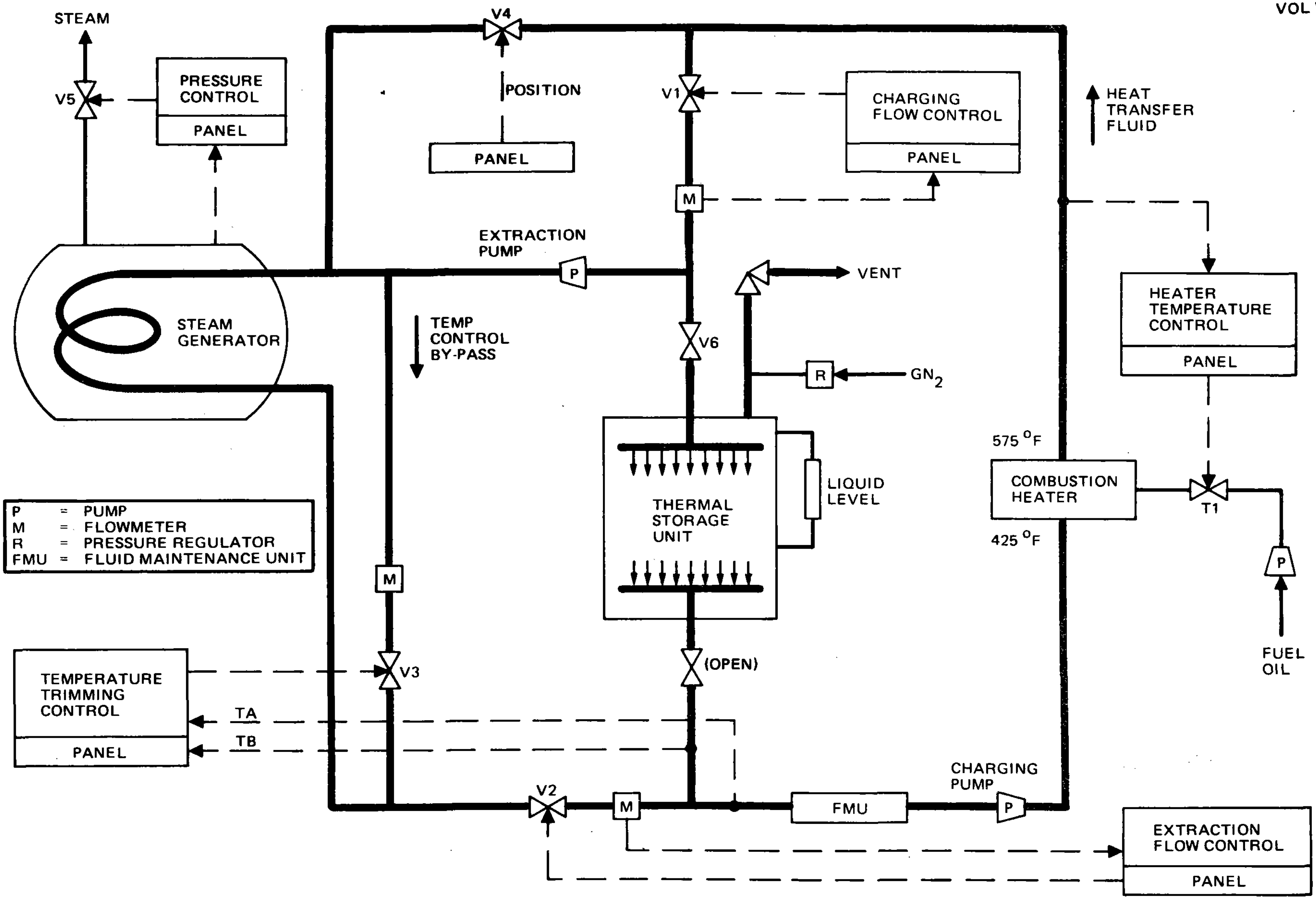


Figure 6-26. SRE Model Substem Operation — Charging

Table 6-11

TEST OPERATIONS SEQUENCE
CHARGING

Mode	Operation
Warmup (Lines, Pumps)	Switch V3 to sense TA
	Close V1, open V2, V3, V4, and V5
	Set heater temperature control, T1 (575°F)
	Start charging pump
	Activate heater
	When heater outlet fluid temperature reaches 575°F reduce V4 flow to bypass values desired for charging TSU
Charging	Open set V1 to desired charging flow
	Reduce V4 until heater reaches mid-range of operating limits
	Adjust V3 and V4 during test to keep heater within operating Range (as steam generator warms up)

The return fluid temperature is controlled by V3, which allows uncooled fluid to mix with cooled fluid from the steam generator to provide a constant fluid temperature into the fluid water. The fluid heater temperature control is set to provide an outlet temperature of 302°C (575°F) during charging operation. During warmup and preheat, the system was operated in the charging mode with the heater outlet temperature set to a lower value.

Heat Extraction

Heat is extracted from the TSU (Figure 6-27) by circulating the fluid upward through the TSU and out the upper manifold through the extraction pump, steam generator, temperature-control bypass loop, V3, V2, flow meter M and back into the TSU through the lower manifold. Heat extraction from the fluid can be controlled by the water temperature in the steam generator which is determined by the steam back-pressure control valve V5. Steam pressure establishes the water temperature when boiling is occurring through the saturation pressure-temperature relationship. For most of the test covered by this report it was found to be more convenient during heat

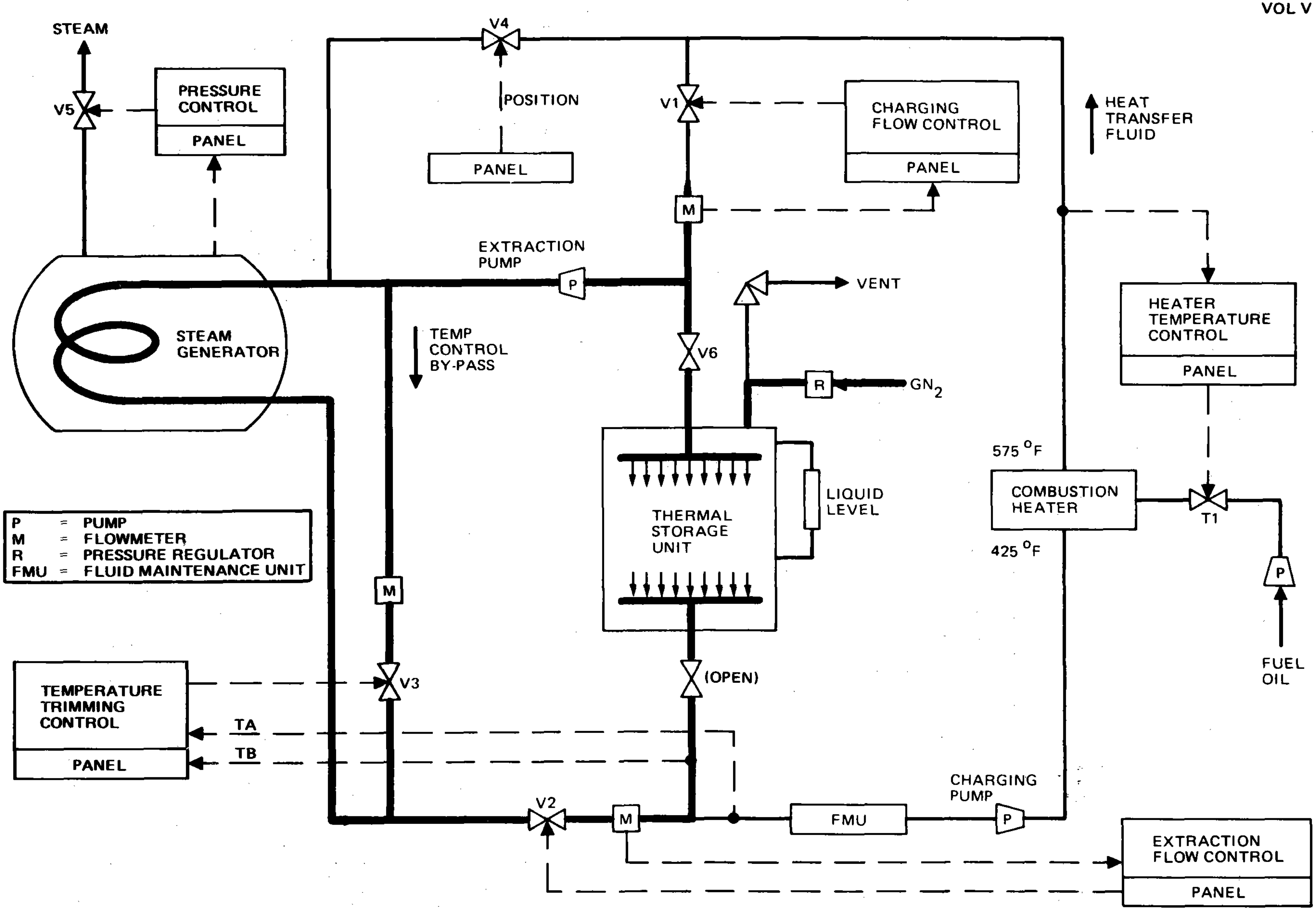


Figure 6-27. SRE Model System Operation — Extraction

extraction to use the sensible heat of the large amount of water in the steam generator as a heat sink instead of generating steam. The fluid leaving the steam generator is overcooled and fluid temperature returning to the TSU was adjusted automatically by mixing hot fluid directly from the outlet of the circulation pump. This allowed accurate control of the thermocline in the TSU without complications from the high thermal inertia of the steam generator. Table 6-12 lists the operational sequence for warm-up and extraction.

6.3.4 Test Results

Results of the thermal storage subsystem tests are described below.

6.3.4.1 Performance and Thermocline Characteristics

The use of a single tank with a solid medium for sensible heat storage of thermal energy depends directly on the ability to create and retain a narrow region of temperature continuity (thermocline) between the hot and cooler regions. Tests conducted covering a thermal charging and extraction range of 0.1 MW to 2 MW and for hold periods of 0 to 100 hr indicate that thermocline performance is reliable and predictable and its shape and characteristics directly predict fluid temperature characteristics as well as flows, and pressures for selected tests are shown in Figure 6-28 through 6-53.* The data are presented as computer CRT displays directly as raw data from the recording equipment.

Figure 6-28 thru 6-34 contain typical performance data for this storage unit, operating at the temperature range planned for the ERDA 10-MWe Pilot Plant (575° to 425°F).

Figure 6-28 shows thermoclines during a charge to full capacity after a 144-hr hold from full charge. Figure 6-29 is a 3.7-hr extraction after full charge.

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*To avoid interrupting text, Figures 6-28 through 6-53 are inserted after Page 6-69.

Table 6-12

TEST OPERATIONS SEQUENCE
EXTRACTION

Mode	Operation
Warmup (Lines, Pumps)	Close V1, V3, and V6, open V2 and V4
	Set V5 to desired steam saturation temperature
	Set heater temperature control, T1 (575°F)
	Start charging pump
	Switch V3 to sense TB
	Minimize V3 flow (to maximize steam generator warmup)
	Terminate warmup after V5 vents and water temperature and steam pressure match desired saturation conditions
	Stop charging pump
	Close V4, open V6
Extraction	Set V2 to desired extraction flow
	Switch V3 to sense TB
	Start extraction pump
	Adjust V5 so that V3 flow is adequate to accurately control TSU inlet temperature to 425°F

Figure 6-30 best shows thermocline performance. Each curve represents the temperature profile (thermocline) at a particular time (the time key is shown in the able at the top of the figure). At the start of the extraction test depicted, the entire TSU was charged to 575°F. Shortly after the start of extraction (curve 1) a small portion of the bottom of the TSU (zero axial distance is the bottom) had dropped to 425°F, but the remainder of the tank, above a sharp thermoincline, remains at the top operating temperature, 575°F. As energy extraction continues, the thermocline moves upward in the TSU (e.g., Curves 2 through 7), with the top of the TSU remaining at the upper operating temperature until almost all of the stored energy is extracted. Then, the thermocline begins to "break through" the top of the bed and the top temperature begins to fall off (e.g., Curve 8).

The temperature of the hot fluid delivered from the top of the TSU is shown in the top half of Figure 6-31. An expanded temperature scale is used to magnify variations in fluid temperature. It can be seen that the temperature is very flat (2°F, which is well within the "noise level" for temperature measurements on such a process) throughout the extraction until very near the end, when the temperature begins to tail off, as predicted by Rocketdyne's computer model of the dual-medium thermal storage system. A temperature of 560°F has been selected as a practical cutoff point to terminate extraction and begin recharging (560°F represents an excursion of only 10% of the operating temperature range of 425 to 575°F). In this test 5.1 MWHth of energy was delivered with fluid temperatures above 560°F, corresponding to a volumetric extraction efficiency of 87% (i.e., 5.1 MWHth is 87% of the energy capable of being stored in the volume occupied by the rock and fluid between the temperature limits of 425° and 575°F). This performance is even better than estimated during design for a unit of this size. Further, the extraction efficiency increases as the TSU size and capacity increase.

Figures 6-32, 6-33, and 6-34 show the flatness and uniformity of the thermoclines across cross sections of the TSU. Figures 6-32 shows the thermocline shape across the tank at 20-ft elevation, as measured by series of 12 thermocouples spaced from one wall to the opposite wall along a diameter. Zero on the radial distance (horizontal) scale represents the tank centerline, and the two extreme points are at the two walls of the tank. Each curve represents the temperature profile at a particular time, as keyed in the table at the top of the figure. At the first four time slices, the temperature is at the top operating temperature (575°F). The profile is flat across the tank diameter, with slight curvature near the two walls, corresponding to heat losses through the walls (a phenomenon which has progressively less impact as the tank diameter increases for larger systems). Curve 5 (Figure 6-32) shows the temperature profile as the thermocline is passing through the 20-ft elevation. The small upward curvature near the walls corresponds to heat being transferred from the temporarily hotter walls to the bed. The last four time slices (Curves 6 through 9) show the temperature at the lower operating temperature (425°F) after passage of the thermoclines. The temperature profiles can be seen to be uniform across the tank, showing absence of channelling, "rat-holing", or other non-uniformities in fluid flow and heat transfer.

Figure 6-33 and 6-34 show similar results for the flatness and uniformity of the temperature profiles across the bed cross section, in this case at 32 and 40-ft elevation. At these elevations there are circular rakes of spaced over a full 360 deg at 3.5-ft radius. As with Figure 6-32, each curve represents a time slice. It can be seen that the temperature profiles are extremely uniform, even when the thermocline moves past these elevations.

During long hold periods, the thermocline stays relatively intact and well-defined. Curve 1, Figure 6-38, is a thermocline after a 34-hr hold where the maximum temperature has degraded from 306°C (575°F) to approximately 232°C (540°F). During charging, Run 40, a secondary thermocline was created between the old and new upper temperatures. During the 10-hr charging period the old thermocline was completely displaced by the new thermocline. During this test, charging rate was varied from 1.1 MW to 0.1 MW at 12 noon for a period of 3 hr. Within experimental error no difference in thermocline shape can be detected from the two widely varying flow rates.

Thermocline characteristics are apparently independent of operating mode. During test 47, Figure 6-45, four periods of alternating charges and extractions resulted in imperceptible changes in thermocline shape over a 9-hr period except the small amount of degradation that normally occurs with time.

Figures not discussed (35-37, 39-44, 46-48, 50, and 53) provide further data on charge, extract, and holds for various rates and durations.

6.3.4.2 Heat Loss

Temperatures were recorded at the top, foundation, and side walls to facilitate heat flow measurements. In addition, a portable thermopile type heat flux gage was used periodically on the outer surfaces to establish values of heat flow.

1	DAY 329	H.M.S 10.34.50	2	DAY 329	H.M.S 11.36.00	3	DAY 329	H.M.S 12.42.04
4	DAY 329	H.M.S 14.02.01	5	DAY 329	H.M.S 14.54.47	6	DAY 329	H.M.S 16.11.42
7	DAY 329	H.M.S 17.11.30	8	DAY 329	H.M.S 18.08.25	9	DAY 329	H.M.S 19.14.50

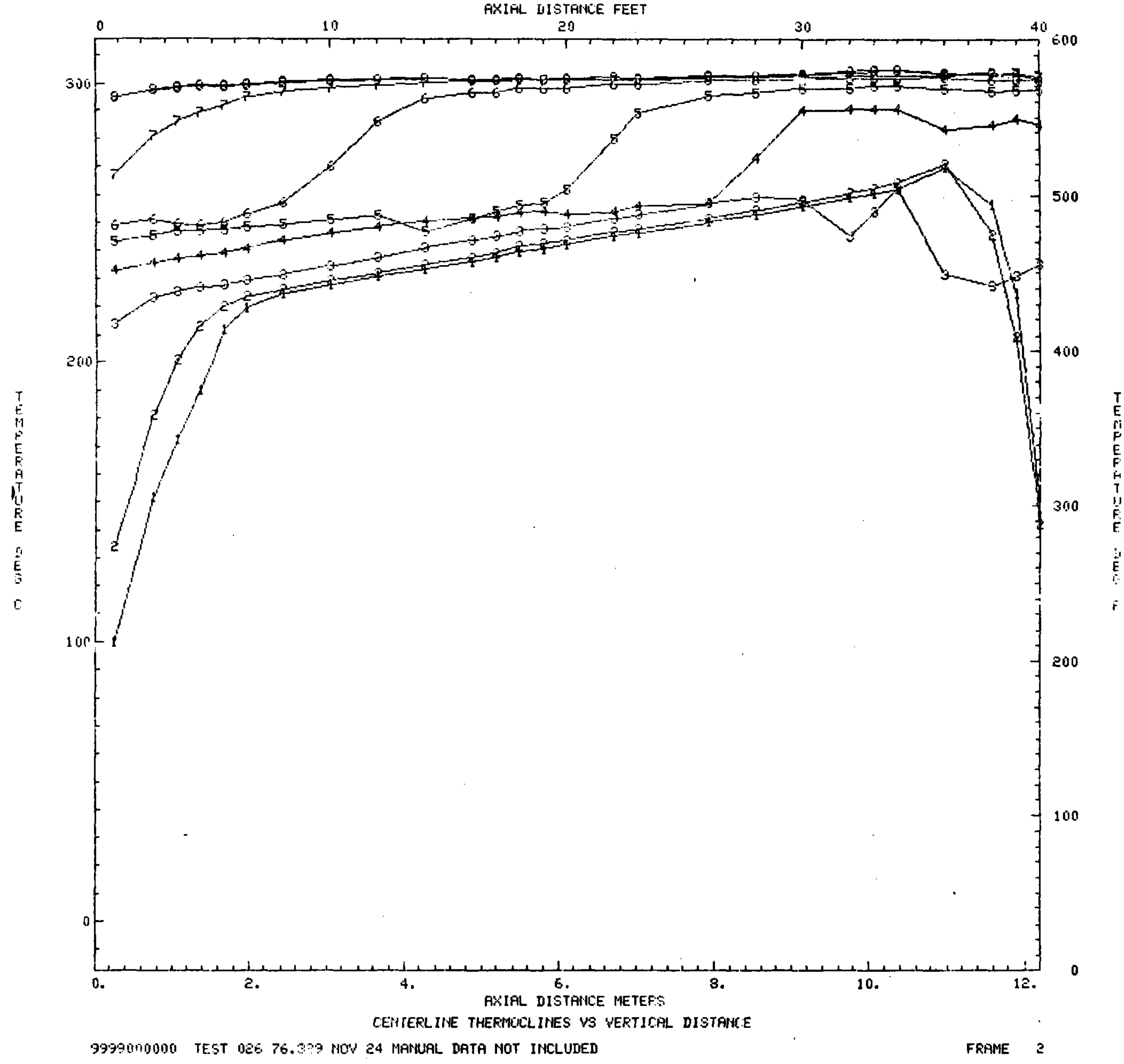
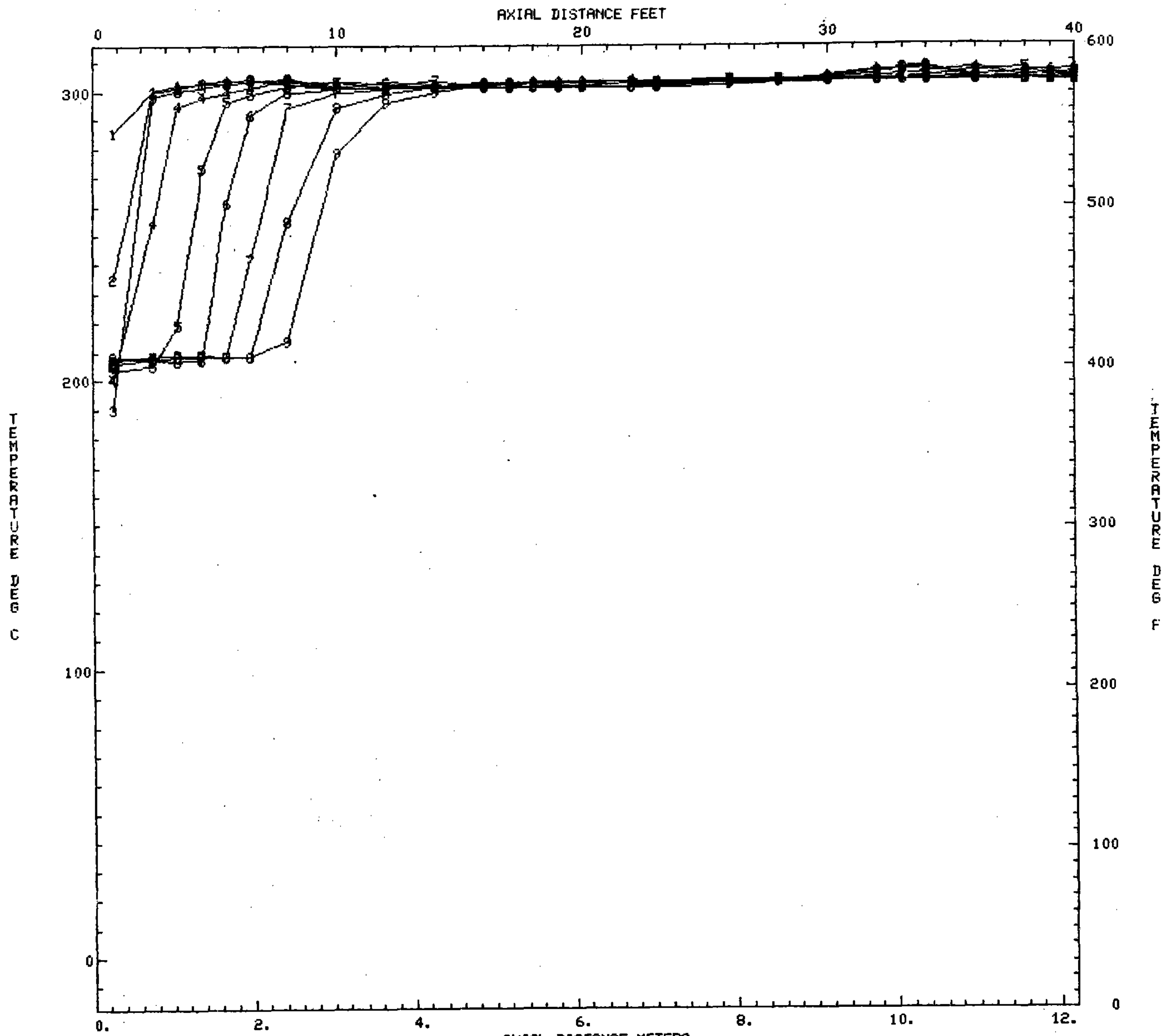


Figure 6-28. Test 26: Charging, Centerline Thermoclines vs Distance

1	DAY 335	H.M.S	19.15.10	2	DAY 335	H.M.S	19.21.03	3	DAY 335	H.M.S	19.30.10
4	DAY 335	H.M.S	19.56.10	5	DAY 335	H.M.S	20.22.10	6	DAY 335	H.M.S	20.49.49
7	DAY 335	H.M.S	21.16.10	8	DAY 335	H.M.S	21.45.10	9	DAY 335	H.M.S	22.09.39



