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CONCEPTUAL DESIGN OF THERMAL ENERGY STORAGE SYSTEMS FOR NEAR TERM ELECTRIC UTILITY APPLICATIONS

VOLUME ONE: SCREENING OF CONCEPTS

W. Hausz, B.J. Berkowitz, and R.C. Hare
General Electric Company—TEMPO

October 1978

Prepared for

National Aeronautics and Space Administration
Lewis Research Center
Under Contract DEN 3-12

for

U.S. DEPARTMENT OF ENERGY
Office of Energy Technology
Division of Energy Storage Systems

and

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Washington, D.C. 20545
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Palo Alto, California 94303

FOREWORD

The work reported was performed by technical staff personnel of the following General Electric components: Energy Systems Technology Division, Large Steam Turbine Division, Corporate Research and Development, and TEMPO (Center for Advanced Studies). Overall project direction was provided by Eldon W. Hall, Energy Technology Operation, ESTD. A number of highly qualified consultants, both within and outside General Electric, assisted in their area of expertise. These, too numerous to list here, included many of the proponents of specific concepts and the authors of references or source documents who freely supplied additional information upon request. Some of these, as appropriate, are named in the text.

One of the in-depth reviews of Task I preliminary results was by a Review Panel consisting of General Electric managers and representatives of electric utilities and an architect-engineering firm.

SUMMARY

The project objective is to examine the field of proposed concepts for thermal energy storage systems (TESS) and select, conceptually design, and analyze the most promising for near-term electric utility applications. This report describes the task concerned with selecting up to three promising concepts for more detailed design and analysis.

Over forty TESS concepts gleaned from the literature and personal contacts were examined for possible application to two reference plants, an 800 MW_e high-sulfur coal plant, and an 1140 MW_e light water nuclear reactor. A preliminary screening on near-term availability and applicability reduced the set to twelve selections, some of which combined the elements of several concepts.

Modifications to the plants favorable to TESS were incorporated in a thermodynamic computer model, which considered the operation of the Turbine Island (turbine generator and associated parts of the plant) under normal conditions, under storage charging conditions, and under storage discharging conditions for the case in which a peaking Turbine Island provided fractional increments of power up to 50 percent of reference plant rated power, and for the case of feedwater heat storage with an enlarged main turbine. The program permitted defining size and performance requirements on the TESS components and the system turnaround efficiency, ie the ratio of peaking electric energy produced to the electric energy reduction during storage charging. Sensitivity analysis for the principal parameters was performed.

Storage media included in the twelve selections included high temperature water (HTW), hot oil, molten salts, and packed beds of solids such as rock. Of these the HTW required high pressure containment; steel vessels, prestressed cast-iron vessels, prestressed concrete pressure vessels, excavated underground caverns, and natural aquifers were considered.

The economic or costing methodology was based on the recommended values in the EPRI *Technical Assessment Guide* which represents customary electric utility planning practice. Capital costs of plants are expressed in total plant costs or in dollars per kilowatt ($/kW) on a TOTAL cost level which includes interest during construction, spare parts, contingencies, overhead, and other elements not normally included in installed or direct costs of equipment. Variable annual costs such as fuel and associated O&M are levelized, ie converted to a uniform

annual payment equivalent to the assumed fuel escalation scenario over the plant life. These two practices lead to higher \$/kW and cost of electricity values than found in many studies.

Capital costs for the TESS and associated components such as the peaking Turbine Island were derived for the twelve selections in specific TOTAL costs (\$/kW) using the same costing basis as used for reference plants. Power-related and energy-related components of cost were expressed separately as well as the sum.

The ranking in cost was compared with subjective rating considerations in technical risk or near-term availability, suitability for utility applications, conservation potential, growth potential, hazards and environmental problems, and diversity of approach to make recommended choices among the twelve selections.