CENTERLINE THERMOCLINES VS VERTICAL DISTANCE

9999000000 TEST 029 76.335 NOV 30 EXTRACTION MANUAL DATA INCLUDED FRAME 2

Figure 6-29. Test 29: Extraction, Centerline Thermoclines vs Distance

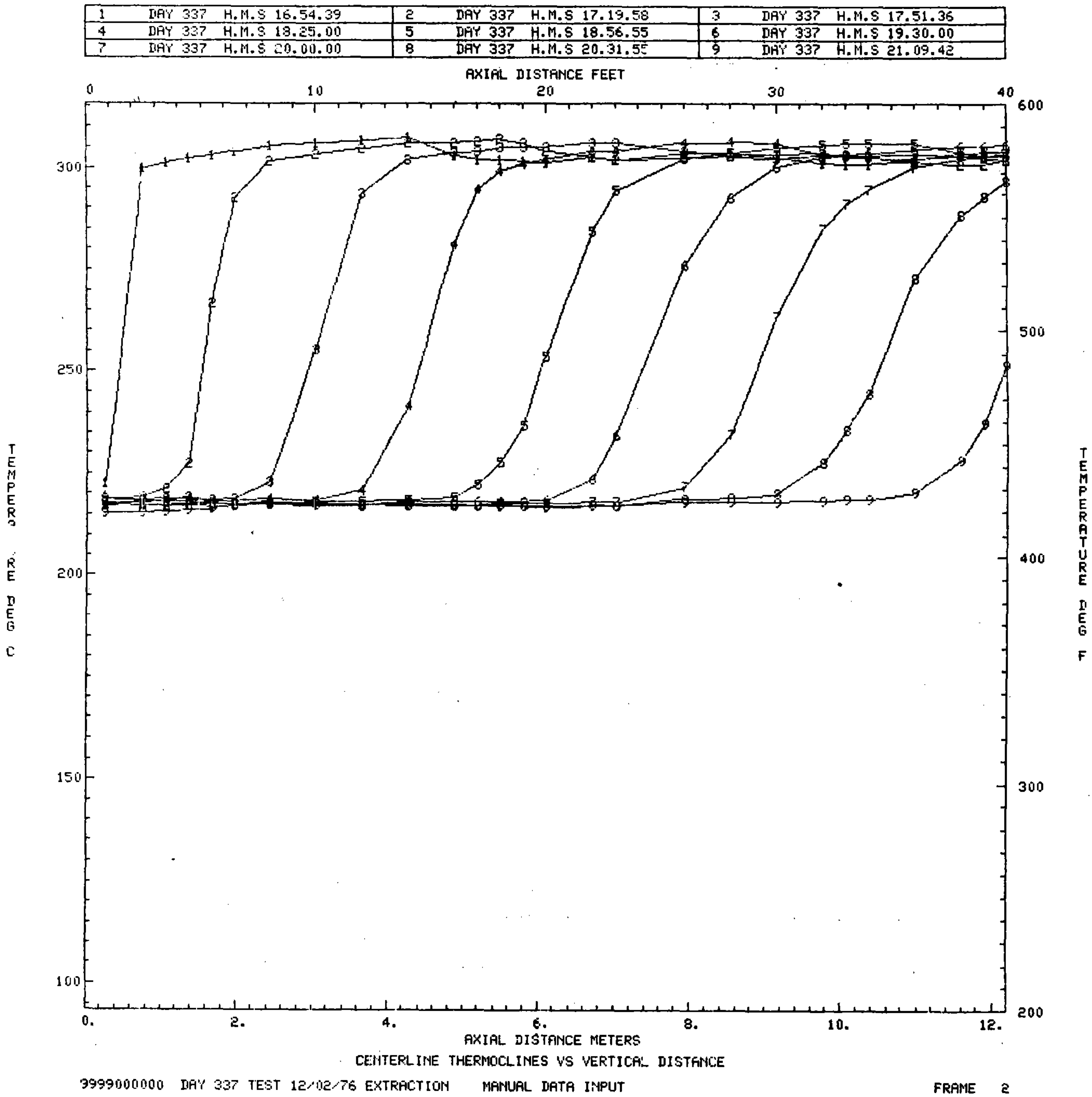


Figure 6-30. Test 32: Extraction, Centerline Thermoclines vs Distance

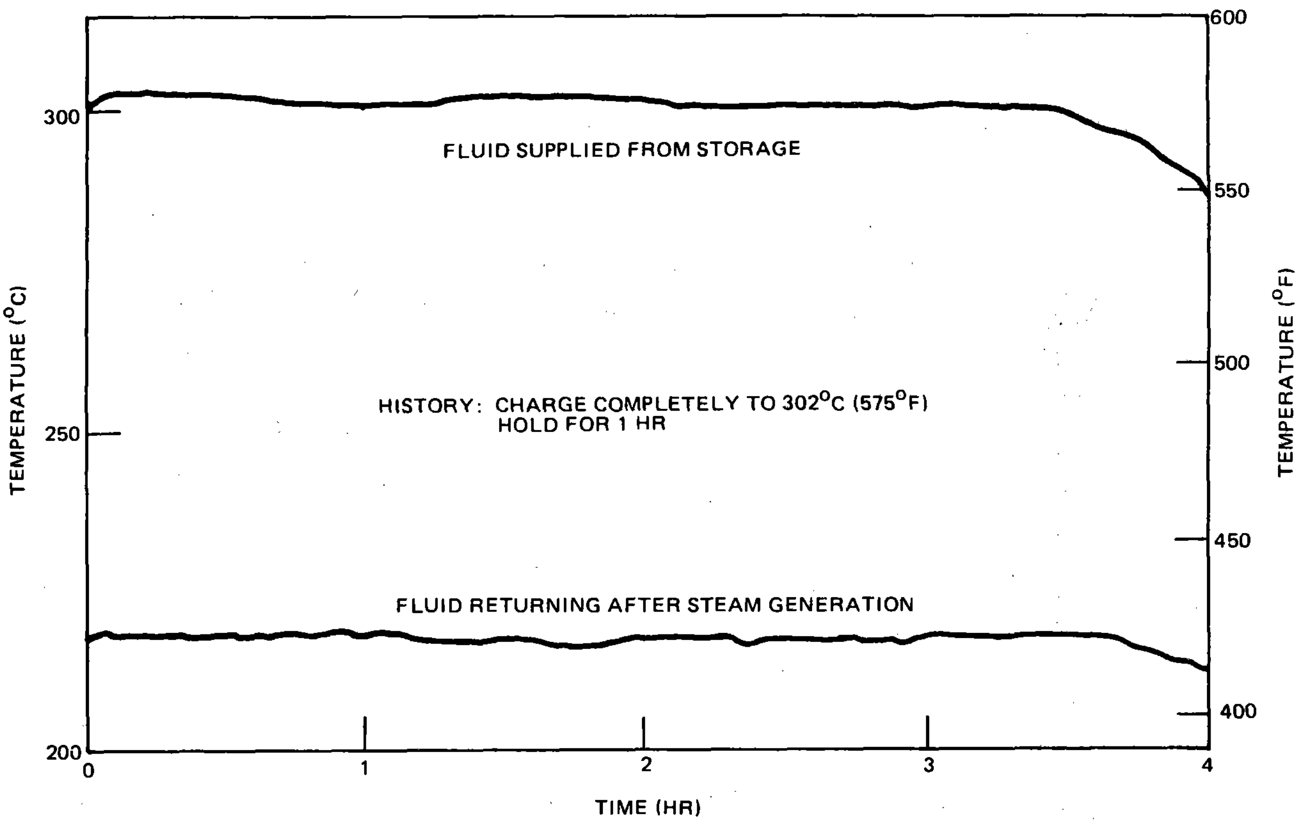
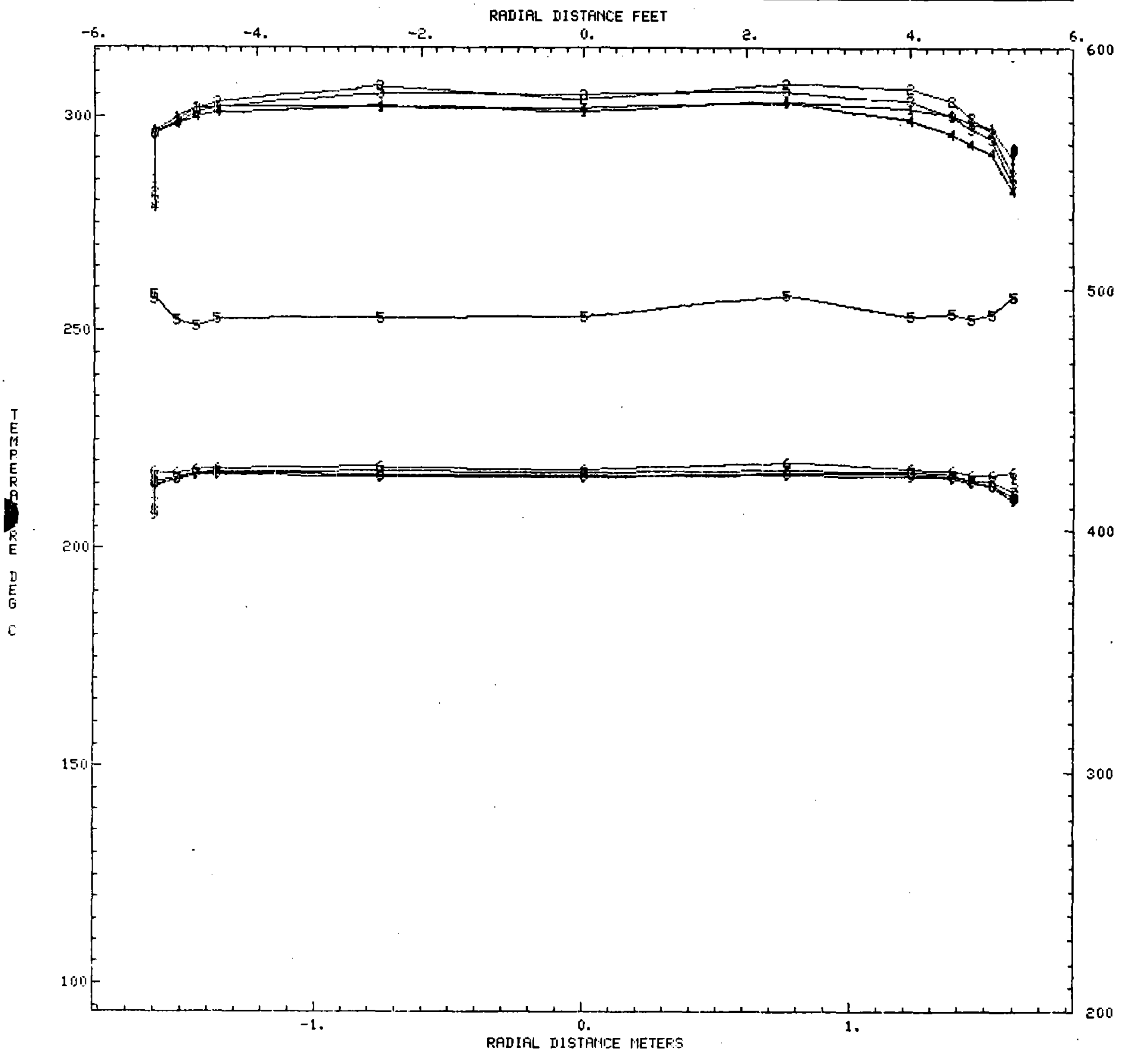


Figure 6-31. Typical Performance Data During Extraction

1	DAY 337	H.M.S 16.54.39	2	DAY 337	H.M.S 17.19.58	3	DAY 337	H.M.S 17.51.36
4	DAY 337	H.M.S 18.25.00	5	DAY 337	H.M.S 18.56.55	6	DAY 337	H.M.S 19.30.00
7	DAY 337	H.M.S 20.00.00	8	DAY 337	H.M.S 20.31.55	9	DAY 337	H.M.S 21.09.42



9999000000 DAY 337 TEST 12/02/76 EXTRACTION MANUAL DATA INPUT FRAME 4

Figure 6-32. Test 32: Extraction, 20-ft Level Radial Temperature Profiles

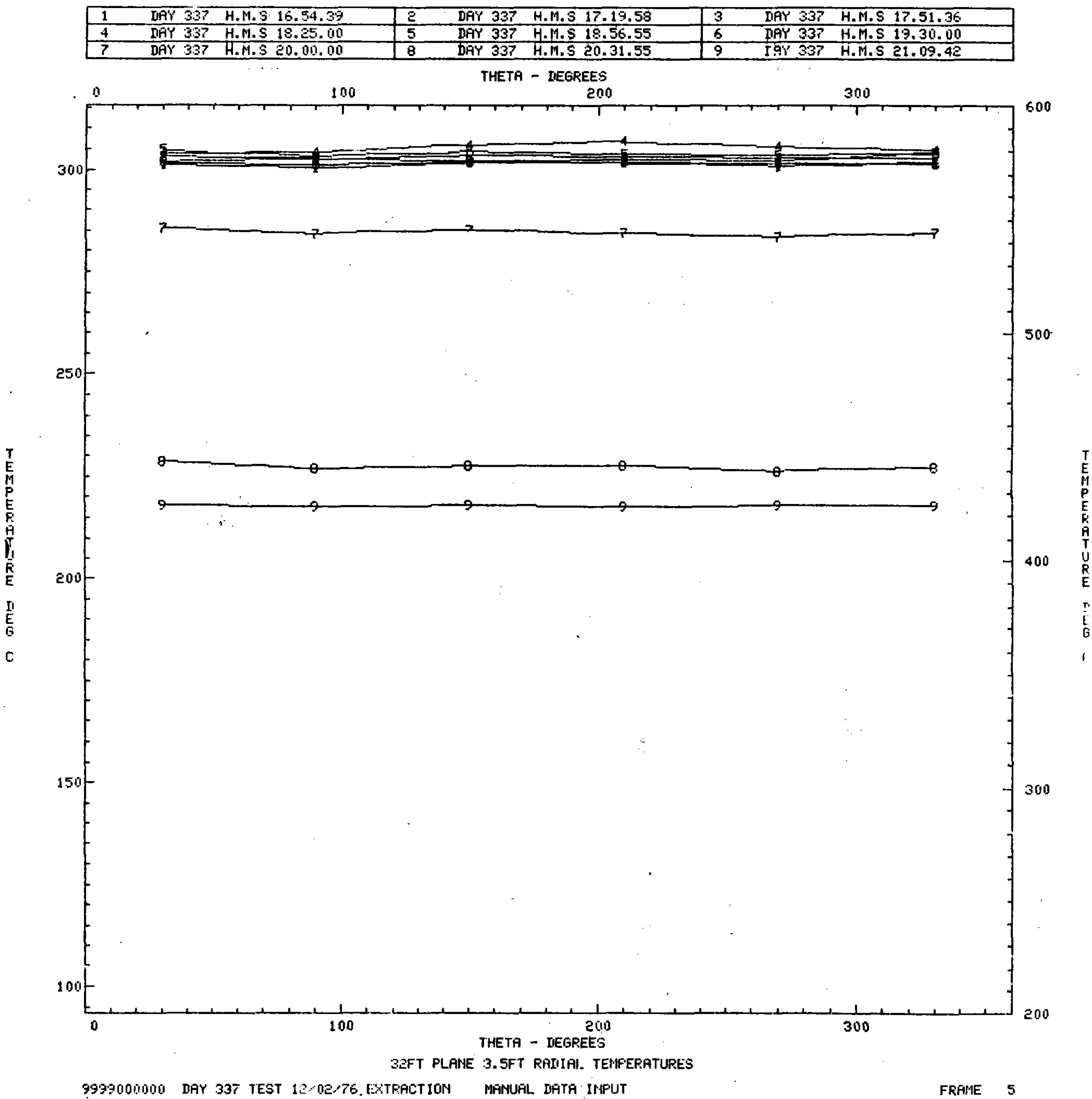


Figure 6-33. Test 32: Extraction, 32-ft Level Circumferential Temperature Profiles

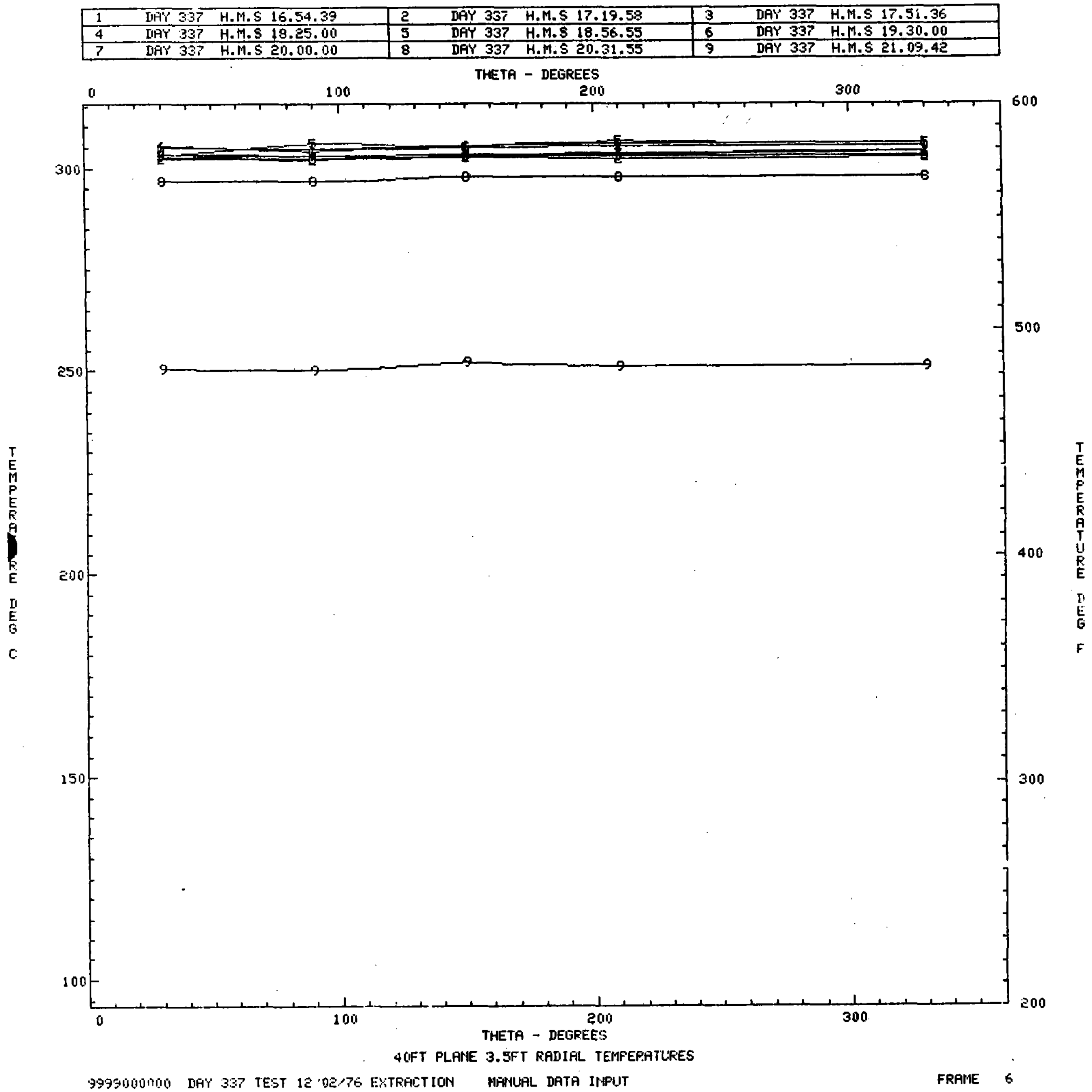


Figure 6-34. Test 32: Extraction, 40-ft Level Circumferential Temperature Profiles

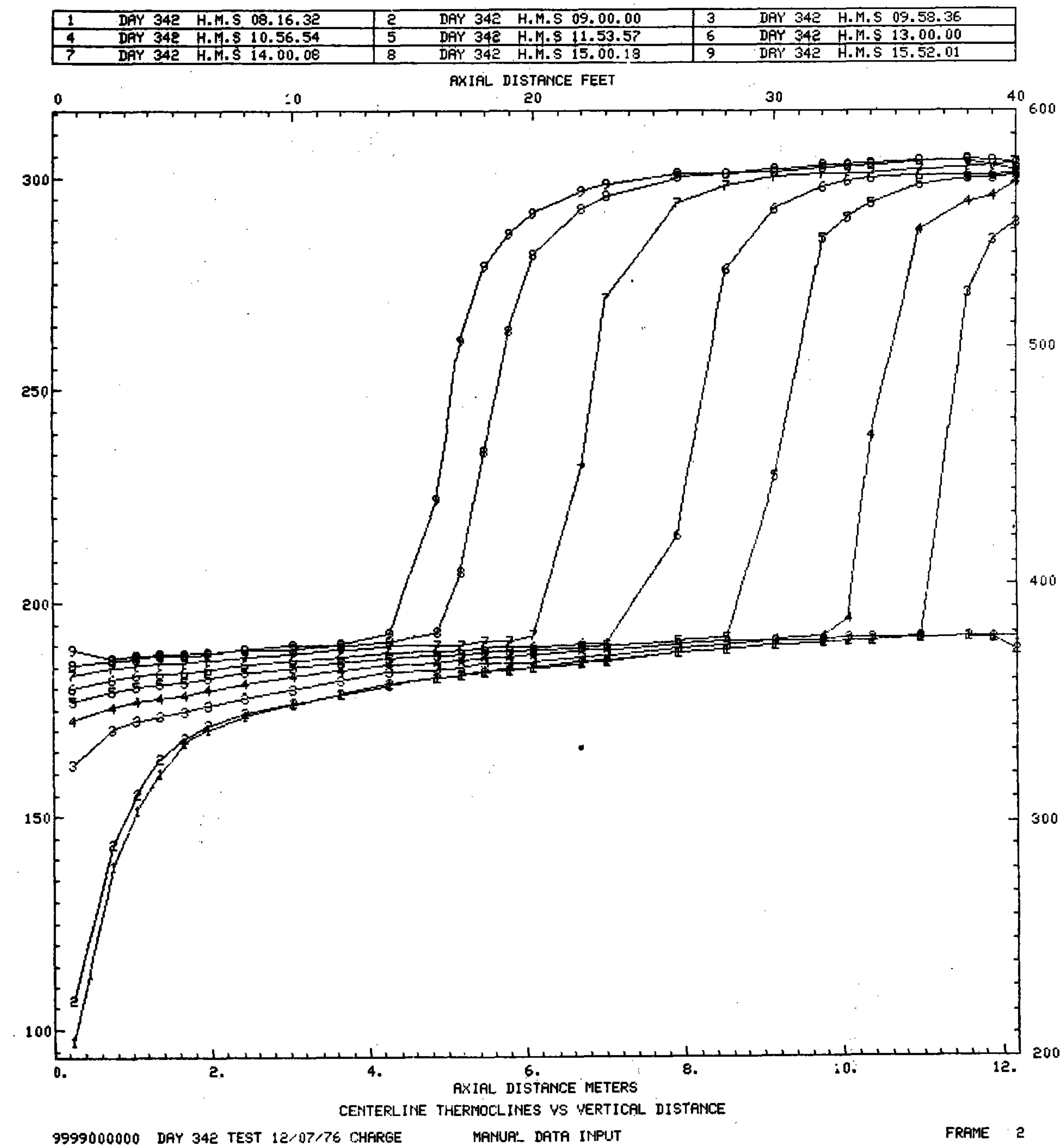


Figure 6-35. Test 35: Charge, Centerline Thermoclines vs Distance

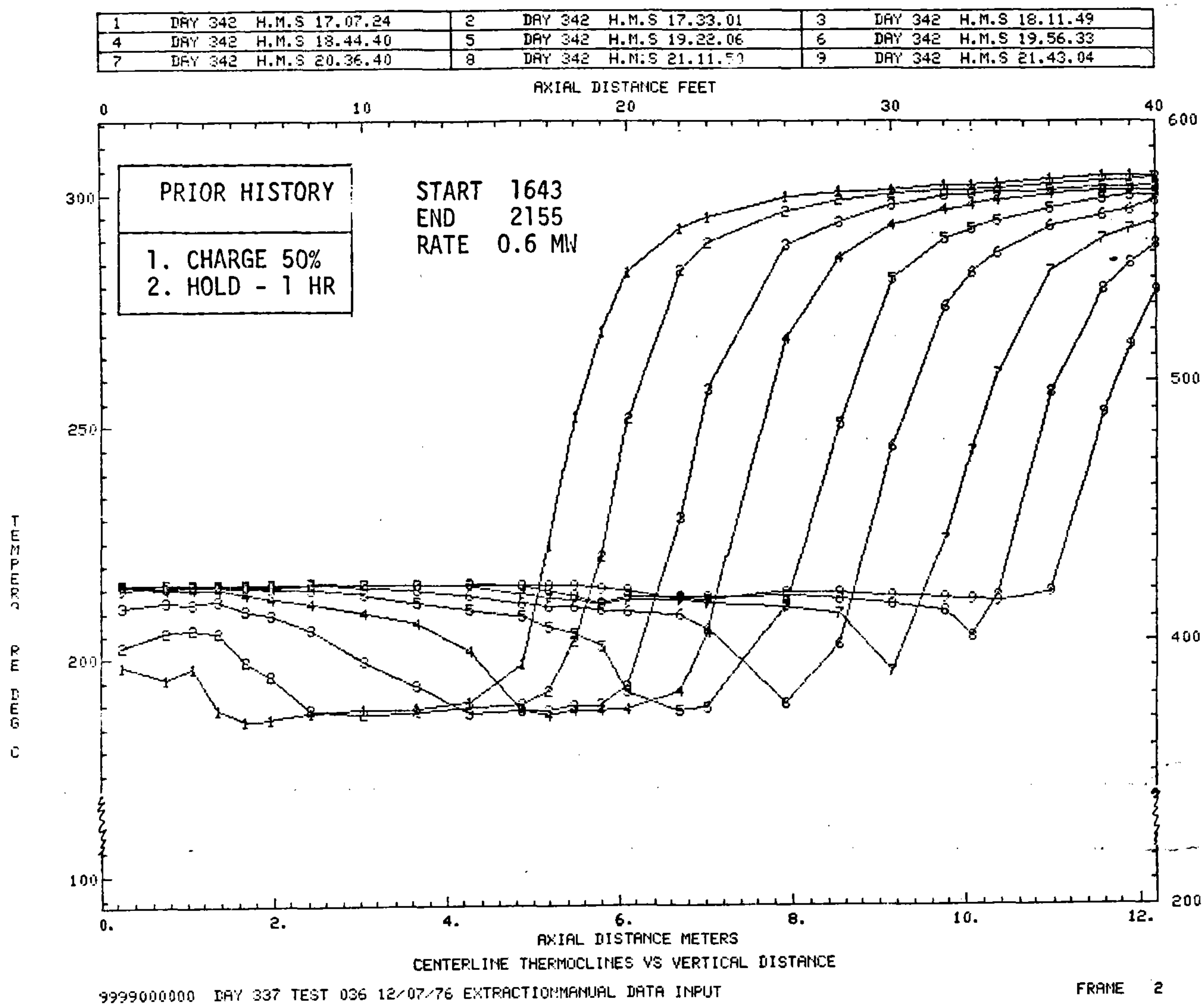


Figure 6-36. Test 36: Extraction, Centerline Thermoclines vs Distance

1 DAY 343 H.M.S 13.51.16	2 DAY 343 H.M.S 13.57.44	3 DAY 343 H.M.S 14.40.30
4 DAY 343 H.M.S 15.30.00	5 DAY 343 H.M.S 16.12.59	6 DAY 343 H.M.S 16.56.15
7 DAY 343 H.M.S 17.45.30	8 DAY 343 H.M.S 18.30.00	9 DAY 343 H.M.S 19.14.48

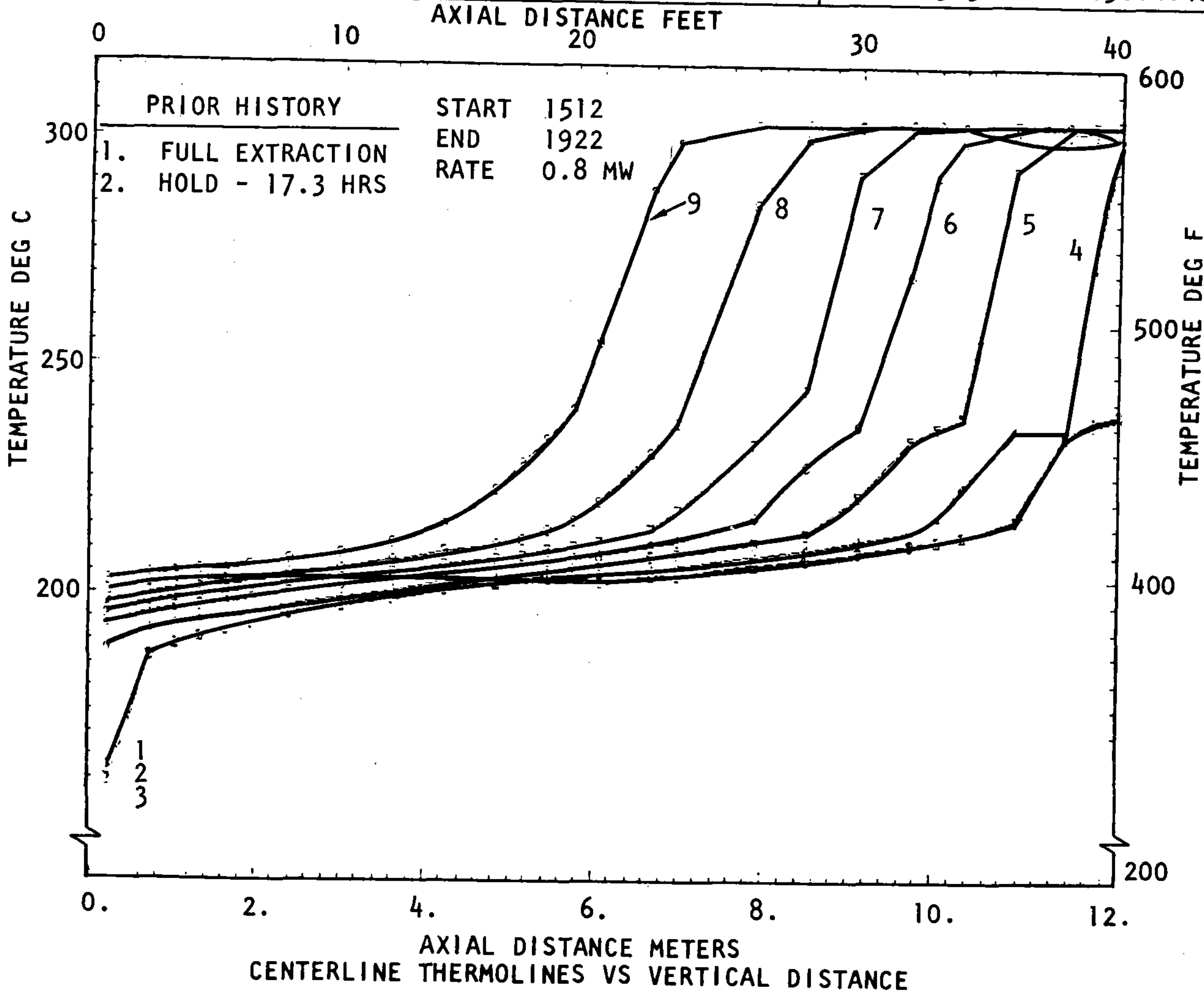
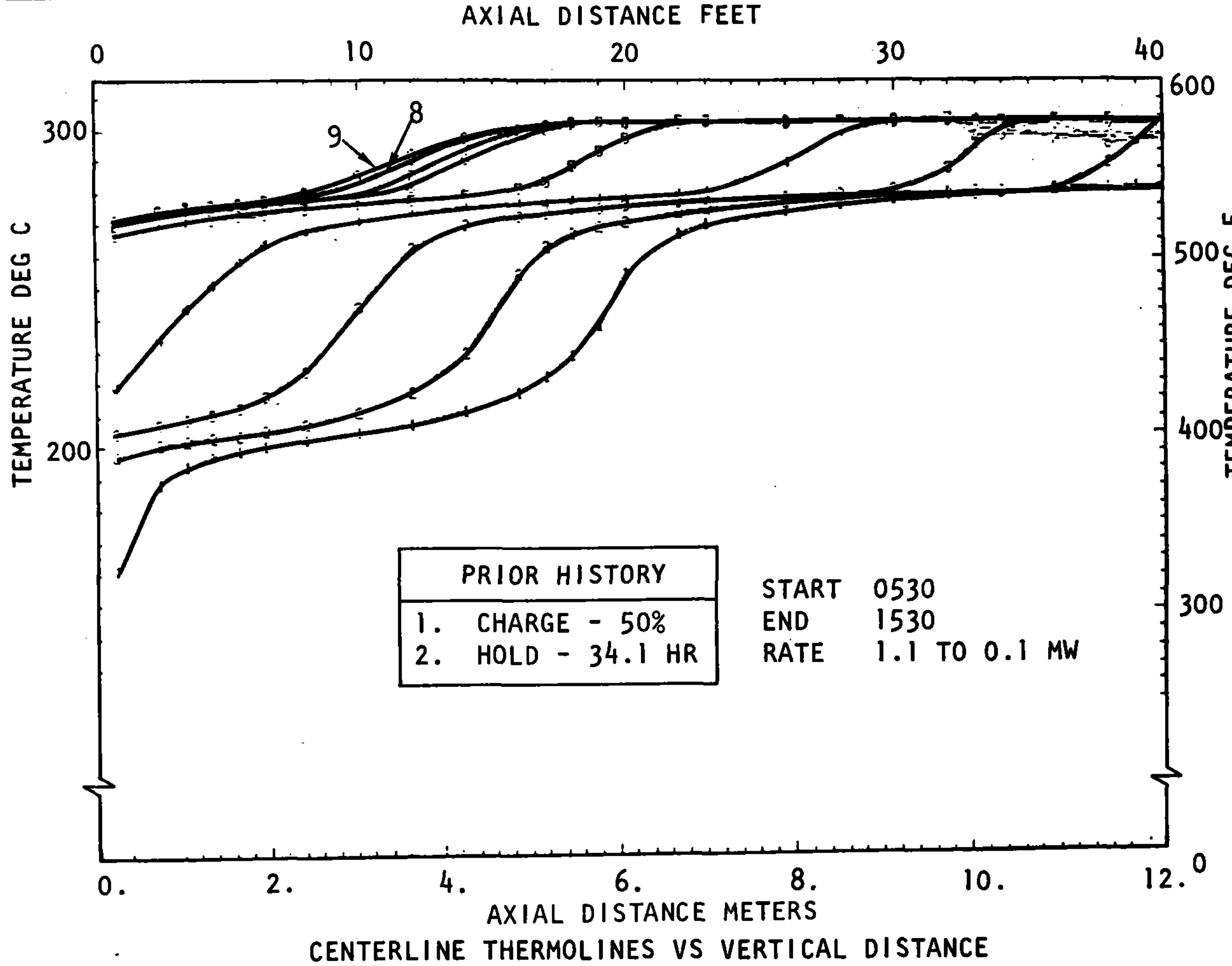


Figure 6-37. Test 38: Charge, Centerline Thermoclines vs Distance

1 DAY 345 H.M.S 06.11.08	2 DAY 345 H.M.S 07.16.49	3 DAY 345 H.M.S 08.24.56
4 DAY 345 H.M.S 09.36.13	5 DAY 345 H.M.S 10.38.06	6 DAY 345 H.M.S 12.09.32
7 DAY 345 H.M.S 12.55.07	8 DAY 345 H.M.S 13.55.40	9 DAY 345 H.M.S 15.03.40



TEST 040 DEC 10 CHARGE

Figure 6-38. Test 40, Charge, Centerline Thermoclines Vs Distance

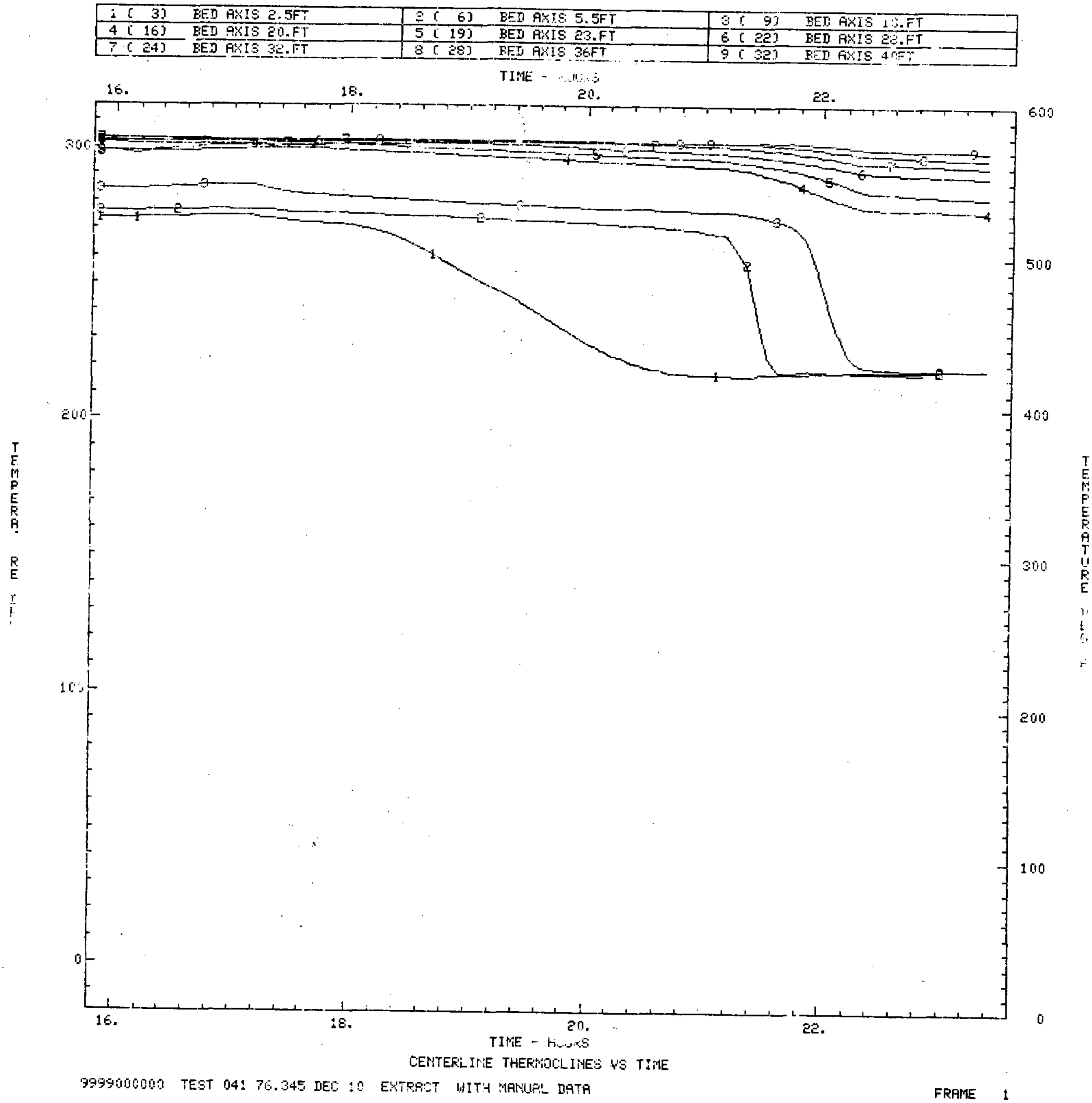


Figure 6-39. Test 41: Extract, Centerline Thermocline vs Time

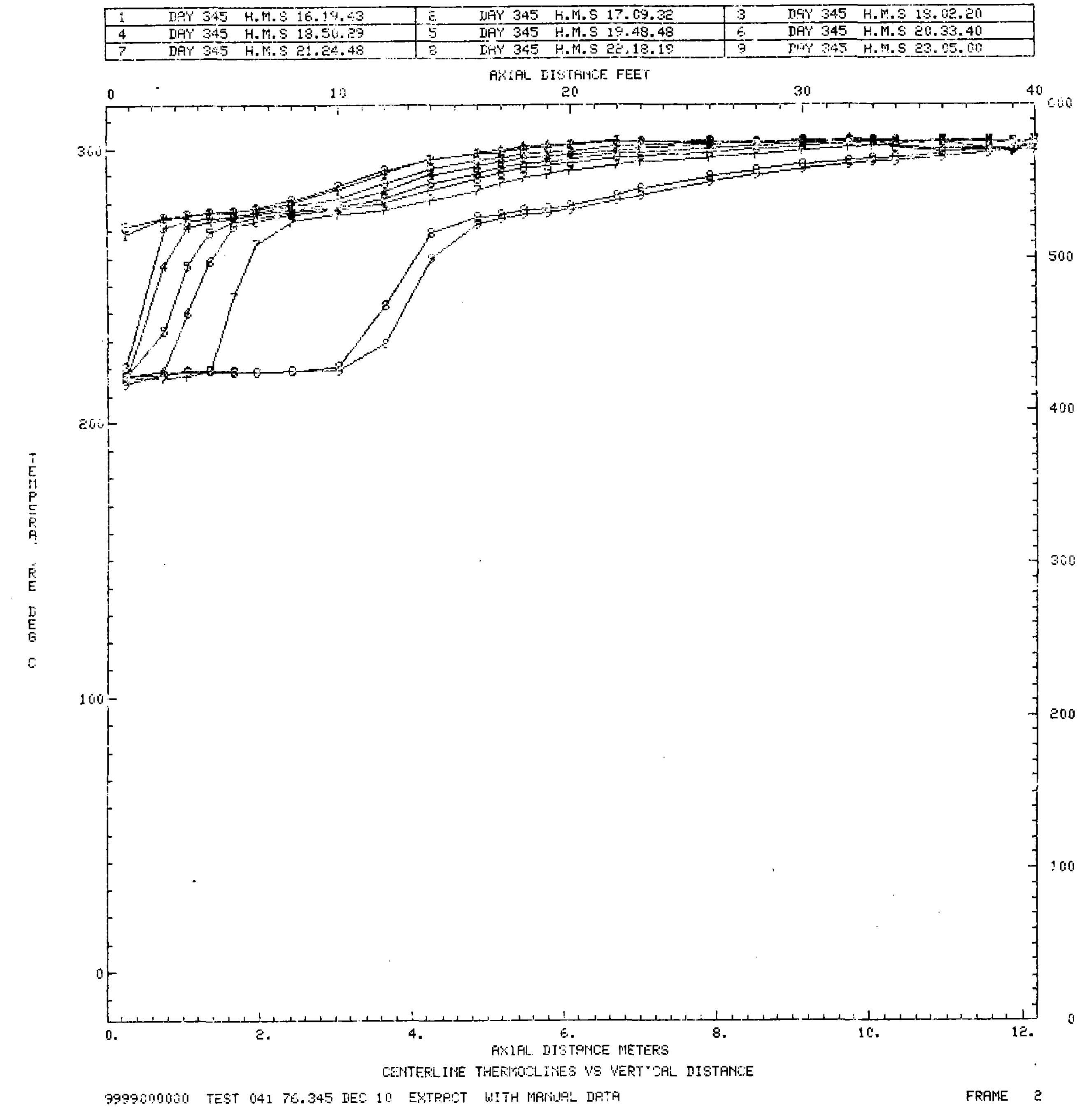


Figure 6-40. Test 41: Extract, Centerline Thermoclines vs Distance

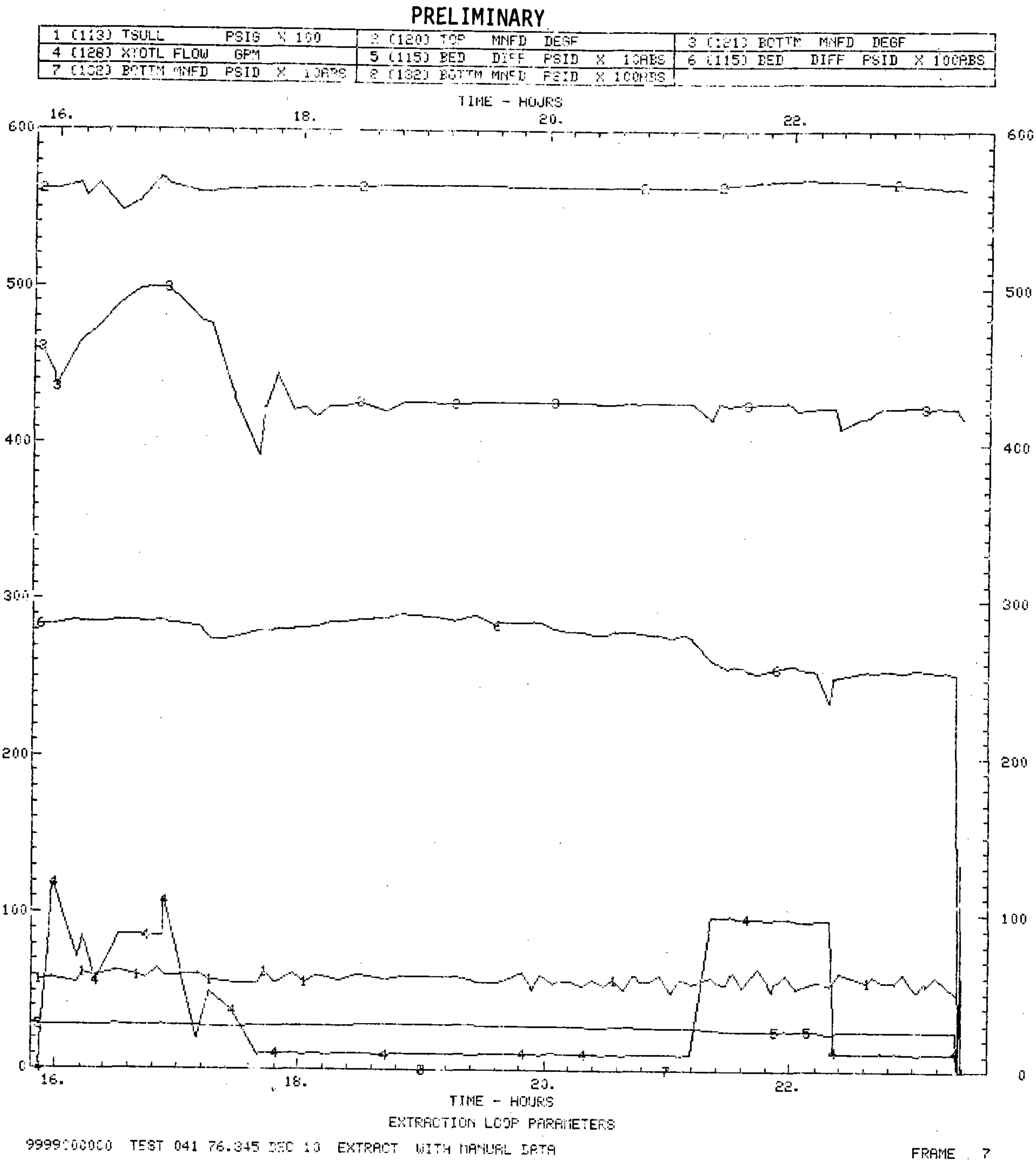


Figure 6-41. Test 41: Extract, Extraction Parameters

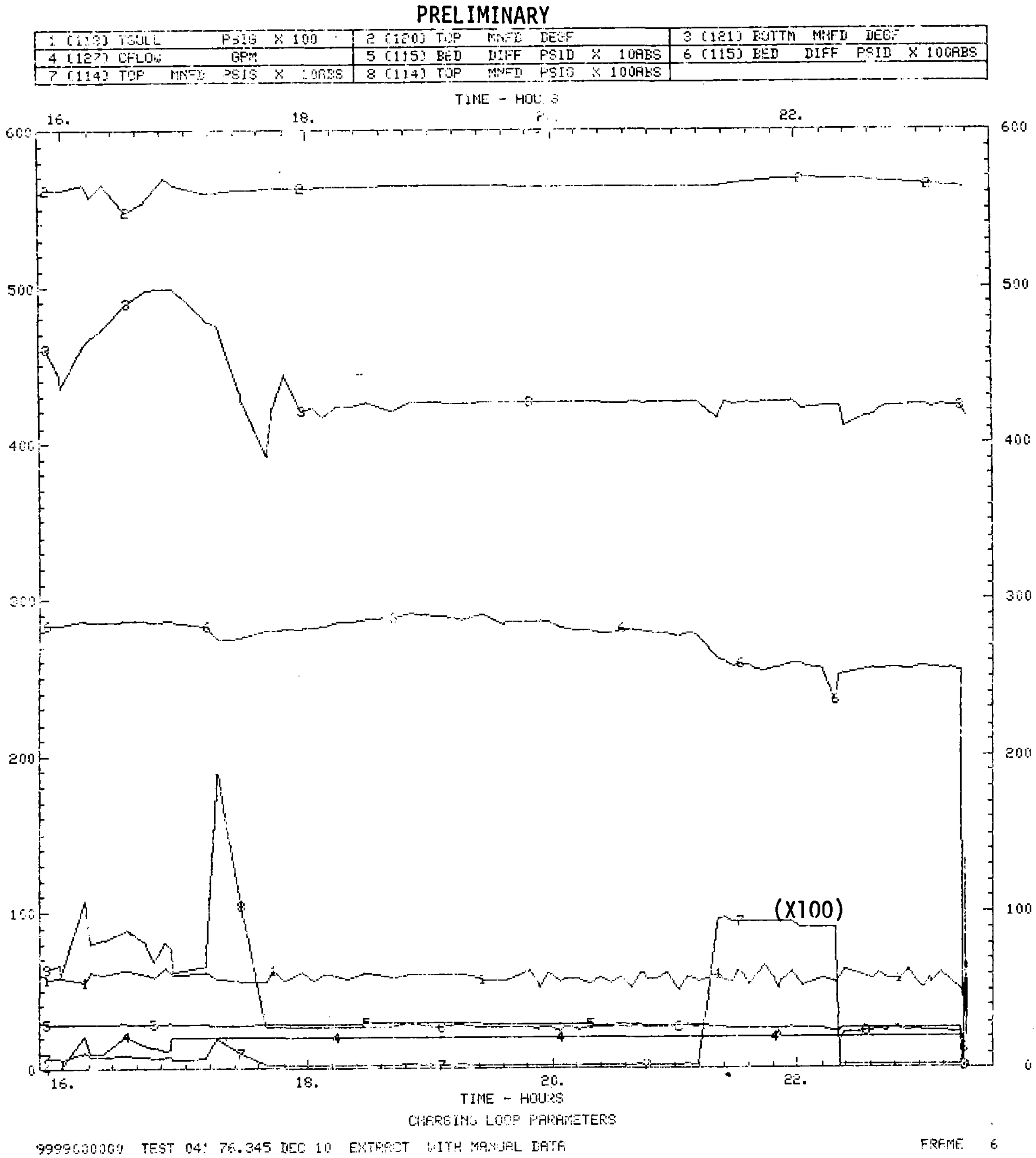


Figure 6-42. Test 41: Extract, Charging Parameters

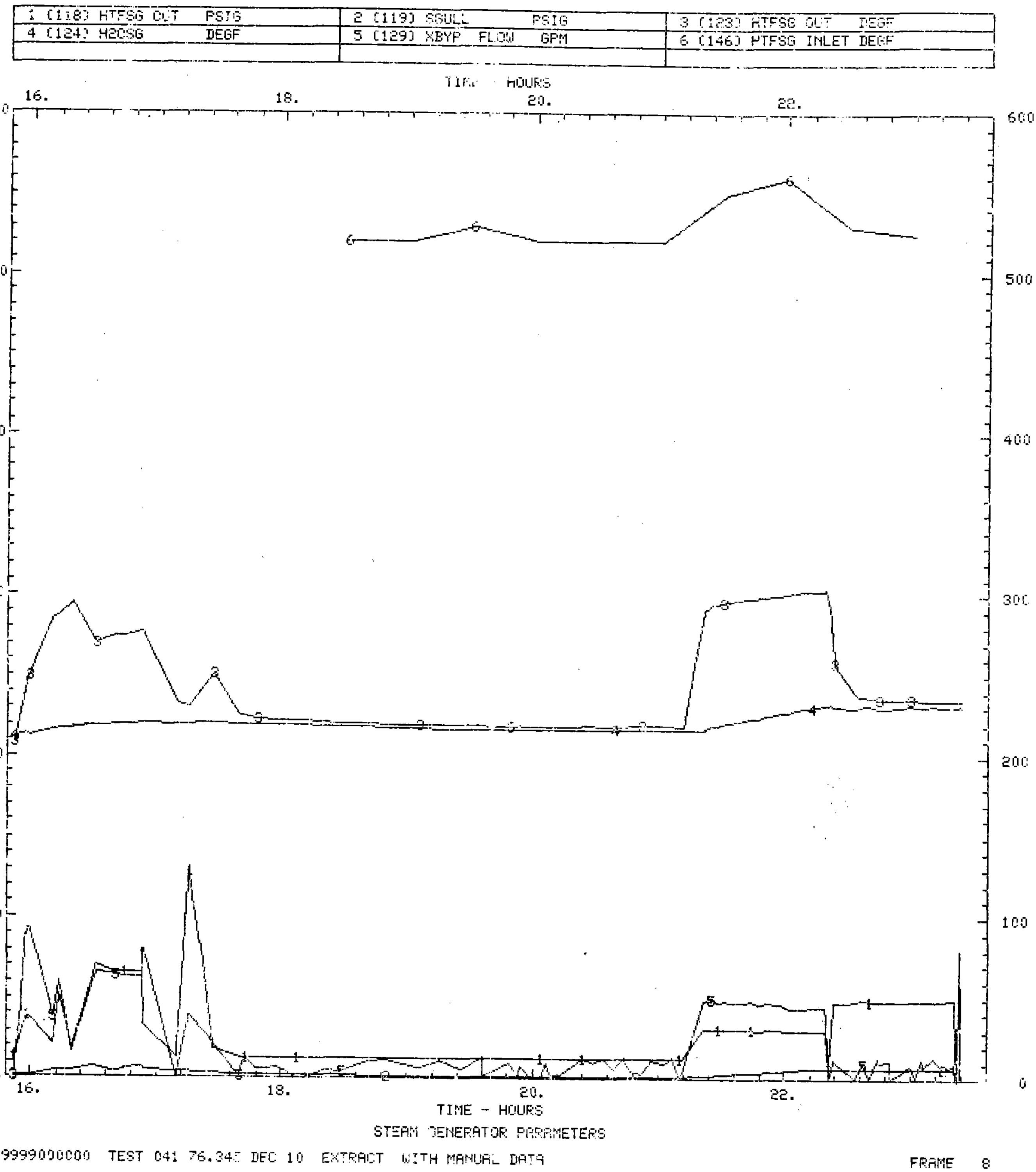
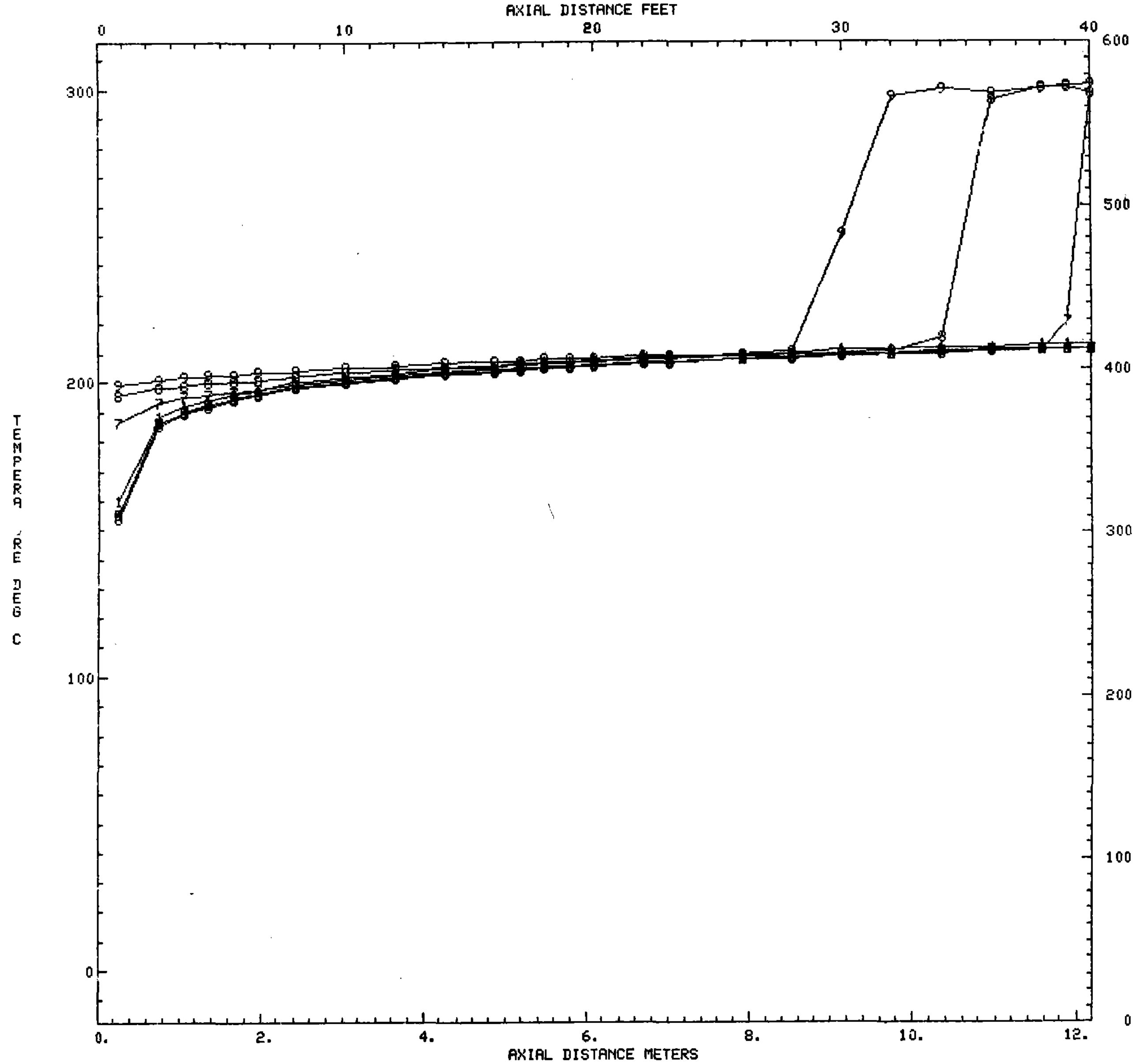


Figure 6-43. Test 41: Extract, Steam Generator Parameters.

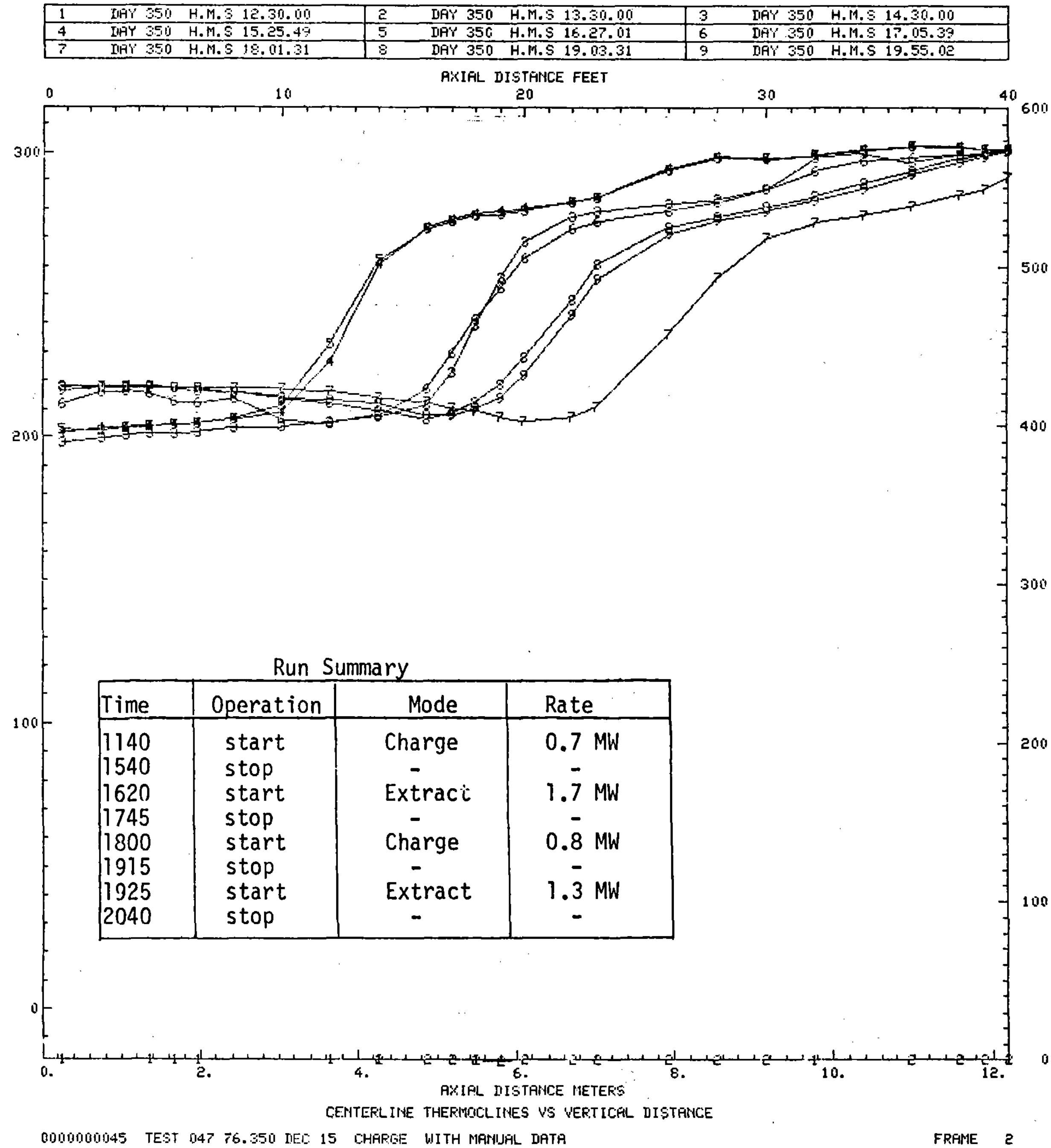
1	DAY 349	H.M.S 10.57.08	2	DAY 349	H.M.S 14.03.46	3	DAY 349	H.M.S 14.04.29
4	DAY 349	H.M.S 14.05.17	5	DAY 349	H.M.S 14.06.09	6	DAY 349	H.M.S 15.03.41
7	DAY 349	H.M.S 16.00.00	8	DAY 349	H.M.S 16.54.10	9	DAY 349	H.M.S 17.54.52



CENTERLINE THERMOCLINES VS VERTICAL DISTANCE

0000000045 TEST 045 76.349 DEC 14 CHARGE WITH MANUAL DATA FRAME 2

Figure 6-44. Test 45: Charge, Centerline Thermoclines vs Distance



1	DAY 350	H.M.S 12.30.00	2	DAY 350	H.M.S 13.30.00	3	DAY 350	H.M.S 14.30.00
4	DAY 350	H.M.S 15.25.49	5	DAY 350	H.M.S 16.27.01	6	DAY 350	H.M.S 17.05.39
7	DAY 350	H.M.S 18.01.31	8	DAY 350	H.M.S 19.03.31	9	DAY 350	H.M.S 19.55.02

Run Summary

Time	Operation	Mode	Rate
1140	start	Charge	0.7 MW
1540	stop	-	-
1620	start	Extract	1.7 MW
1745	stop	-	-
1800	start	Charge	0.8 MW
1915	stop	-	-
1925	start	Extract	1.3 MW
2040	stop	-	-

Figure 6-45. Test 47: Charge/Extract Centerline Thermoclines vs Distance

1	DAY 350	H.M.S 21.58.02		2	DAY 350	H.M.S 22.58.02		3	DAY 350	H.M.S 23.58.02	
4	DAY 351	H.M.S 00.58.02		5	DAY 351	H.M.S 02.58.02		6	DAY 351	H.M.S 03.58.02	
7	DAY 351	H.M.S 04.20.24		8	DAY 351	H.M.S 05.21.44		9	DAY 351	H.M.S 07.01.02	

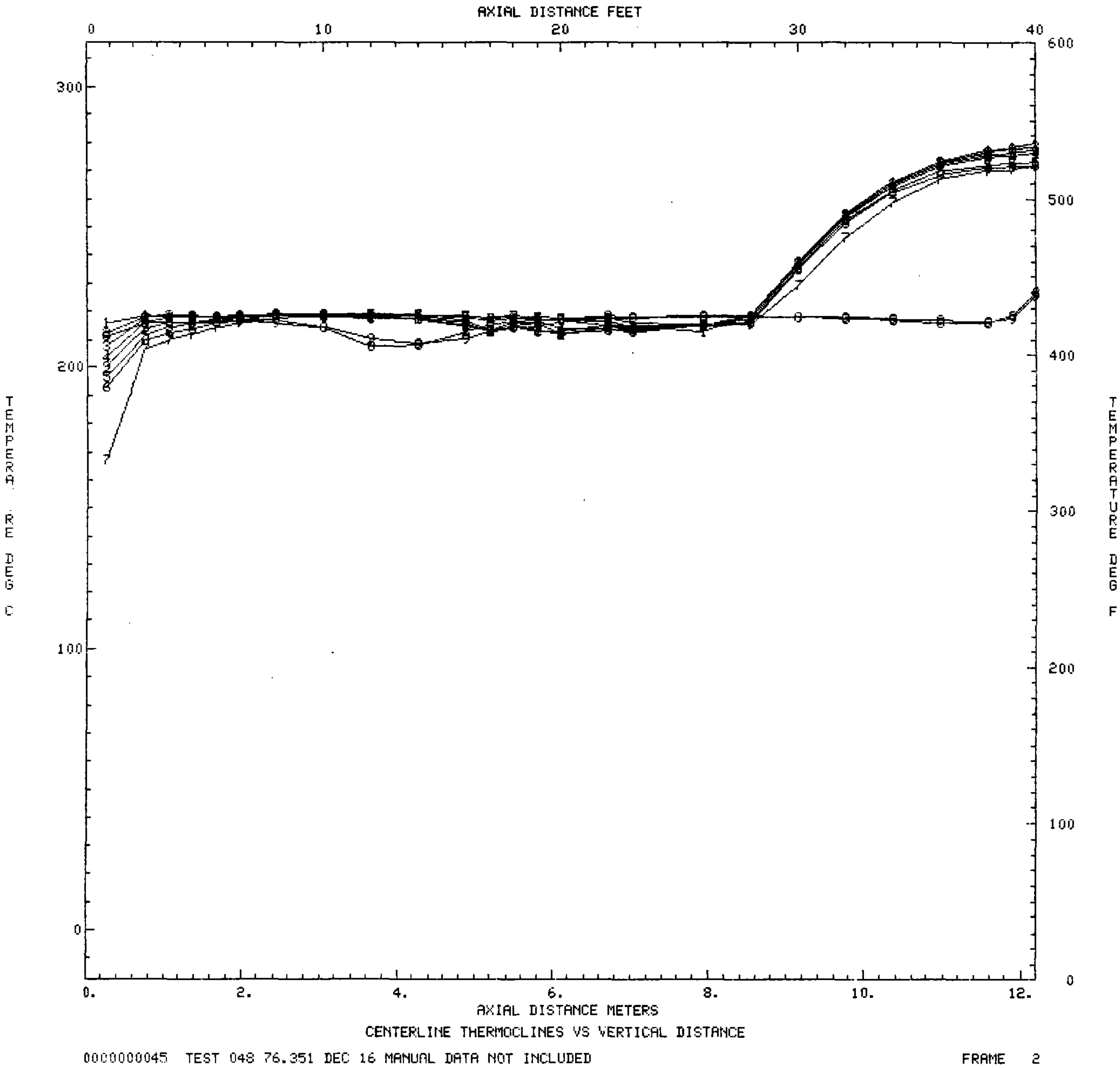
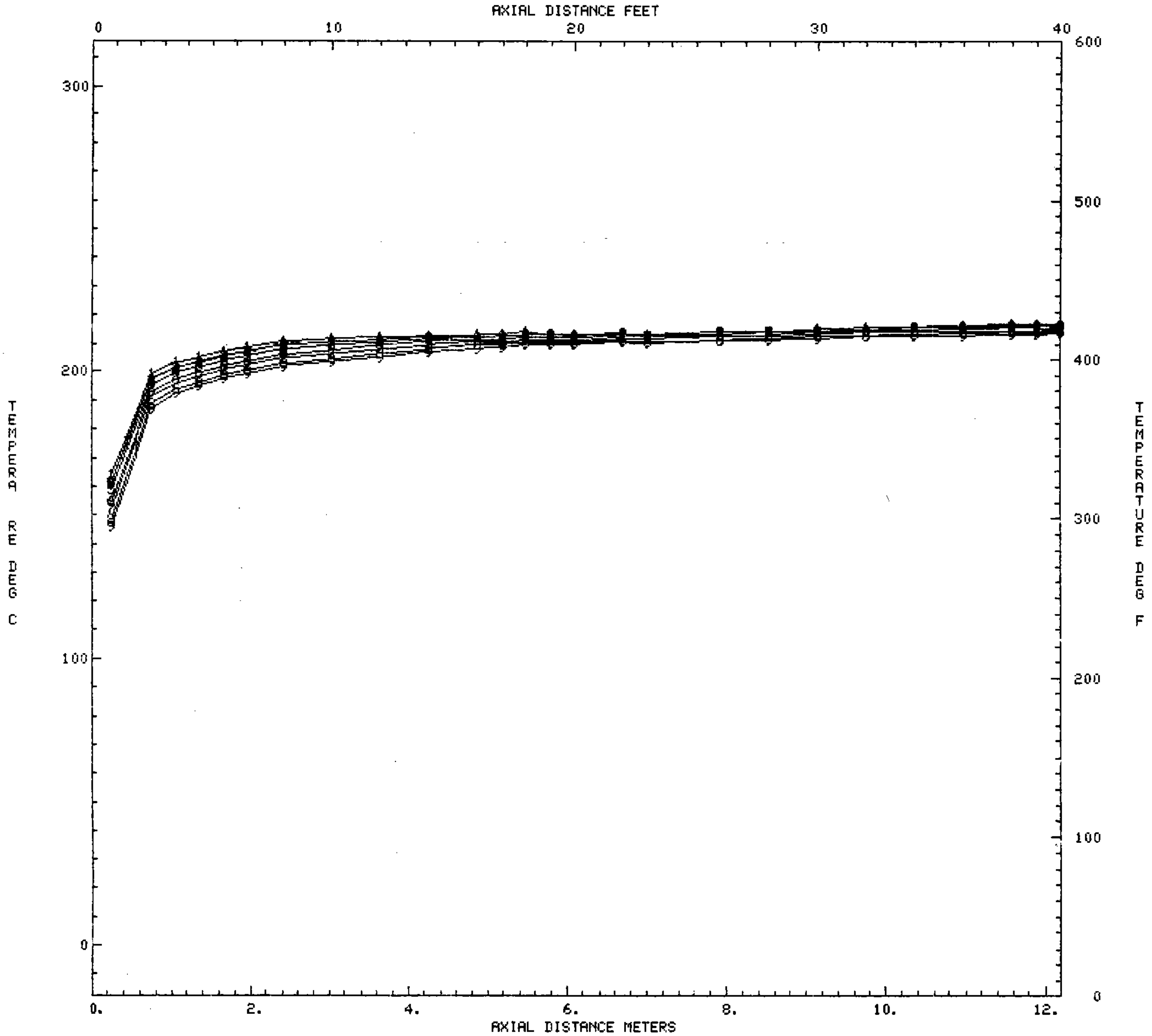


Figure 6-46. Test 48: Hold, Centerline Thermoclines vs Distance

1	DAY 351	H.M.S 15.46.12	2	DAY 351	H.M.S 17.23.27	3	DAY 351	H.M.S 17.25.02
4	DAY 351	H.M.S 17.25.36	5	DAY 351	H.M.S 18.54.03	6	DAY 351	H.M.S 21.25.02
7	DAY 351	H.M.S 23.25.02	8	DAY 352	H.M.S 02.25.02	9	DAY 352	H.M.S 04.25.02



0000000045 TEST 049050 76.352 DEC 17 MANUAL DATA NOT INCLUDED FRAME 2

Figure 6-47. Test 50: Hold, Centerline Thermoclines vs Distance

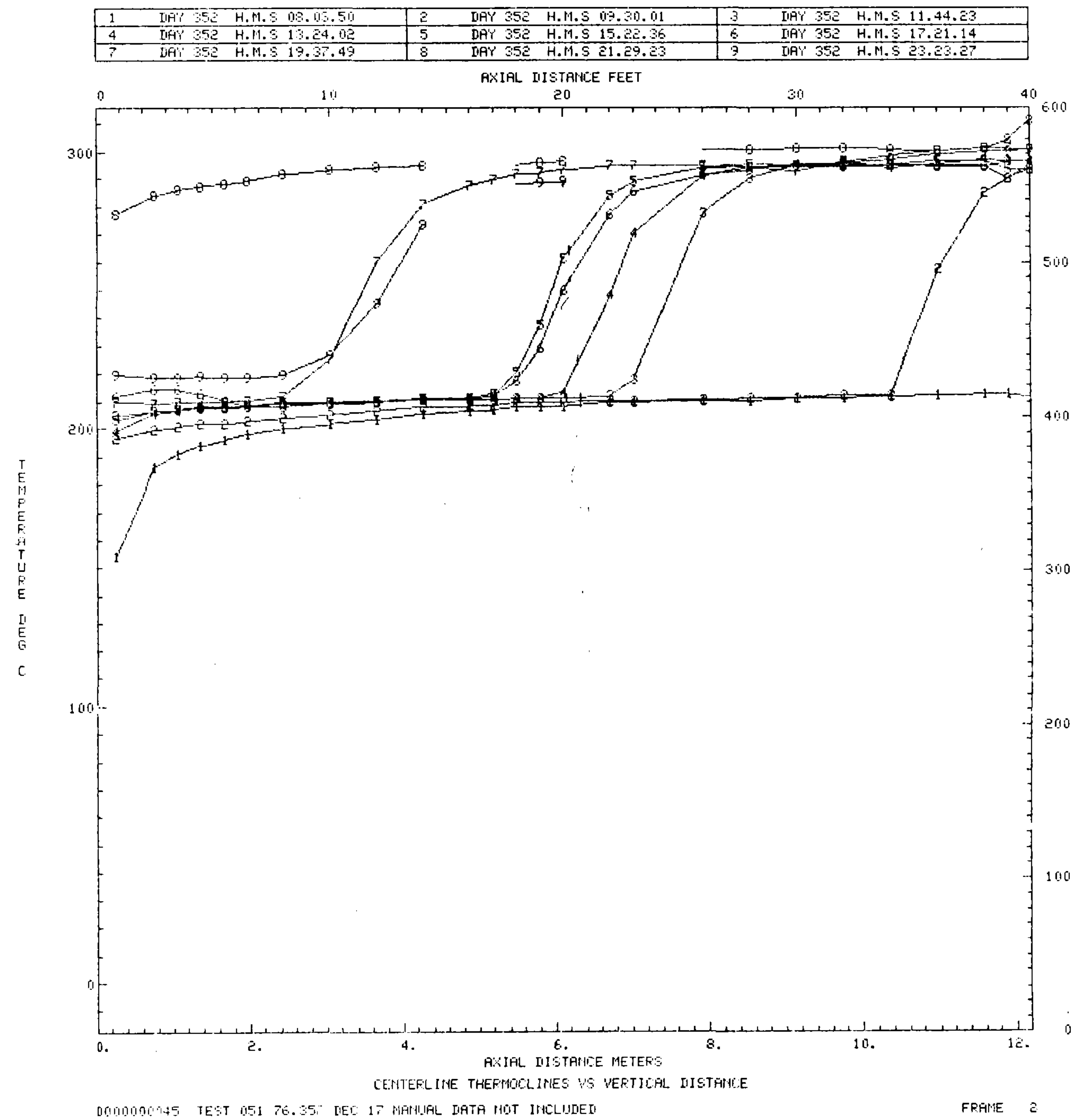
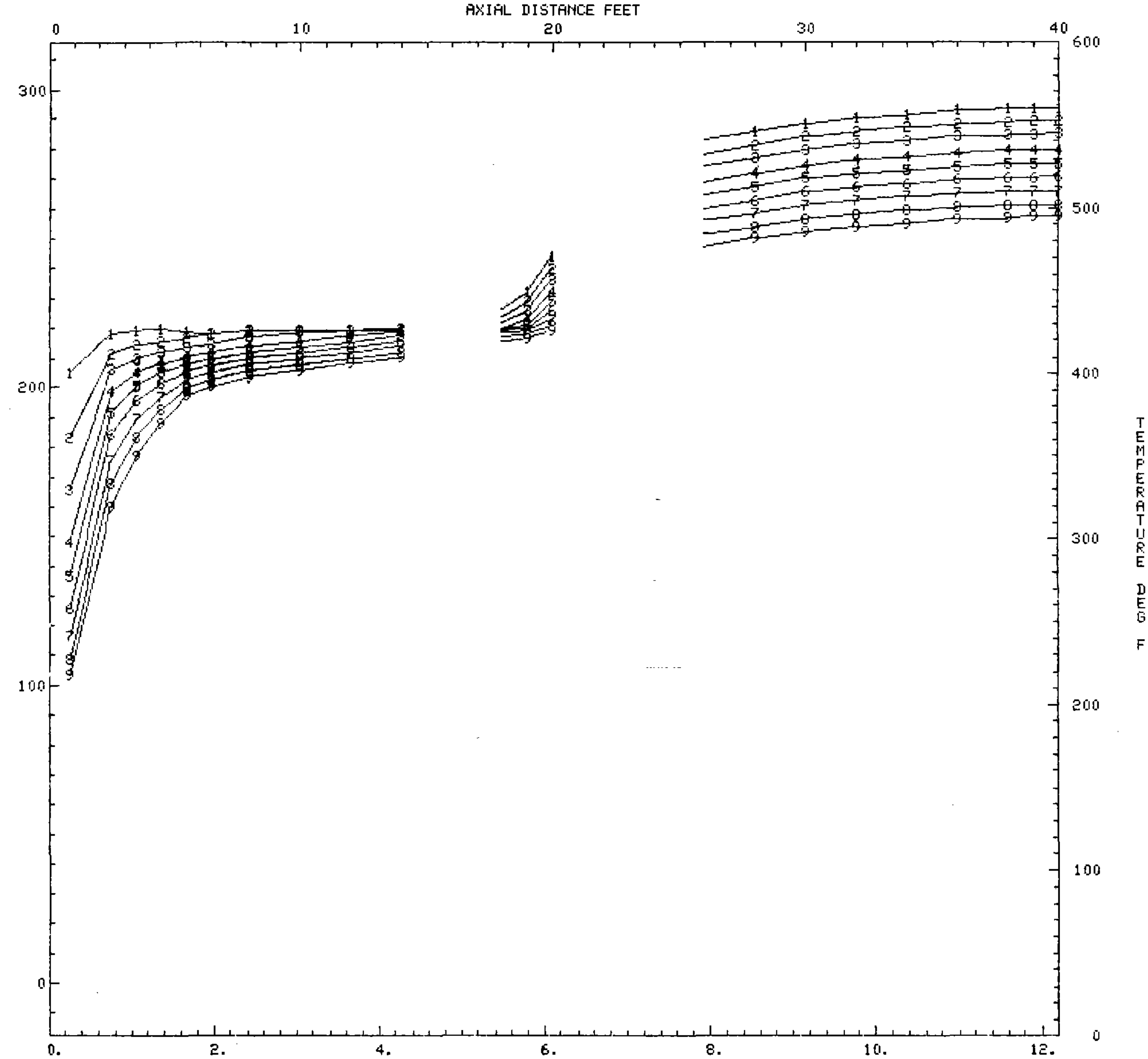


Figure 6-48. Test 51: Duty Cycle, Centerline Thermoclines vs Distance

1	DAY 353	H.M.S 04.28.02	2	DAY 353	H.M.S 10.28.02	3	DAY 353	H.M.S 16.28.02
4	DAY 353	H.M.S 23.28.02	5	DAY 354	H.M.S 05.28.02	6	DAY 354	H.M.S 11.28.02
7	DAY 354	H.M.S 18.28.02	8	DAY 355	H.M.S 00.28.02	9	DAY 355	H.M.S 06.28.02



CENTERLINE THERMOCLINES VS VERTICAL DISTANCE

0000000045 TEST 052 76.355 DEC 20 MANUAL DATA NOT INCLUDED FRAME 2

Figure 6-49. Test 52: Hold, Centerline Thermoclines vs Distance

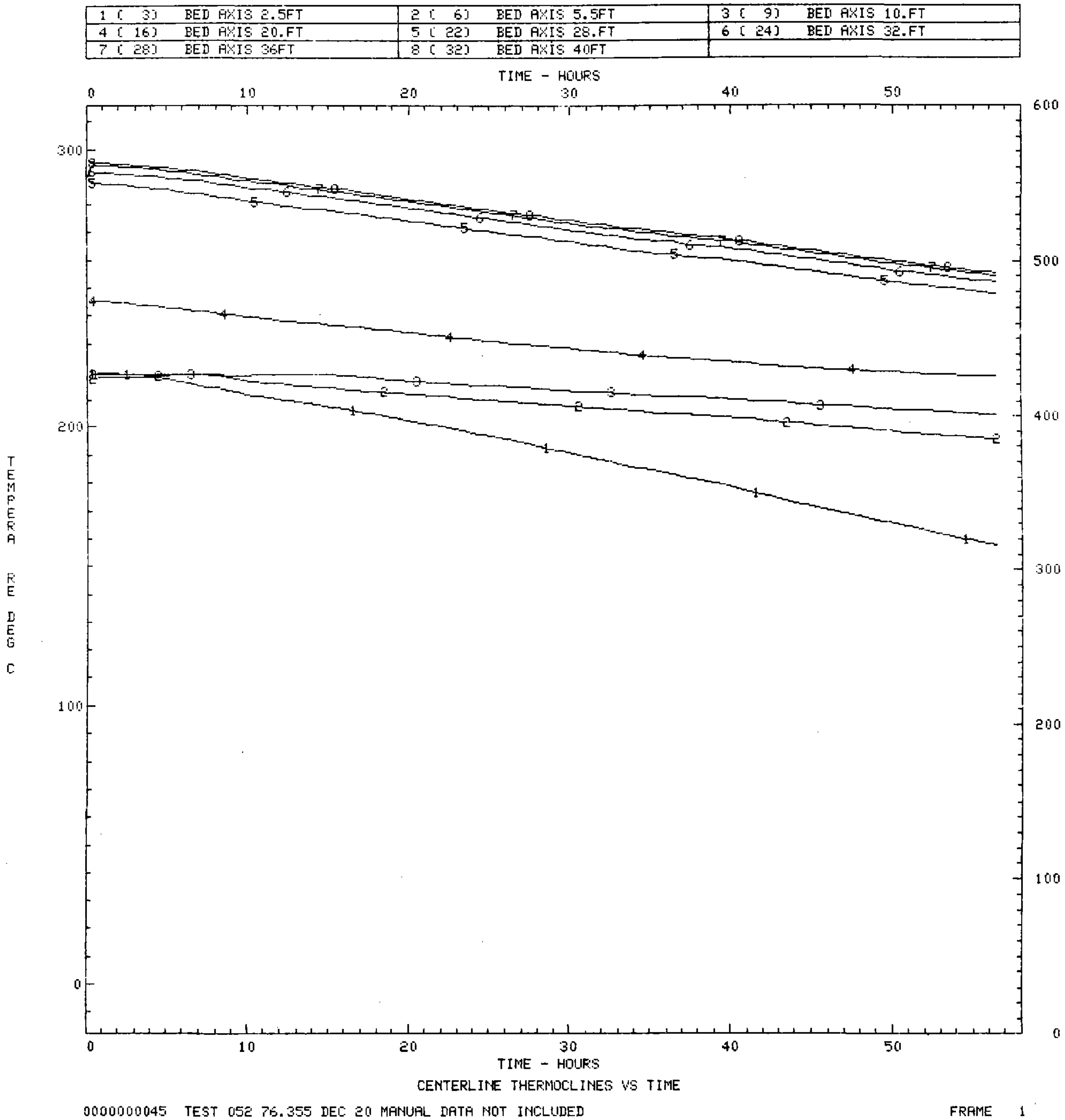


Figure 6-50. Test 52: Hold, Centerline Thermoclines vs Time

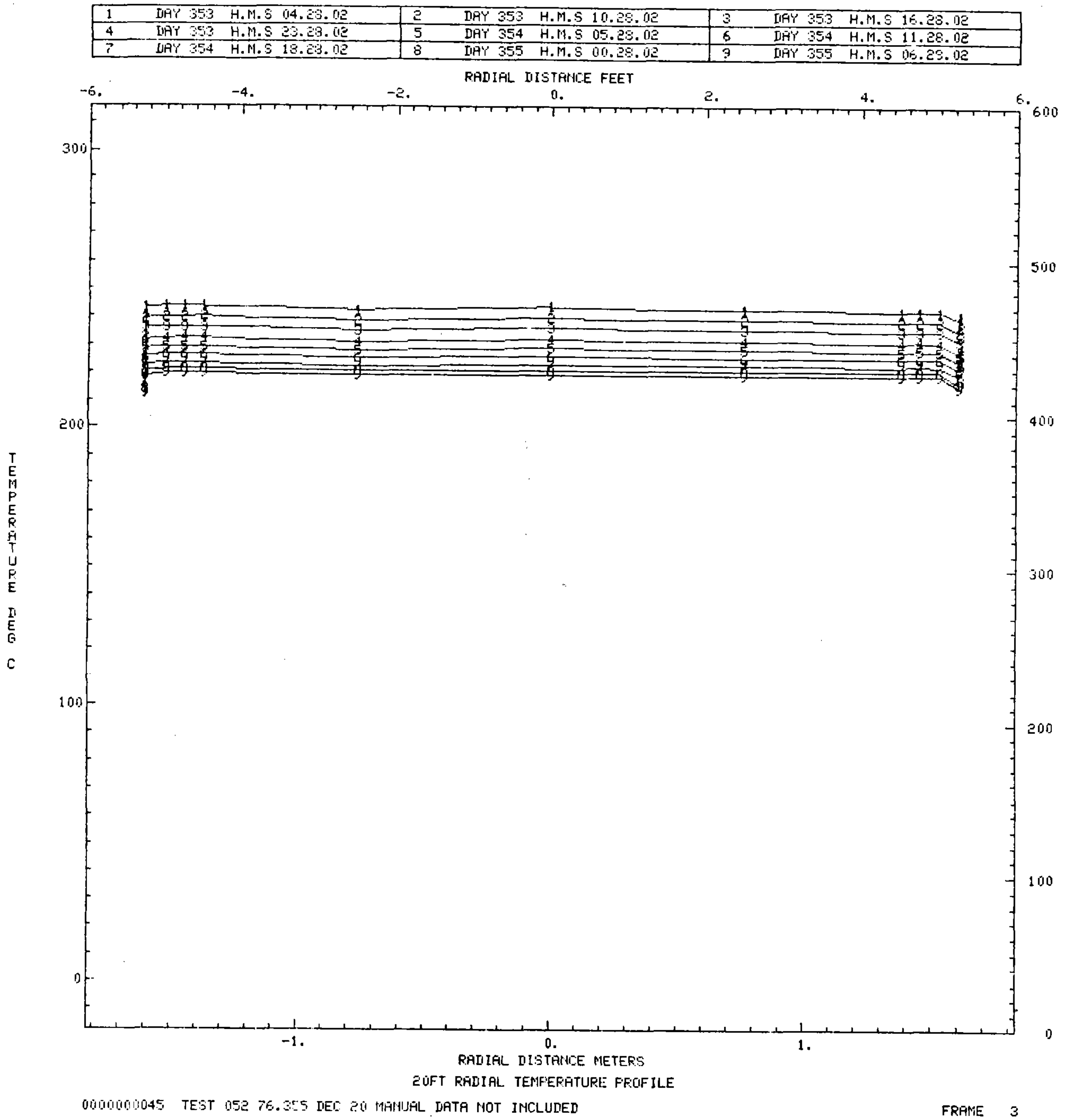


Figure 6-51. Test 52: Hold, 20-ft Radial Temperature Profile

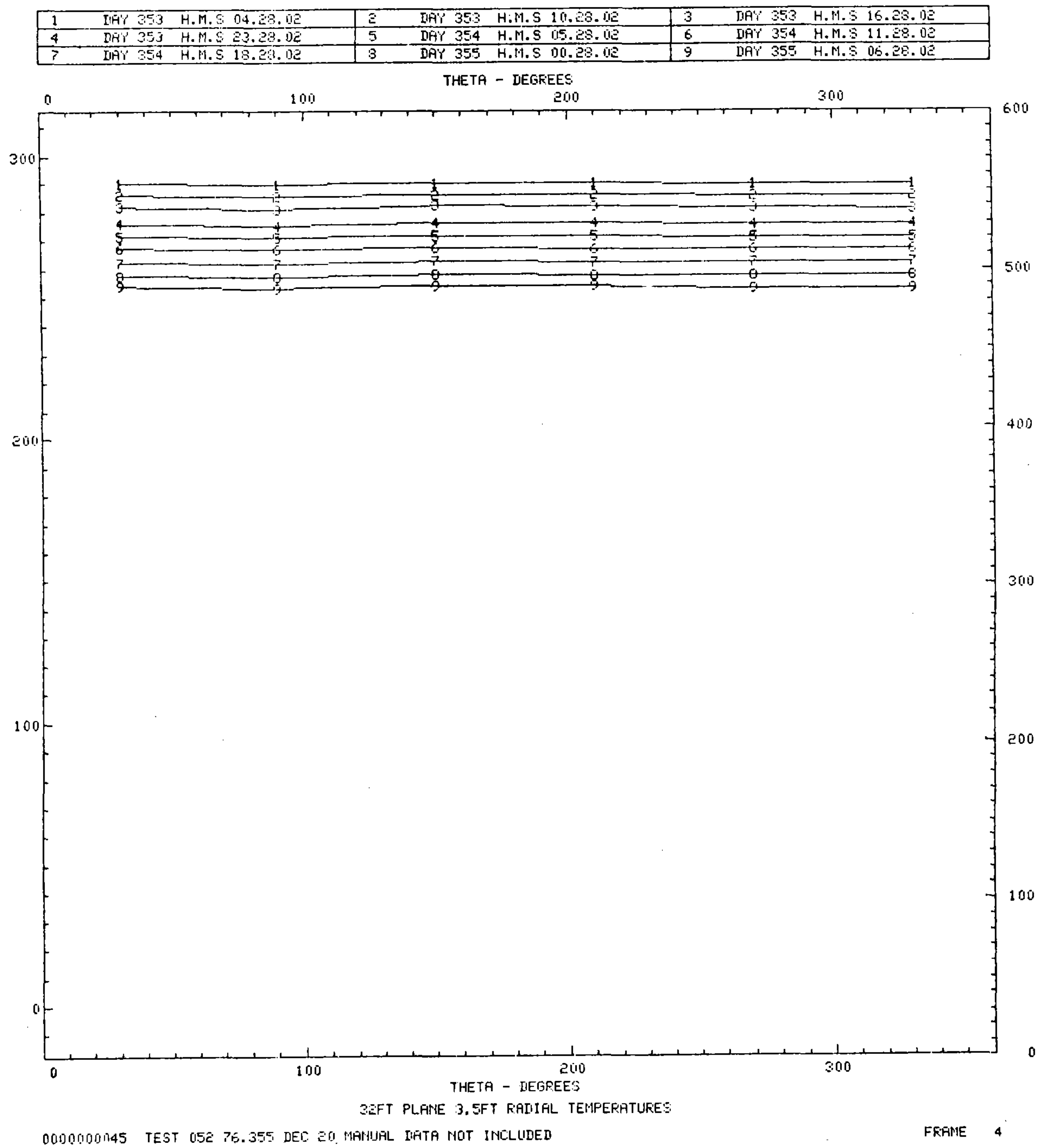
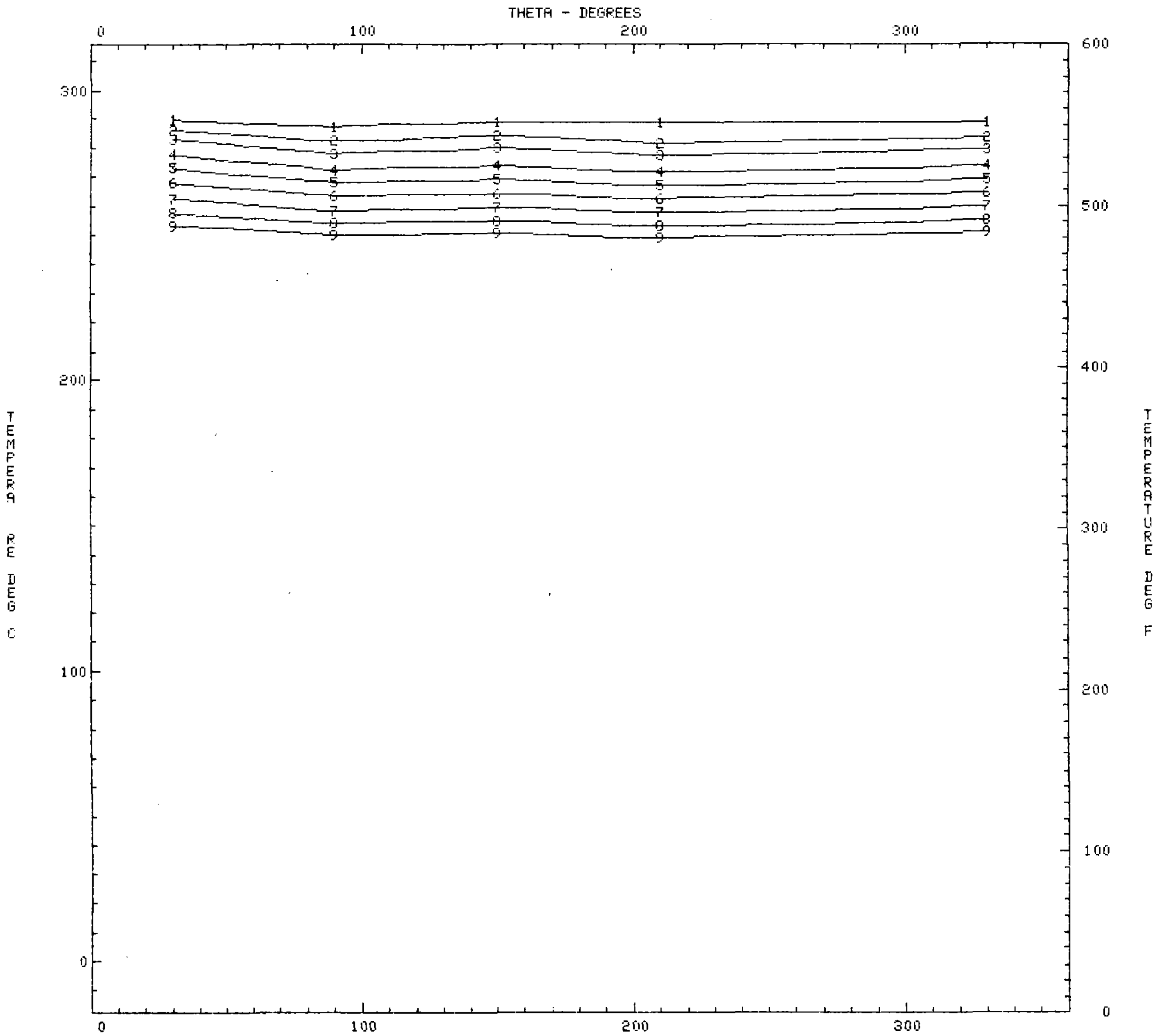


Figure 6-52. Test 52: 32-ft Level Circumferential Temperature Profile

1	DAY 353	H.M.S 04.28.02	2	DAY 353	H.M.S 10.28.02	3	DAY 353	H.M.S 16.28.02
4	DAY 353	H.M.S 23.28.02	5	DAY 354	H.M.S 05.28.02	6	DAY 354	H.M.S 11.28.02
7	DAY 354	H.M.S 18.28.02	8	DAY 355	H.M.S 00.28.02	9	DAY 355	H.M.S 06.28.02



40FT PLANE 3.5FT RADIAL TEMPERATURES

0000000045 TEST 052 76.355 DEC 20 MANUAL DATA NOT INCLUDED FRAME 5

Figure 6-53. Test 52: Hold, 40-ft Level Circumferential Temperature Profile

Data analysis included heat loss based upon measurements with the heat flow gage and estimates based upon total integrated heat capacity of the total system for a short period of time. The results and how they relate to the Pilot and Commercial Plants are plotted on Figures 6-54 and 6-55.

SRE Heat Loss

Overall heat loss of the TSU was on the order of 20% of the available extractable thermal energy over a 24-hr period when charged to maximum capacity. Although the lowest flux was measured in the side wall, the cylindrical side accounted for 60% of the total since it represented 89% of the total heat loss area. Of the remaining 40% loss, approximately 25% passed through the bottom and 15% through the top.

In designing the SRE TSU is was recognized that heat loss duplications of much larger units would be expensive and were not necessary to accomplish the results of the SRE. No attempt was made to bring losses down to the values representative of Pilot and Commercial Plant scale. Figure 6-54 shows both the actual experimental data for the modestly insulated SRE unit and a curve for the estimated heat loss versus size with insulation typical of the Pilot and Commercial Plants.

Scaling

Adjustments in thicknesses and conductivities necessary to achieve the design heat losses for the individual portions (side, bottom, top) are shown in Figures 6-54 and 6-55. In addition to the effect of reduced flux, the lower heat loss area to total volume ratio results in significant reduction in adjusting to Pilot and Commercial Plant scale. Figure 6-54 represents the relative heat loss over the scale range of the three sizes, 5-MWh SRE, 103.3-MWh Pilot, and 1,857-MWh Commercial Plant sizes for the insulation values typical of the Pilot and Commercial Plants.

The largest heat loss of the SRE is through the cylindrical wide wall which represented 89% of the heat loss surface. Side wall loss varied greatly depending upon the bed temperature. At times the lower region became cool and side losses in this region were greatly reduced. For the Pilot and

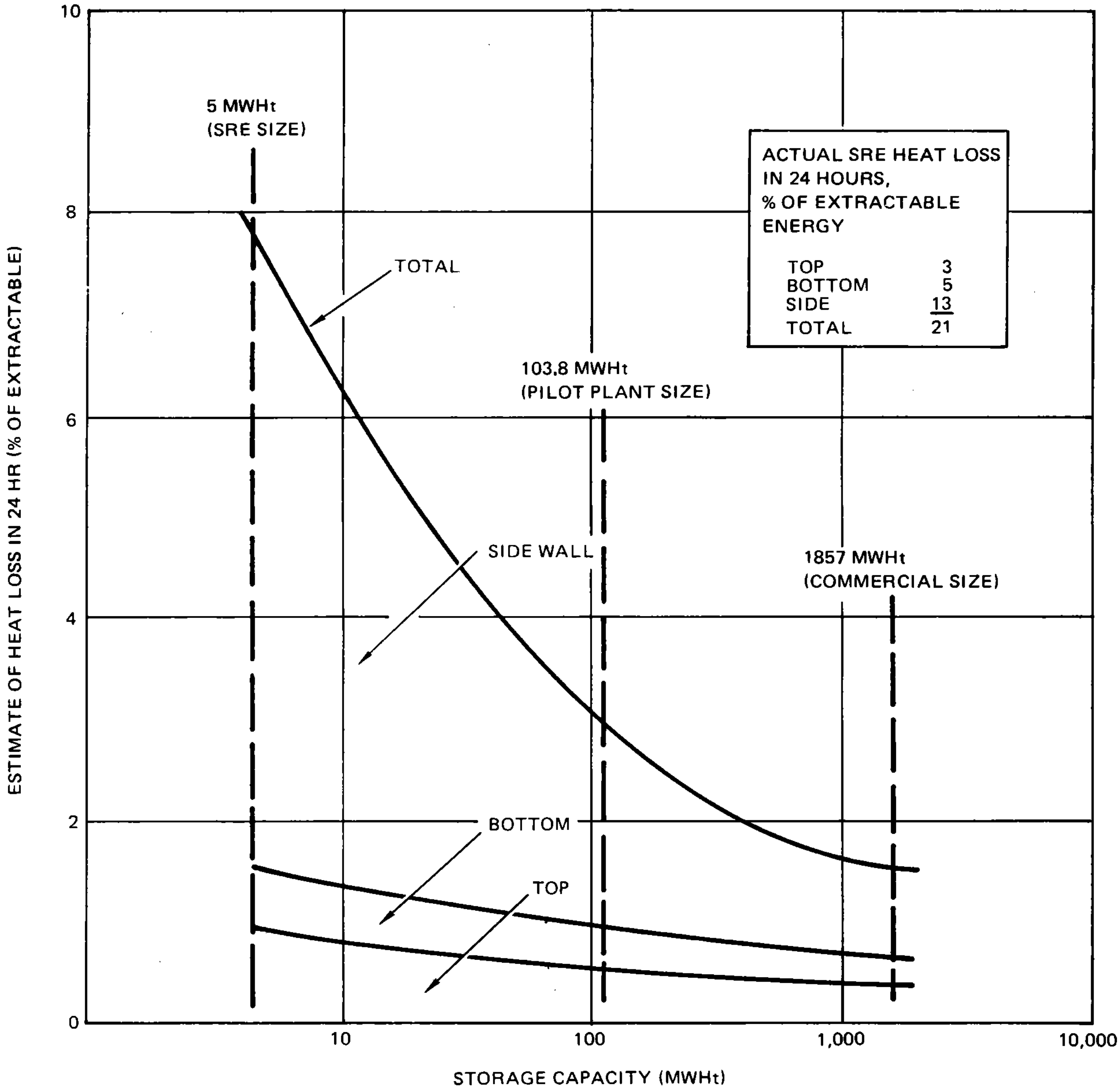


Figure 6-54. Estimate of TSU Heat Loss Scaling

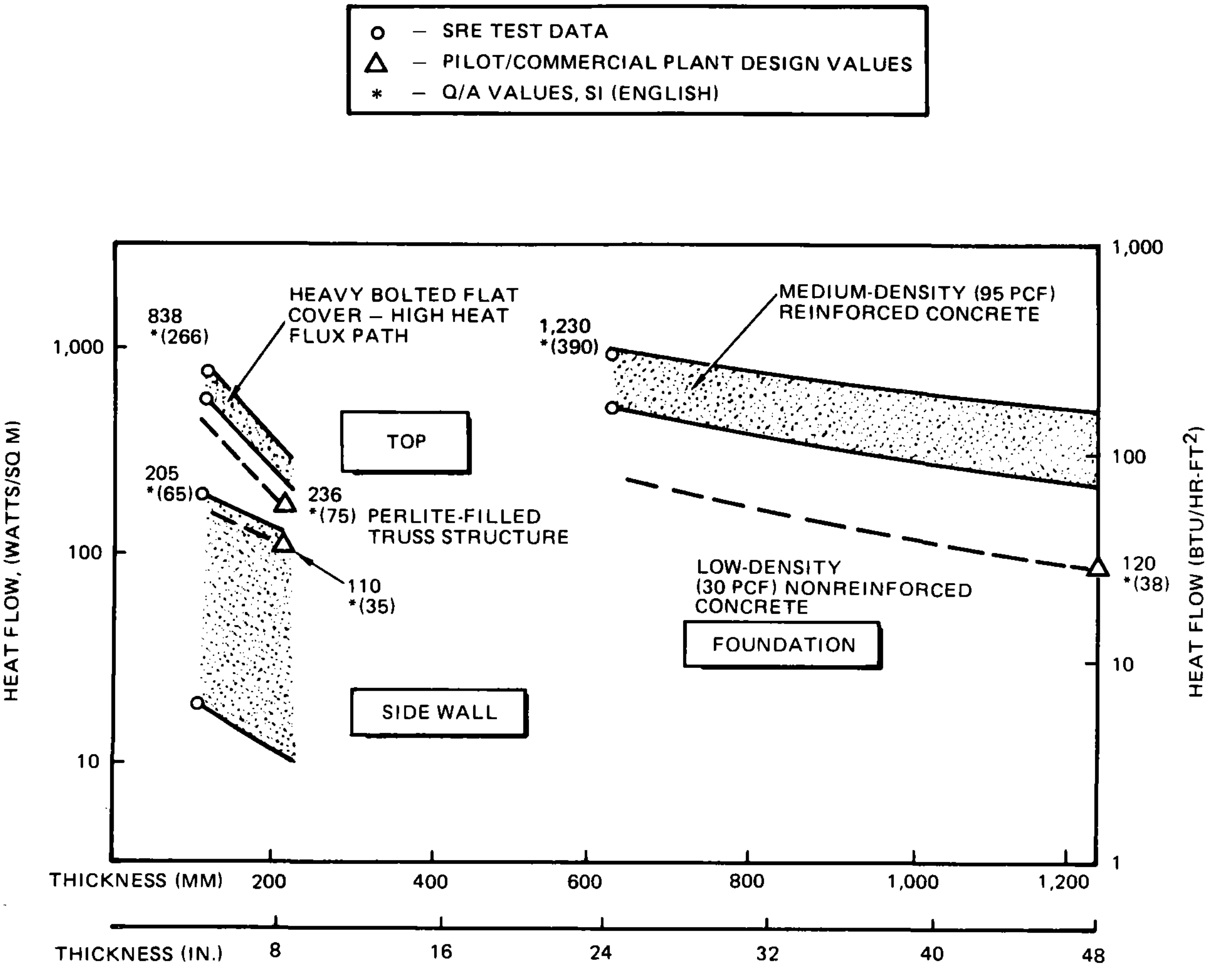


Figure 6-55. TSU Heat Flow Paths

Commercial Plants the heat flux was reduced from 65 to 35 W/cm^2 by doubling the thickness of the fiberglass insulation.

Heat losses through the top were aggravated by the heavy flange and support structure which provided a heat-leak path around the top insulation. The top was intentionally made heavy for the SRE to sustain pressure buildup in the ullage space. It was not known at the beginning of the program what value of ullage pressure would occur during water removal. During testing these were found to be low and controllable, allowing conventional lightweight construction for the Pilot and Commercial Plants. By completely covering the top with 200 mm (in.) of fiberglass-type insulation, the heat flux will be reduced from a value of 266 to 236 W/cm^2.

The bottom of the SRE had the highest heat loss flux, 1,230 W/cm^2, and is an order of magnitude higher than is allowable for the Pilot and Commercial Plants. Although the foundation is constructed of insulating concrete, it is much more conductive because of the amount of reinforcing steel used for structural purposes. A heavy structural base was necessary to provide earthquake and wind support for the relatively tall and narrow TSU configurations selected to facilitate thermocline operation. Since the Pilot and Commercial Plants will be much larger in diameter they will not need to be fastened to a hard foundation and thermal losses can be greatly reduced. The use of dry soil as a foundation is estimated that the base heat flux will be on the order of 120 W/cm^2.

Based on scaling the SRE results, it is estimated that overall TSU heat losses will be on the order of 2 to 3% for the Pilot and Commercial Plants.

6.3.4.3 Tank Wall Strain

Twenty channels of strain gage measurements were continuously recorded and are in the process of evaluation. The strain gage electrical output was designed to provide accurate readings if the values of wall stress approached the maximum predicted for which the tank was designed (passive load). For stress values corresponding to the minimum anticipated load (active) the values of electrical output are near the threshold value of the combined sensor recording system.

Preliminary analysis of selected strain gage readings in the lower portions of the tank indicate that strainvalues on the order of the active load and well below the passive load were measured during the test programs. Figure 6-56 is a summary of these strain gage readings from initial mounting prior to rock filling through the elevated temperature of the program and back to ambient temperature. Because of the low signal level and variability of strain gage readings over a wide temperature range, the strain gage values can only be considered as a trend. However, the trend is informative in that for the test conditions the strain values are low and for the number of cycles no or little residual strain remained after cooldown.

6.3.4.4 Manifold and Bed Pressure Drop Characteristics

Figure 6-57 provides representative values of pressure drop for the distribution manifolds and the rock/sand bed. These data were reduced for selective tests during which special care was used to bleed and vent pressure lines to the transducers to avoid errors that could be induced by the ullage pressure, and fluid static head which were relatively high compared to the manifold and bed pressure drops.

The manifold pressure drop was essentially the same whether charging or extracting although the few extraction points near 400 liters/min (100 gpm) indicate a possibility of a trend to slightly higher pressure drop during extraction at the higher flow rates. The shape of the curve is not well established by the data but tends to follow a straight-line relationship which would indicate laminar-flow characteristics.

Bed pressure drop data is not as consistent as the manifold pressure drop. Bed pressure drop was measured with a differential pressure transducer at the bottom referenced to the top of the tank with 6 mm (0.25 in.) tubing on the outside of the tank to minimize the variations in ullage and static head. Since the fluid in the tubing was at ambient temperature, a static pressure difference was always imposed, on the order of 17 KPa (2.5 psia), and varied depending upon the temperature of the fluid in the tank. Thus, during a typical charging or extraction the static head varied by as much as 3.4 KPa (0.5 psi) from the beginning to the end of the run. Precise dynamic bed

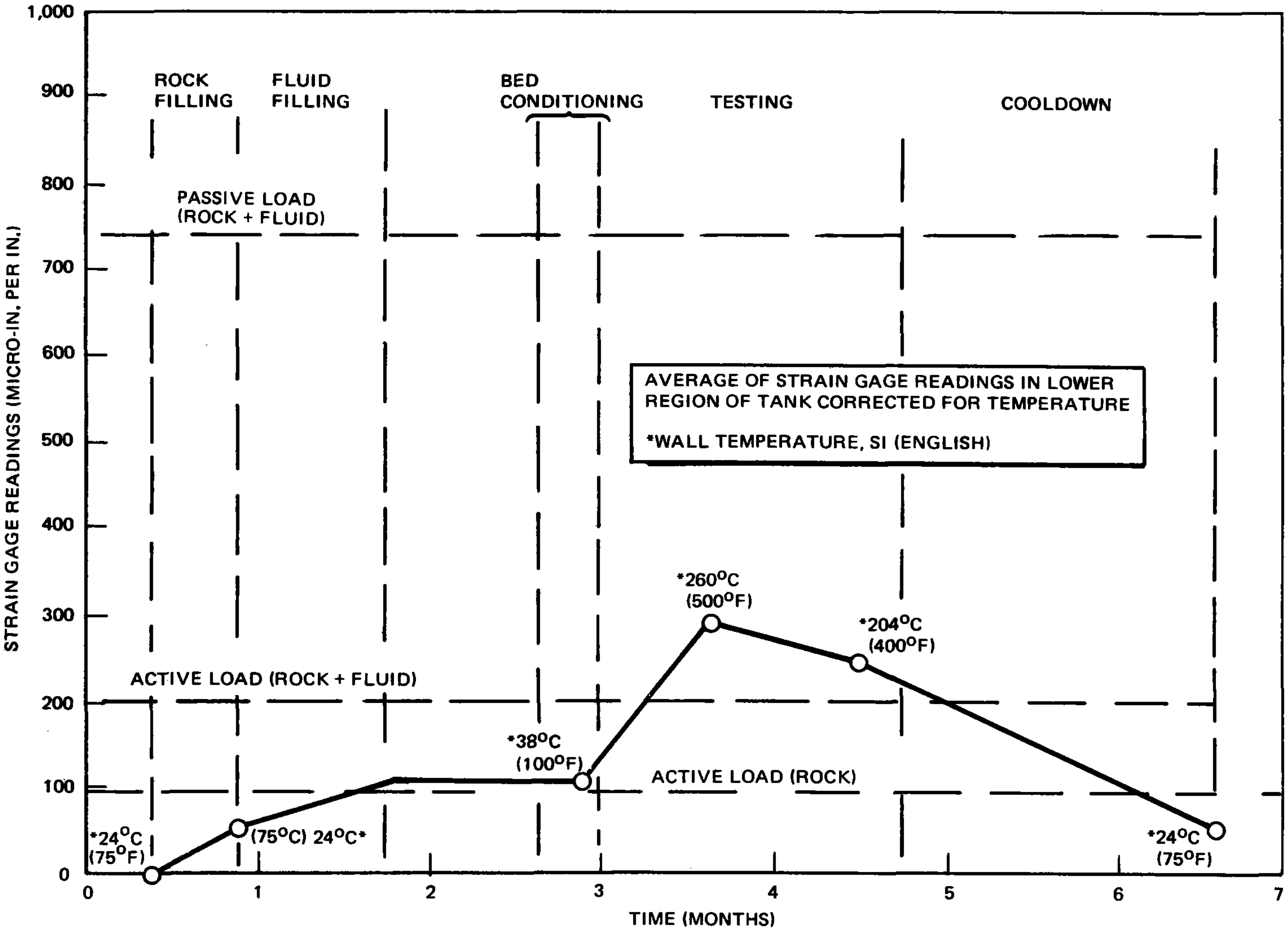
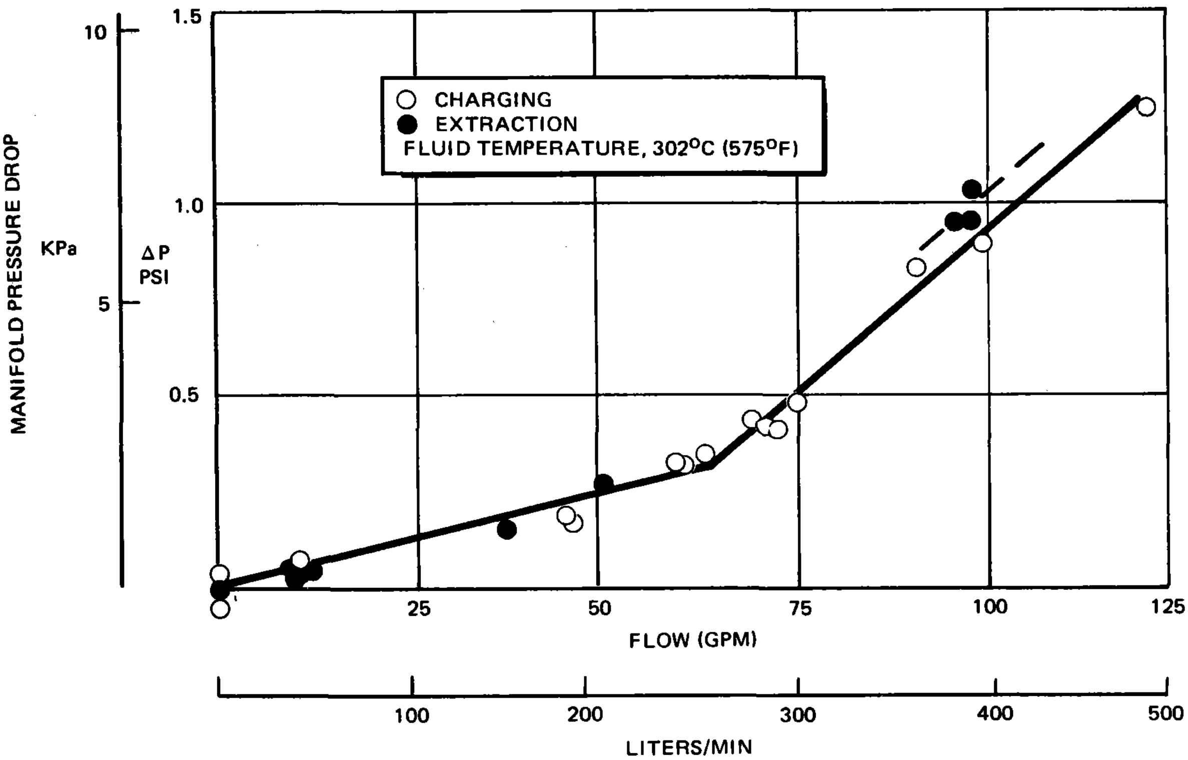


Figure 6-56. SRE TSU Tank/Rock Strain Measurements



BED TEMPERATURE °C (°F)	MASS, VELOCITY, KG/SEC-M^2 (LB/SEC-FT^2)	FLOW, LITER/MIN (GPM)	Bed ΔP KPA (PSI)	COMMENTS
204 (400) TO 302 (575)	0	0	0	NO FLOW
	0.54 (0.11)	380 (100)	2.5 (0.4)	CHARGING AND EXTRACTION

BED PRESSURE DROP

Figure 6-57. Manifold and Bed Pressure Drop

pressure drop was not derived from the data, but a conservative value (upper bound) is estimated to be 2. 5 KPa (0. 4 psi) at 380 liters/min (100 gpm (100 gpm).

6.3.4.5 Transient Performance

Response of the TSU and portions of the system applicable to the Pilot and Commercial Plants was rapid and reproducible. Heat-transfer fluid flows were established at selected values within 2 to 3 sec of pump activation. The TSU evidenced little or no thermal inertia since it did not cool to any significant degree. Lines, pumps, and valves produced the greatest thermal inertial which resulted in warmup periods of 10 to 15 min (from cold) before fluid temperatures stabilized completely. The effects are directly scalable with appropriate factors to the Pilot and Commercial Plants.

6.3.4.6 Bed Conditioning

Removal of water from the rock bed was accomplished easily, with operations as planned. Initial heating of the bed produced a considerable amount of water through the liquid overflow line relief valve. Although the volume of water removed was not measured accurately, it was estimated from the duration and approximate flow measurements with a bucket that several hundred gallons of water were eliminated from the tank during the bed conditioning. It might be noted that there was a rainstorm shortly before loading of the TSU rock bed. This same type of water removal was noticed earlier in Rocketdyne's small-scale laboratory test loop.

During water removal, bubbling through the rock bed could be heard near the top of the tank. Unusual thermocline behavior also accompanied the water removal. During heat addition a significant amount of warm liquid was flowing down the walls and mixing with the tank outflow resulting in a continued increase in outlet temperature all during the charging period.

It is most likely that steam bubbles rising through the bed had a higher population density in the center and thus produced a higher flow resistance in the center than at the tank walls, which were cooler. This induced a greater flow adjacent to the wall which resulted in warm fluid bypassing the main portion of the bed and mixing with and increasing the temperature of the normally cool outflow. Once water removal was complete, the warming of the cool outflow ceased and thermoclines were fully developed with uniform cross-section profiles.

6.3.4.7 Silt Removal

An important consideration in operation of the dual medium concept is the elimination of any adverse effects of particulates (dust, sand, silt) that might be picked up by the fluid and carried through the system.

The system was designed and operated with two large in-line quick-cleaning filters in the charging circuit for removal of any particulates. These filters are between the charging pump and TSU, are connected in parallel and are valved so that the charging flow loop can be operated with either or both in line. When first activated, both filters were used to minimize pressure drop between the TSU and the pump inlet and avoid the possibility of pump cavitation. At room temperature the Caloria HT42 has a viscosity similar to SAE 10 motor oil and filter pressure drop is much higher than at the normal operating temperatures (new 300°C).

Initial cold flow operation (no heating) occurred as expected for approximately 1 hr when pump cavitation occurred. Both filter elements were removed and examined. Each element was completely covered with approximately 3 mm of fine silt. The filter elements were cleaned and replaced with flow returning to normal upon restarting of the system.

During subsequent operation, the time between cleaning increased markedly. During the last 100 hr of system operation, filter cleaning occurred infrequently and did not interrupt normal operation. Cleaning periods increased to a value that would require only minimal maintenance in the Pilot and Commercial Plants (in the range of 1 cleanout per week to 1 per month, Figure 6-58).

6.3.4.8 Operational Modes

The SRE subsystem was operated in all modes simulating or duplicating the Pilot and Commercial Plants. Warmup, start and stop transients, charging,

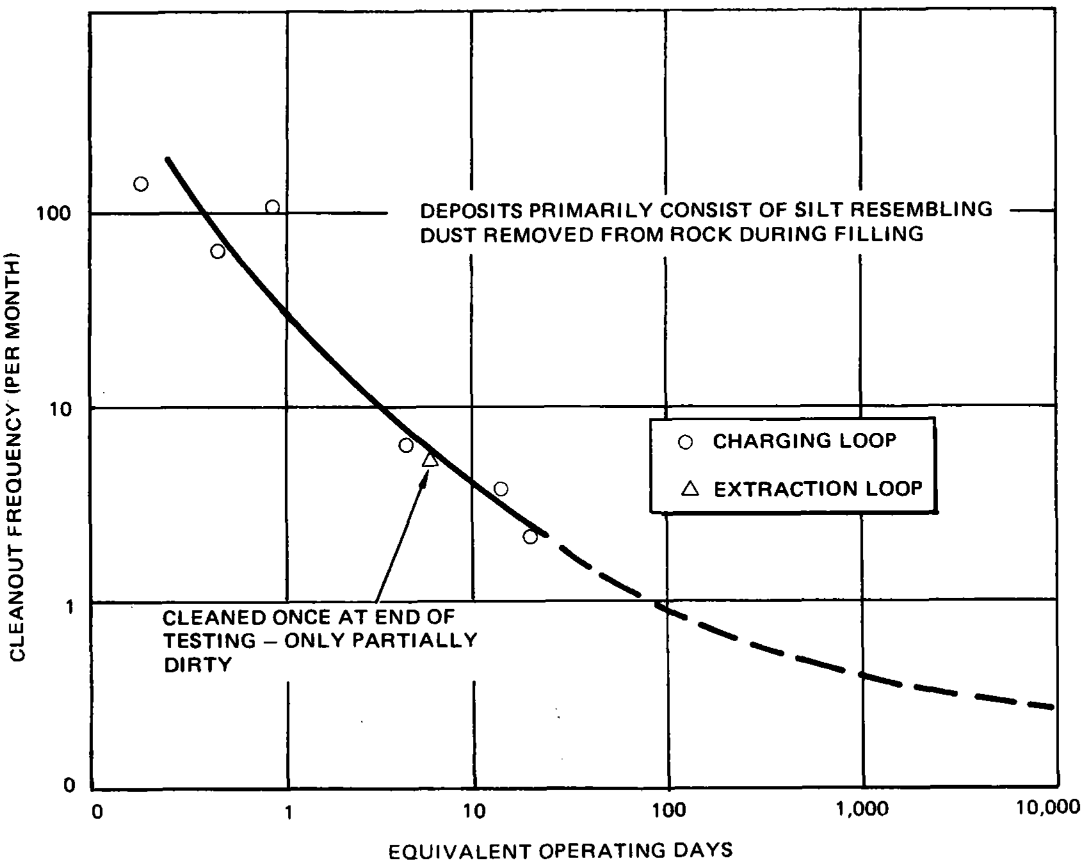


Figure 6-58. SRE Subsystem Particulate Removal

and extraction, and standby hold were all replicated over the complete range of operating conditions.

6.3.5 Conclusions

Preliminary conclusions from the TSS test program are as follow:

1. The practicality and high performance of the dual-medium thermal storage concept have been demonstrated on a significantly large scale. Scale-up to the 10-MWe Pilot Plant can now be made with high confidence.
2. The development and vertical movement of a sharp thermal gradient of thermocline in a dual-medium (liquid-solid) system is a predictable and reproducible phenomenon.
3. Thermocline integrity and stability are high enough to provide high-energy recovery performance for daily operation.
4. Partial charging and extraction, and repeated cycling, do not significantly degrade the thermocline.
5. Fluid flow and temperature uniformity are high across the thermal storage unit cross section. Flow channeling and rat-holing are insignificant in the SRE unit.
6. Heat loss from the thermal storage unit is not severe. With larger units or improved insulation, little energy would be lost over typical hold periods of 1 to 8 hr.
7. The use of low-cost, commercial river bed gravel provides adequate performance. Conventional filters incorporated in the charging circuit are adequate to rapidly remove, in situ, dust from the rock bed.
8. For the test time accumulated, tank wall stresses were far below design values (i.e., passive load design), and there was no indication of high stresses induced from rock settling and packing. However, this may be a time-related effect and should be verified by continuing cyclic heating and cooling over a longer period of time.
9. Initial removal of water from the bed occurs readily and controllably during initial bed heating. No special design features other than adequate vapor venting are necessary for water removal.

6.4 REFERENCES

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Appendix A

REQUIREMENTS SPECIFICATION FOR PILOT PLANT THERMAL STORAGE SUBSYSTEM

1.0 SCOPE

This specification establishes the performance, design, and test requirements for the Pilot Plant thermal storage subsystem (TSS).

2.0 APPLICABLE DOCUMENTS

The equipment, materials, design, and construction of the thermal storage subsystem shall comply with all Federal, state, local, and user standards, regulations, codes, laws, and ordinances which are currently applicable for the selected site and using power utility. These shall include but not be limited to the government and nongovernment documents itemized below. If there is an overlap in or conflict between the requirements of these documents and the applicable Federal, state, county, or municipal codes, laws, or ordinances, that applicable requirement which is the most stringent shall take precedence.

2.1 GOVERNMENT DOCUMENTS

2.1.1 Specifications

- Occupational Safety and Health Administration (OSHA) Standards Title 29 - Labor: (a) CFR Part 1910 - Occupational Safety and Health Standards, (b) CFR Part 1926 - Safety and Health Regulations for Construction.
- Title 8, California Administrative Code, California Occupational Safety and Health Regulations.

2.1.2 Other Publications

- National Motor Freight Classification 100B - Classes and Rules Apply on Motor Freight Traffic
- Uniform Freight Classification 11 - Railroad Traffic Ratings Rules and Regulations
- CAB Tariff 96 - Official Air Transport Rules Tariff
- CAB Tariff 169 - Official Air Transport Local Commodity Tariff
- R. H. Graziano's Tariff 29 - Hazardous Materials Regulations of the Department of Transportation.
- CAB Tariff 82 - Official Air Transport Restricted Articles Tariff

2.2 NONGOVERNMENT DOCUMENTS

2.2.1 Standards

- ASME Boiler and Pressure Vessel Code, as applicable
  Section II, Material Specifications
  Section V, Nondestructive Examination
  Section VIII, Pressure Vessels
  Section IX, Welding and Brazing Qualifications
- American Petroleum Institute (API), Standard 650, Welded Steel Tanks for Oil Storage
- American National Standards Institute (ANSI) B31.1, Power Piping
- ANSI Standards for Safety and Construction, as applicable
- National Fire Protection Association (NFPA)
- National Fire Codes - 1975 (Vol 1-15) as applicable
- Standards of the American Institute of Steel Construction and American Concrete Institute.
- National Electrical Code, NFPA 70-1975 (ANSI CI-1975)
- Building Codes of the County of San Bernardino

3.0 REQUIREMENTS

3.1 THERMAL STORAGE SUBSYSTEM DEFINITION

The thermal storage subsystem (TSS) for the Pilot Plant shall provide a means of transferring to stored thermal energy a portion of the thermal output from the receiver subsystem and subsequently transferring stored thermal energy to steam in a form suitable for generating electrical power with a conventional turbine-generator. The thermal storage subsystem shall buffer the electrical power generating subsystem (EPGS) from excessive variations in insolation, and extend the plant's generating capacity into periods with low or no insolation.

The TSS shall consist of:

(a) The thermal storage unit, including the storage media, containers, structures, internal piping, and insulation required for the actual storage of thermal energy.

(b) The ullage maintenance unit with the valves, piping, and instrumentation required to maintain and control pressure and an inert atmosphere in the ullage of the thermal storage unit.

(c) The fluid maintenance unit, valves, and connections in the fluid circulation loops to provide refurbishment and makeup of the heat-transfer fluid.

(d) The desuperheater to reduce the temperature of the steam from the receiver downcomer to a level satisfactory for heating the heat-transfer fluid.

(e) The thermal storage heater, valves, fittings, and piping required to transfer and thermal energy from the incoming steam into the heat-transfer fluid and then into the complete thermal storage media.

(f) The steam generator, valves, fittings, and connections to the superheated steam header of the electrical power generation subsystem.

(g) The pumps, instrumentation, and controls required to regulate and direct the steam and fluid flows, temperatures, and pressures as appropriate to provide full flexibility of the thermal storage subsystem.

3.1.1 Thermal Storage Subsystem Diagram

The TSS relationship to and interfaces with the overall plant are shown in Figure A-1. The subsystem functional schematic diagram is given in Figure A-2, which shows all major components and their interrelationships.

3.1.2 Interface Definition

The interfaces between the TSS and other Pilot Plant subsystems are illustrated in Figure A-1, and those between major components of the TSS are shown in Figure A-2.

3.1.2.1 Thermal Storage Subsystem/Receiver Subsystem

Desuperheater Steam Inlet/Downcomer: Piping, connections, and mounting fixtures shall be provided to match those of the receiver downcomer. The TSS shall be designed to accept steam (1) at a pressure of 10.1 MPa (1465 psia), (2) at a temperature of 510°C (950°F), and (3) with thermal energy at rates up to 30 MWt.

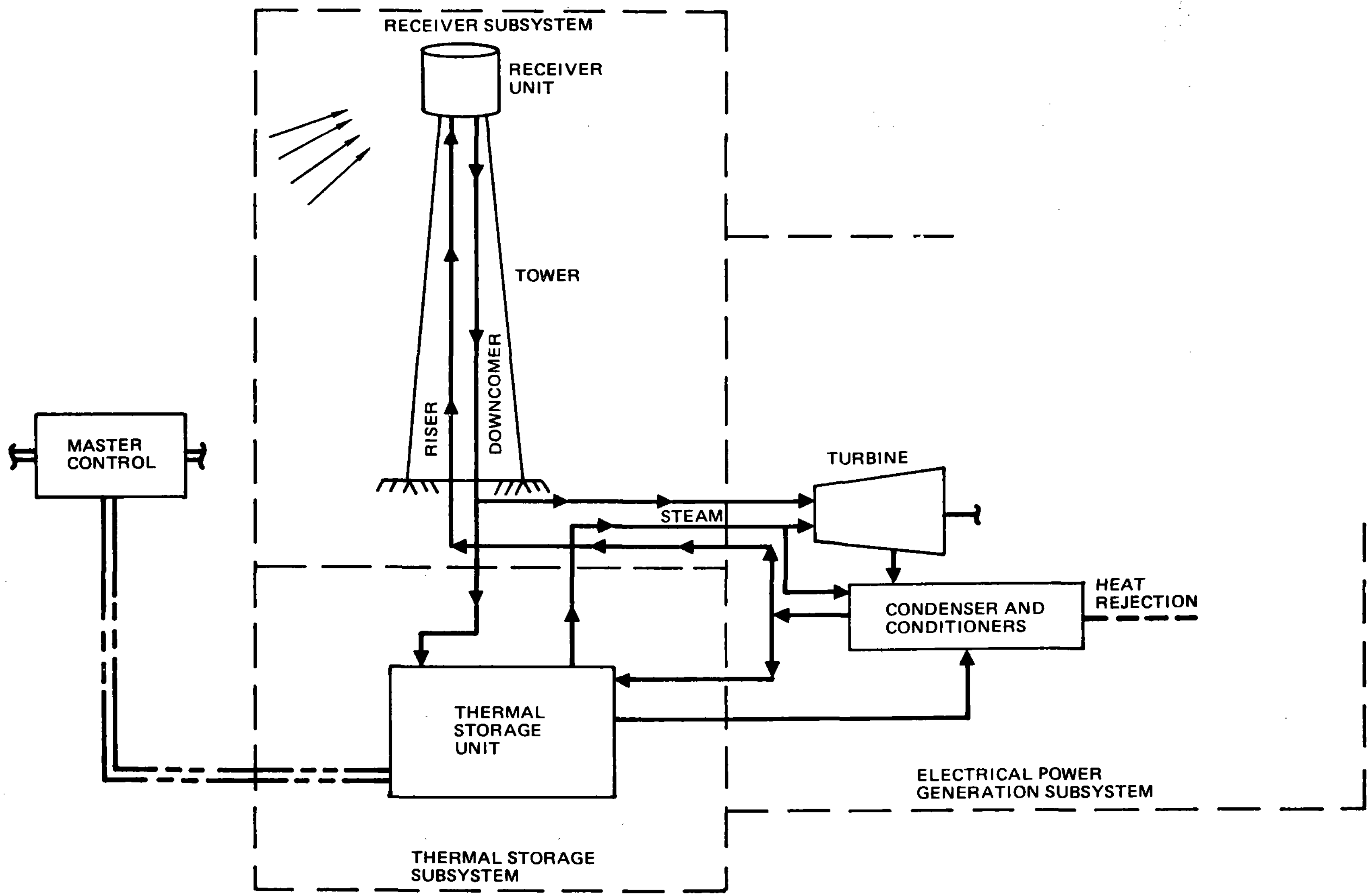

Figure A-1. Thermal Storage Subsystem Major Interfaces

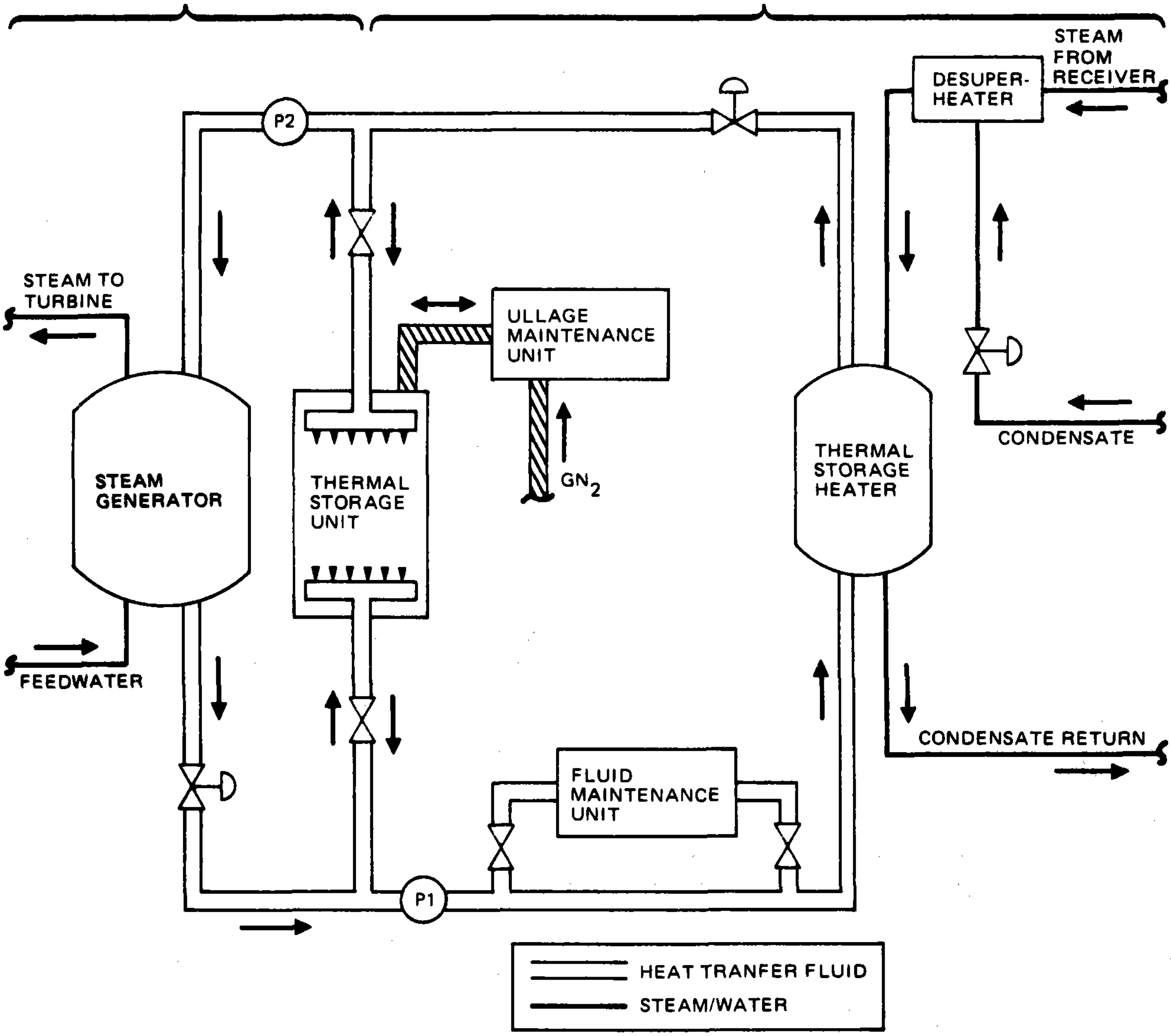

Figure A-2. Schematic of Pilot Plant Thermal Storage Subsystem

3.1.2.2 Thermal Storage Subsystem/EPGS

Steam Generator/Automatic Admission Port Header

Piping, connections, and mounting fixtures shall be provided to match those of the electrical power generation subsystem (EPGS) at the entrance of the turbine automatic admission port header. Steam shall be supplied by the thermal storage subsystem at a temperature of 277°C (530°F) and a pressure of 2.76 MPa (400 psia).

Condensate Loop/Steam Generator

Piping, connections, and mounting fixtures shall be provided to match those at the outlet of the ES condensate loop. The water to the TSS shall be at a temperature of 121°C (250°F) and a pressure of 2.9 MPa (420 psia).

3.1.2.3 Master Control

The subsystem must be capable of providing information to and responding to commands from both a computerized master control subsystem and a human operator.

3.1.3 Major Components

The major components comprising the thermal storage subsystem shall be:

(a) Thermal storage unit (including storage media, containers, structures, internal piping, insulation)
(b) Ullage maintenance unit
(c) Fluid maintenance unit
(d) Desuperheater
(e) Thermal storage heater
(f) Steam generator
(g) Circulation and condensate pumps
(h) Thermal storage control

3.2 CHARACTERISTICS

3.2.1 Performance

The thermal storage subsystem shall perform in accordance with the requirements of the following paragraphs.

The total subsystem life expectancy shall be 30 yr, with maintainability as specified in Paragraph 3.2.4. The instrumentation and control response times shall be adequate to ensure stable, flexible, safe operation in all normal and emergency operating modes.

3.2.1.1 Thermal Storage Unit

The thermal storage unit shall have an extractable storage capacity of not less than 103.8 MWHt, which is composed of 7.5 MWHt to provide a turbine hot start and 96.3 MWHt to permit the turbine-generator to supply 7 MWe net (7.8 MWe gross) for 3 hr following turbine startup. This extractable capacity is to be available following a full charge and a 20-hr hold period. The required charging rates are 1.5 MWt to 30 MWt. Required discharging rates are 3.1 to 32.1 MWt. The maximum allowable heat loss is 3% of extractable capacity in 24 hr, starting in a fully charged condition. The subsystem is required also to provide nighttime seal steam at a temperature of at least 135°C (275°F) and at a rate of 0.33 MWt for at least 16 hr.

3.2.1.2 Ullage Maintenance Unit

The ullage maintenance unit for the thermal storage unit shall be designed to fulfill two main functions: (1) to maintain at all times an inert atmosphere in the ullage (gas) space of the thermal storage unit, and (2) to maintain a controlled pressure range in the ullage, with provisions for venting steam during the initial startup. The inert gas shall be nitrogen.

3.2.1.3 Fluid Maintenance Unit

The fluid maintenance unit shall be designed to clean the heat-transfer fluid, both during initial startup and during normal operations. The unit shall remove solid particles down to a size of 0.18 mm and also provide processing of the fluid to remove decomposition products as necessary to prevent degradation of the fluid-side heat-transfer surfaces and to avoid significant changes in fluid flow properties.

3.2.1.4 Desuperheater

The desuperheater shall accept steam from the receiver subsystem at a pressure of 10.1 MPa (1465 psia) and a temperature of 510°C (950°F). The desuperheater shall also be capable of accepting steam from the receiver at a pressure of 10.1 MPa (1465 psia) and a temperature of 343°C (650°F).

3.2.1.5 Thermal Storage Heater

The thermal storage heater shall accept steam from the desuperheater at 10.1 MPa (1465 psia) and 343°C (650°F). The heater shall be capable of completely condensing the steam under these conditions and shall be capable of heating the heat-transfer fluid to at least 304°C (580°F) under maximum steam-flow conditions, based on entering heat-transfer fluid at 218°C (425°F). The heater shall be designed to ensure that the maximum film temperature of the fluid does not exceed 343°C (650°F).

3.2.1.6 Steam Generator

The steam generator shall be capable of producing steam at a temperature of 277°C (530°F) and a pressure of 2.76 MPa (400 psia). These performance figures are predicated on having available feedwater at a nominal temperature of 121°C (250°F) and pressure of 2.9 MPa (420 psia) at the necessary flow rates. The steam generator shall be designed for safe operation at the maximum pressures and temperatures which can be experienced during operation.

3.2.1.7 Thermal Charging Loop Pumps

The hot heat-transfer fluid pumps in the thermal charging loop (pump P1 in Figure A-2) shall be centrifugal, high-temperature pumps capable of pumping heat-transfer fluid safely at temperatures up to 316°C (600°F).

3.2.1.8 Energy Extraction Loop Pumps

The hot heat-transfer fluid pumps in the energy extraction loop (pump P2 in Figure A-2) shall be centrifugal, high-temperature pumps capable of pumping transfer fluid safely at temperatures up to 316°C (600°F).

3.2.2 <u>Physical Characteristics</u>

The TSS shall be designed to provide safe and reasonable ingress, egress, and access for inspection, maintenance, and repair of the structure, steam lines, utilities, instrumentation, and controls consistent with the availability requirements of Paragraph 3.2.5. The TSS shall be so located within the Pilot Plant as to minimize adverse effects on the other subsystems. All of the above shall be consistent with safety and cost considerations.

3.2.3 Reliability

High reliability shall be achieved in the subsystem design by providing adequate operating margins and using proven standard parts and conservative design practices such that the reliability performance shall not degrade the capability to achieve the availability specified in Paragraph 3.2.5 when operated at the planned conditions.

Single-point failures that prevent subsystem operation shall be eliminated wherever cost effective. In cases where it is impractical to eliminate such failure modes, suitable devices shall be used to detect and signal the occurrence of a failure.

3.2.4 Maintainability

The TSS shall be designed such that required service can be accomplished by personnel of normal skills with a minimum of nonstandard tools or special equipment. The subsystem shall be designed to provide malfunction indication and fault-isolation information required by the master control concerning critical components. Critical components are those components that, because of failure risk, down-time, or effect on the overall Pilot Plant performance, materially affect the capability to achieve the system availability requirements. Items which do not have a redundant mode of operation shall incorporate maximum capability for on-line repair or replacement. The subsystem shall be designed such that potential maintenance points can be easily reached, replaceable components such as electronic modules can be readily replaced, and elements subject to wear or damage can be easily serviced or replaced. Modular construction shall be employed for selected components in order to simplify maintainability, decrease subsystem downtime, and increase operating flexibility.

3.2.5 Availability

The TSS shall operate in accordance with the performance requirements of Paragraph 3.2.1 during 99.45% of its scheduled operating time, based on reliability and maintainability and exclusive of insolation conditions. Determinations of availability shall use a period of 1 yr as a time reference.

3.2.6 Environmental Conditions

3.2.6.1 General

The conditions described in Volume II, Pilot Plant Environmental Conditions, are representative of the site characteristics and the transportation and operating environments to be encountered by the TSS. For design purposes, safety margins shall be used that are commensurate with availability and performance requirements to ensure operation in accordance with Paragraph 3.2.1 during and/or after exposure to these conditions, as appropriate, for the 30-yr life of the system.

All critical (frangible) components of the TSS shall be designed or packaged such that the conditions described in Volume II do not induce a dynamic environmental condition which exceeds the structural capability of the component. All components shall be designed to withstand handling and hoisting inertial loads, considering number, location, and type of hoisting points.

Practices recommended in ASCE Paper 3269, Volume 126 and the Uniform Building Code, 1973, Volume 1 shall be employed in designing TSS elements for winds.

Subsystem components shall be protected from the electrostatic charging and discharging associated with sand and dust storms.

3.2.7 Transportability

Thermal storage subsystem materials and components shall be designed for transportability within applicable Federal and state regulations by highway and railroad carriers using standard transport vehicles and material handling equipment. The components, in their packaged condition, shall be capable of withstanding the climatic conditions, shock, and vibration environments of transportation. Whenever feasible, components shall be segmented and packaged in sizes that are transportable under normal commercial transportation limitations. Subsystem components that exceed normal transportation limits shall be transportable with the use of special routes, clearances, and permits, or the necessary parts for field assembly shall be transportable within these limits.

3.3 DESIGN AND CONSTRUCTION

The thermal storage unit shall be designed in accordance with API Standard 650 as modified by the ASME Boiler and Pressure Vessel Code - Section VIII for elevated temperature operation. Construction and inspection shall be in accordance with applicable API standards. All heat exchangers shall be designed, fabricated and inspected in accordance with the ASME Boiler and Pressure Vessel Code - Section VIII, Unfired Pressure Vessels. Piping shall be designed, fabricated, and inspected in accordance with the American National Standard Code for Pressure Piping.

3.3.1 Materials, Processes, and Parts

Materials of construction throughout shall be selected to ensure compatibility with the process fluids at the maximum operating conditions. The TSS components shall be fabricated from materials as specified in the applicable codes. Special attention shall be directed to preventing unnecessary use of costly materials. No exotic or potentially toxic materials shall be used. Wherever possible, standard and commercial parts shall be used. Specifically, the following shall be commercial parts:

(a) Heat-transfer fluid pumps
(b) Water pumps
(c) Thermal storage heater
(d) Steam generator
(e) Valves and controls

3.3.2 Electrical Transients

The subsystem operation shall not be adversely affected by external or internal power-line transients caused by normal switching, fault clearing, or lightning. Switching transients and fault-clearing functions shall require less than six cycles of the fundamental frequency (100 ms) and shall be limited to 1.7 P.U. voltage (1.7 per unit or 170%). Lightning arresters shall be installed which will limit the resultant line voltage to 5 PU on a line-to-ground basis during the time interval of peak current. Components of the TSS shall be shielded from the lightning threat specified in Volume II. Shielding shall protect the electrical components from both the bound-charge and induced-current threats.

3.3.3 Electromagnetic Radiation

The TSS shall be designed to minimize susceptibility to electromagnetic interference and to minimize the generation of conducted or radiated interference.

3.3.4 Nameplate and Product Marking

All deliverable end items shall be labeled with a permanent nameplate listing, as a minimum, manufacturer, part number, change letter, serial number, and date of manufacture.

3.3.5 Workmanship

The level of workmanship shall conform to practices defined in the codes, standards, and specifications applicable to the selected site and the using power utility. Where specific skill levels or certifications are required, current certification status shall be maintained, with evidence available for examination. Where skill levels or details of workmanship are not specified, the work shall be accomplished in accordance with the level of quality currently in use in the construction, fabrication, and assembly of Commercial Plants. All work shall be finished in a manner such that it presents no unintended hazard to operating and maintenance personnel, is neat and clean, and presents a generally uniform appearance.

3.3.6 Interchangeability

Major components, circuit cards, and other items with a common function shall be provided with standard tolerances and connector locations to permit interchange for servicing. Components that have similar appearance but different functions shall incorporate protection against inadvertent erroneous installation through the use of such devices as keying, connector size, or attachment geometry.

3.3.7 Safety

The TSS shall be designed to minimize safety hazards to operating and service personnel, the public, and the equipment. The TSS shall be designed to shut down safely in the event of the failure of any single component.

Electrical components shall be insulated and grounded. All parts or components associated with elevated temperatures shall be insulated against contact with or exposure to personnel. Any moving elements shall be shielded to avoid entanglement, and safety override controls and interlocks shall be provided for servicing. Isolation valves shall be provided on all major assemblies and on all interface utility lines to permit isolation and shutdown of assemblies and segments of the subsystem. Piping shall be designed, fabricated, and inspected in accordance with the American National Standard Code for Pressure Piping, Section 31.1, and shall be of welded construction wherever practical. Concrete and/or earth berms and dikes shall be provided to contain the maximum quantity of heat-transfer fluid which can be emptied from all above-grade sections of the subsystem. The storage tank, heat exchangers, and piping shall be visually inspected and leak-checked after assembly but before insulation is installed. Safety showers and eye-washes shall be provided. Ladders, handrails, and platforms shall meet OSHA standards. Fire-protection equipment shall be provided.

3.4 DOCUMENTATION

3.4.1 Characteristics and Performance

Equipment functions, normal operating characteristics, limiting conditions, test data, and performance curves, where applicable, shall be provided.

3.4.2 Instructions

Instructions shall cover assembly, installation, alignment, adjustment, checking, lubrication, maintenance, and operation. All phases of subsystem operation shall be addressed including startup, normal operation, regulation and control, shutdown, and emergency operations. A guide to troubleshooting instruments and controls shall be provided.

3.4.3 Construction

Engineering and assembly drawings shall be provided to show the equipment construction, including assembly and disassembly procedures. Engineering data, wiring diagrams, and parts lists shall be provided.

3.5 LOGISTICS

Elements required to support the TSS are:

- (a) Maintenance
  - (1) Support and test equipment
  - (2) Technical publications
  - (3) Field service
  - (4) Data file
- (b) Supply
  - (1) Spares, repair parts, and consumables
  - (2) Transportation, handling, and packaging
- (c) Facilities

Plans shall be provided to ensure proper logistics support.

3.6 PERSONNEL AND TRAINING

The Pilot Plant is to be installed and checked out by contractor personnel, then taken over and operated as a Commercial Plant by utility personnel. Operation and maintenance personnel requirements shall be satisfied by recruitment from the established utility labor pool. Unique aspects of the TSS may dictate a need for training existing utility people but shall not establish a need for new skills or trades.

3.7 PRECEDENCE

Specific characteristics and requirement precedence shall be established based on system cost-effectiveness sensitivity analyses. This specification has precedence over documents referenced herein. The contractor shall notify the procuring activity of each instance of conflicting, or apparently conflicting, requirements within this specification or between this specification and a referenced document.

4.0 QUALITY ASSURANCE PROVISIONS

4.1 GENERAL

4.1.1 Responsibility for Tests

All tests shall be performed by the contractor. These tests may be witnessed by ERDA or its representatives, or the witnessing may be waived.

In either case, substantive evidence of hardware compliance with all test requirements is required.

4.1.2 General Test Requirements

Tests shall be conducted in accordance with a test plan to be prepared by the contractor and approved by ERDA. Tests shall be classified in the test plan as:

(a) Development tests for the purpose of determining the design.

(b) Acceptance tests for the purpose of verifying conformance to design and determinating acceptability of product for further operations or for delivery.

(c) Qualification tests for the purpose of verifying adequacy of design and method of production to yield a product conforming to specified requirements.

Where performance of the TSS is to be evaluated in conjunction with another subsystem, the test is regarded as a system test.

4.1.3 Previous Tests

Maximum use shall be made of test data available from the subsystem research experiments and hardware tests already completed. Where conformance to TSS requirements can be established at less cost by analysis of such data, tests shall not be repeated.

4.2 SPECIFIC TEST REQUIREMENTS

Specific required tests are defined herein. Additional tests shall be defined by the contractor in the course of design and development where necessary to properly validate the performance and design integrity of the TSS.

4.2.1 Subsystem Performance Tests

Tests shall be defined for evaluation of the TSS as fully assembled in order to ensure compatibility before attempting system tests.

4.2.1.1 Subsystem Integrity

The integrity of the complete subsystem when installed shall be verified by subjecting it to hydrostatic pressure testing and to cold-flow and hot-flow checkouts.

4.2.2 Assembly and Subassembly Performance Tests

Tests shall be defined to include but not be limited to the following:

(a) Charging loop flow control

(b) Discharge loop flow control

(c) Control equipment

4.2.3 Materials and Processes Control Tests

Necessary tests shall be defined.

4.2.4 Life Tests and Analyses

Major subassemblies may be subjected to accelerated- or extended-life testing or analysis where data are not sufficient to estimate MTBF and MTBR for the given design configuration. Consideration shall be given to the need for testing such components as seals, pumps, valves, and heat exchangers to establish maintenance and life characteristics.

4.2.4.1 Mean Time Before Failure or Replacement

Sufficient data shall be gathered from the above life tests and also from component suppliers to estimate the MTBF and MTBR for the given design configuration.

4.2.5 Engineering Critical Component Qualification Tests

Components for which reliability data are not available shall be subjected to sufficient accelerated- or extended-life testing to estimate their impact on overall device MTBF.

4.3 VERIFICATION OF CONFORMANCE

Verification that the requirements of Sections 3.0 and 5.0 of this Appendix are fulfilled shall be performed by the contractor by the following methods:

(a) Inspection – examination and measurement of product

(b). Analysis – examination of the design and associated data, which may include relevant test information.

(c) Similarity – demonstration or acceptable evidence of the performance of a product which is sufficiently similar to permit conformance to be inferred.

(d) Test — functional operation or exposure under specified conditions to evaluate product performance

(e) Demonstration — exhibition of the product or service in its intended modes and conditions.

4.3.1 Hardware Acceptance for Pilot Plant

The contractor shall provide a system by which conformance of hardware to the approved design and any authorized changes thereto will be verified prior to initiation of subsystem-level tests. This verification of conformance to design shall include proof by assembly and the examination of records as elements of inspection. A record shall be made of each inspection or test performed and of the verification. In addition, evidence shall be maintained of satisfactory accomplishment of inspections and tests required by codes and standards applicable to the thermal storage subsystem.

5.0 PREPARATION FOR DELIVERY

5.1 GENERAL

Packaging for TSS components shall provide adequate protection during shipment by common carrier from the supplier to the first receiving activity for immediate use or temporary storage.

5.2 PRESERVATION AND PACKAGING

Subsystem components that may be harmed when exposed to the normal transportation and handling environments (Volume II) shall be protected by inert environments, barrier materials, or equivalent techniques.

5.3 PACKING

Shipping containers and their cushioning devices shall ensure protection of TSS components when exposed to the shock and vibrations loads defined in Volume II. Containers shall meet, as a minimum, the requirements in the following regulations as applicable to the mode of transportation used.

(a) National Motor Freight Classification (Highway Transportation)

(b) Uniform Freight Classification (Railroad Transportation)

(c) CAB Tariff 96 and 169 (Air Transportation)

(d) R. M. Graziano's Tariff 29 (for Dangerous Articles, Surface)

(e) CAB Tariff 82 (for Dangerous Articles, Air)

5.4 HANDLING AND TRANSPORTABILITY

Containers with gross weights exceeding 60 lb shall have skids and other provisions for handling by standard material handling equipment. When feasible, container sizes and configurations shall be compatible for efficient use of transport vehicles.

5.5 MARKING

Unless otherwise specified, container marking shall be in conformance with standard commercial practice.

Appendix B

THERMAL STORAGE FLOW SCHEMATICS

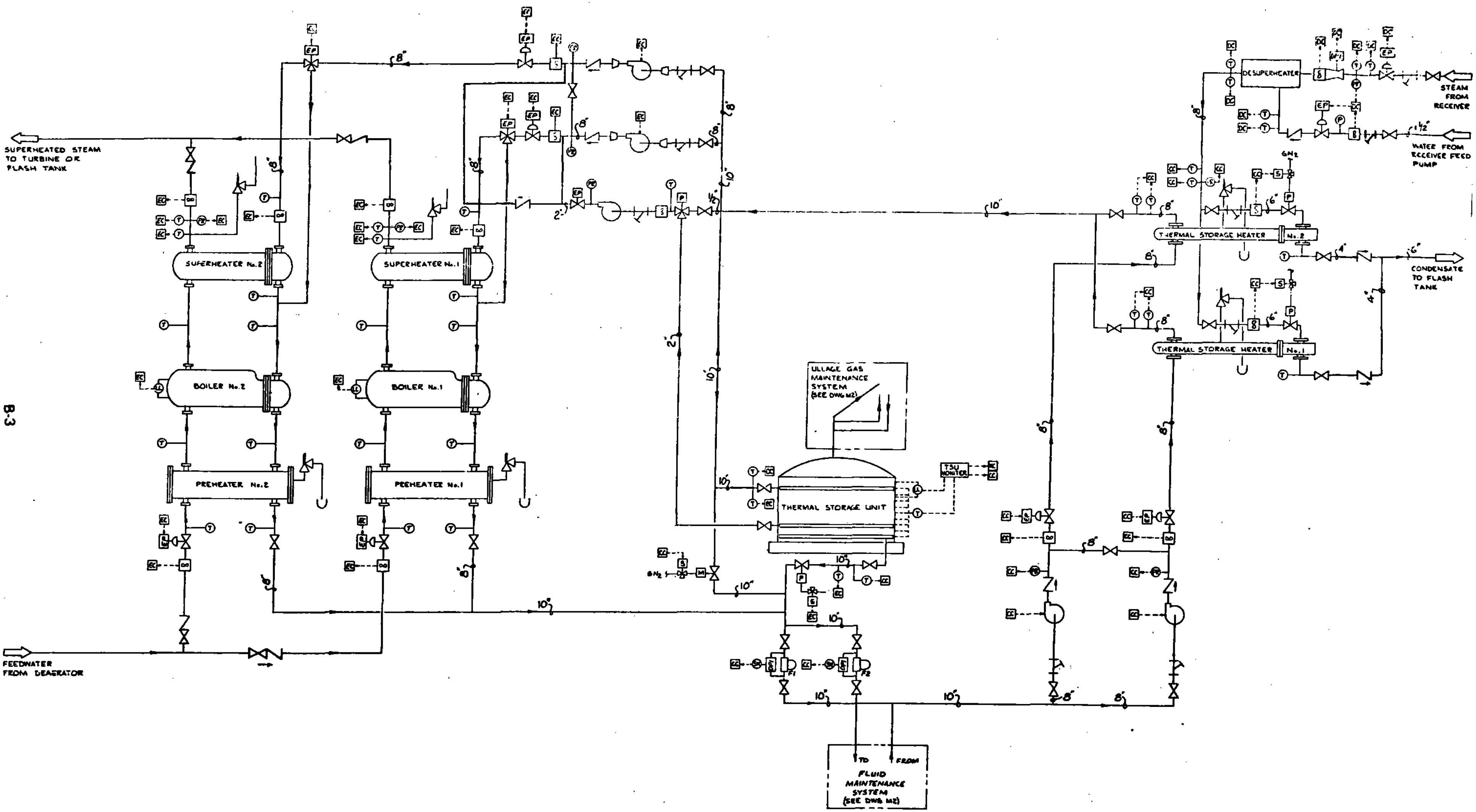

Figure B-1. Flow Diagram of 10-MW Pilot Plant

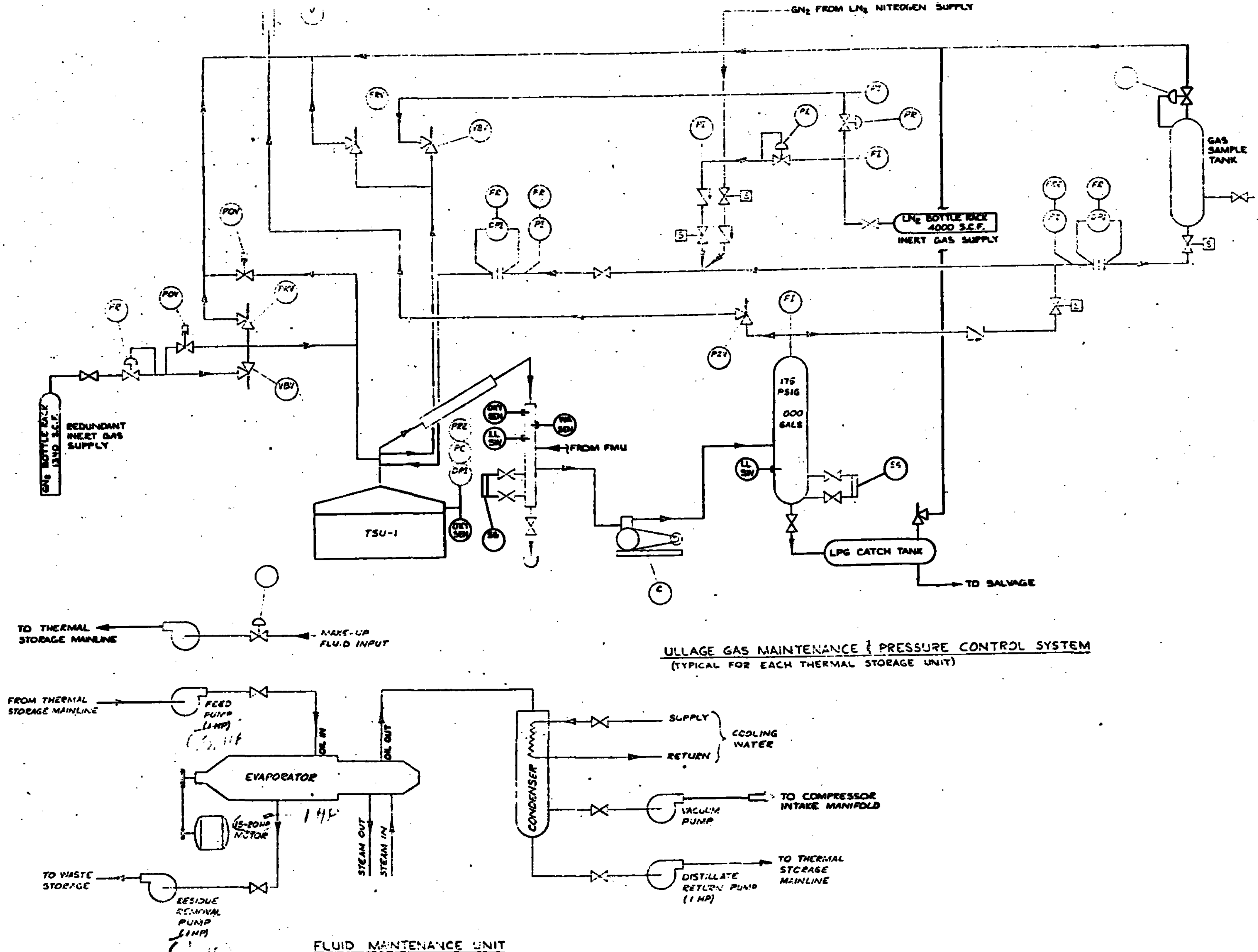

Figure B-2. Fluid and Ullage Maintenance Design for 10-MW Pilot Plant

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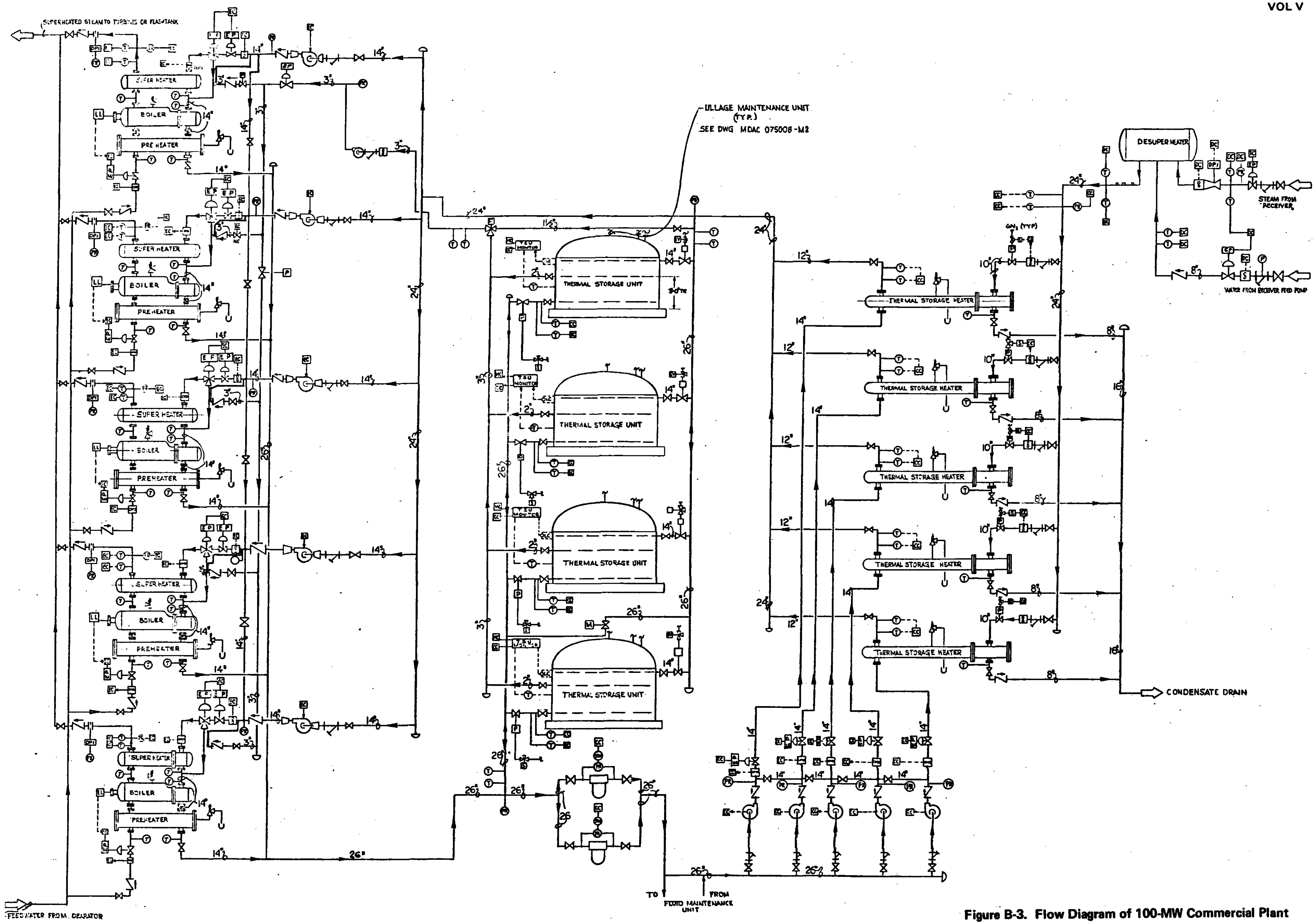

Figure B-3. Flow Diagram of 100-MW Commercial Plant

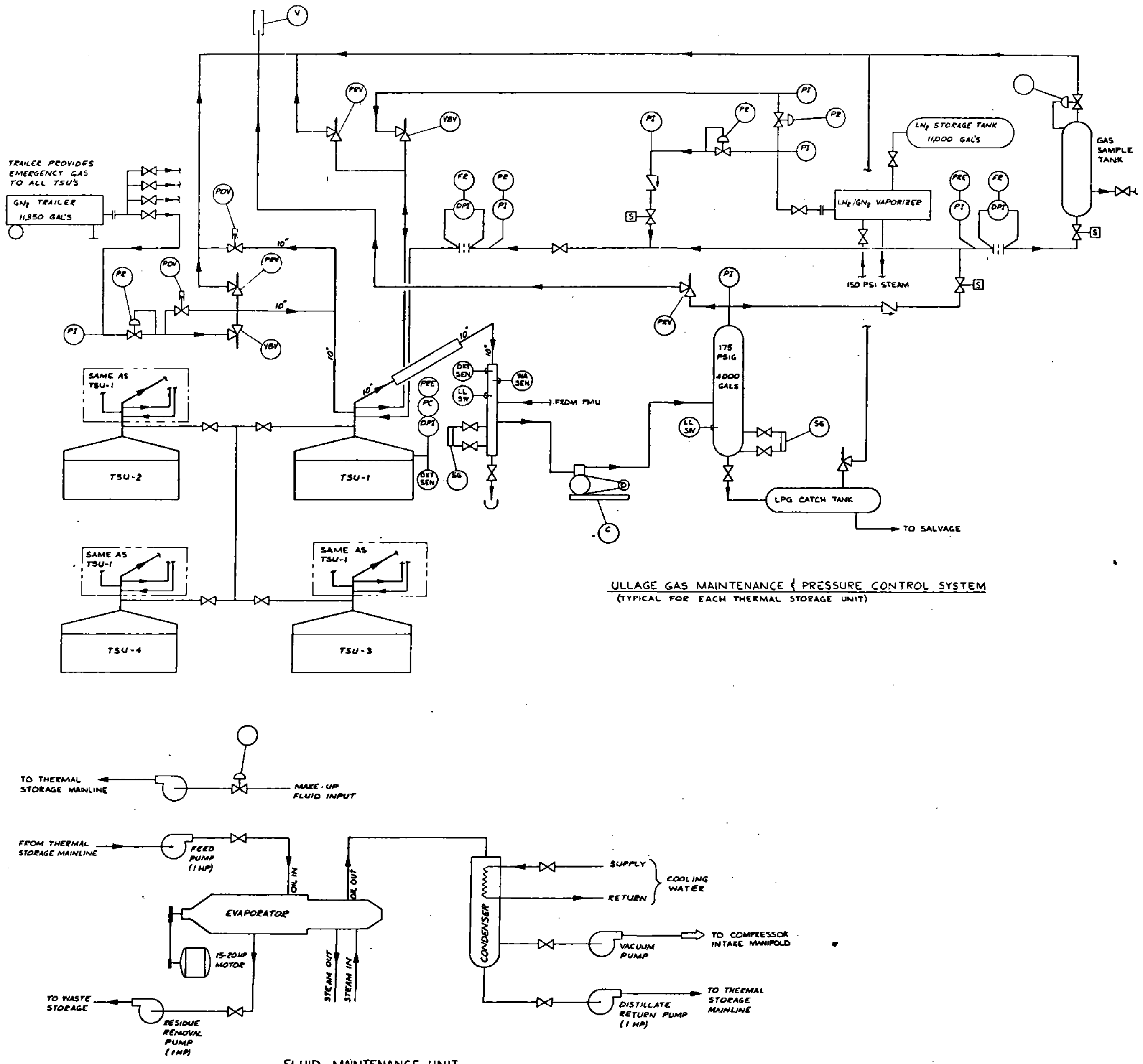

Figure B-4. Fluid and Ullage Maintenance Design for 100-MW Commercial Plant