The recommended choices as approved by DOE/EPRI/NASA for further study and conceptual design in the remaining project tasks were:

- An underground cavity in hard rock with steel liner and concrete between liner and rock. HTW is contained. High pressure steam is injected into the water for storage charging. Lower pressure steam is withdrawn for peaking turbine output.
- Storage is in tanks packed with solid particles, eg rock. The voids between particles contain a heat transfer fluid, eg oil or molten salt which passes through heat exchangers to charge storage with energy from condensing steam and to discharge storage by producing lower pressure steam for a peaking turbine. These two choices are applied to the 800 MW_e HSC plant.
- Prestressed cast-iron vessels (PCIV) are used as containment for HTW. The feedwater mode of storage is used in which excess hot feedwater is stored during off-peak hours, reducing the feedwater heating needs during peaking.
- The dual-media concept of the second choice above (oil or salt with packed beds) is used in the feedwater storage mode. The third and fourth choices are applied to the 1140 MW_e LWR plant.

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SECTION 1
INTRODUCTION

This report describes work done by the General Electric Company between December 1977 and May 1978 on projects sponsored by the Department of Energy, *Conceptual Design of Thermal Energy Storage Systems for Near Term Electric Utility Applications* (NASA-Lewis Research Center contract DEN3-12), and by the Electric Power Research Institute, Inc., *Comparative Analysis of Utility Sensible Heat Storage Systems* (EPRI contract RP1082-1). This report is the required output of a systems selection task that identifies the thermal energy storage system (TESS) concepts, of the many considered, that warrant more detailed conceptual design and economic analysis in the remaining tasks of the project.

BACKGROUND

There is a need in electric utility operation for an economic means of supplying the varying demand for electric power. While there are seasonal and weekly demand patterns, the daily load pattern is of primary concern in this project. Load-following with conventional base load generating capacity may not be the most economic way since its high capital cost and low fuel cost per unit of energy delivered favor continuous operation over all available hours (ie not unavailable because of forced outages or scheduled maintenance outages).

Two alternatives for meeting peak load demands are the use of gas turbines and the use of energy storage. The former has a low capital cost per kilowatt of capacity but uses petroleum, a fuel that is more costly than coal or nuclear fuel, and the use of which is to be minimized by utilities as a national policy. Energy storage has long been used in pumped-hydro form where off-peak power moves water from a lower to an upper reservoir, and electricity is generated during peak-demand hours as the water returns to the lower reservoir through a hydraulic turbine.

The final reports prepared by the Public Service Electric and Gas Company of New Jersey (PSE&G), *An Assessment of Energy Storage Systems Suitable for Use by Electric Utilities,* EPRI EM-264 Project 225, ERDA E(11-1)-2501, identified and compared a number of energy storage concepts including above- and below-ground pumped hydro, compressed air storage, thermal energy storage, battery storage, and flywheel storage. Thermal energy storage was identified as a potentially viable contender because of its technical and economic features and potential for early commercialization.

The stated objective of this project is to confirm the apparent attractiveness of thermal energy storage and, if confirmed, to select and conceptually design the most promising systems for near-term utility applications.

SCOPE

To accomplish the project objective, the study scope was defined to examine the widest range of thermal energy storage concepts and to perform comparative, design, and economic analyses as specified by the following tasks:

- System Selection: A large number of thermal energy storage systems (TESS) concepts are identified and defined by searching the literature, consultation with industry, universities, and government agencies, and by combination or innovation. A methodology for comparative evaluation is used for two successive screenings: first, the ensemble of concepts is reduced to a maximum of 12; then, on approval of the results of the preliminary screening, a more detailed comparison selects a maximum of three systems for conceptual design.

- Conceptual System Design: For each of the selected systems a detailed conceptual design is prepared, after redefining and optimizing the parameters of the TESS and the baseline plant designs.

- Benefit Analysis: The characteristics of the conceptual system designs and their value to the industry are evaluated, and economic and benefit analyses conducted. The market potential, a practical scale implementation, and related conservation benefits are estimated.

- Development and Demonstration Program Recommendation: A recommended program for the development and near-term power plant demonstration of the designed systems are outlined.

The project considers TESS concepts under criteria of particular interest to utilities and evaluates TESS operations in a utility system context (rather than a single plant context) by hour-by-hour simulation to determine production costs over an annual cycle. Summary plant booklets describing the three systems carried through conceptual design are also required.

In the context of the above description of the task sequence, this report confines itself to the description of the methodology and results of the system selection task.

PROCEDURES AND CONSTRAINTS

Thermal energy storage differs from other storage forms for electric utility applications as shown on the right of Figure 1-1. In conventional generating capacity, there is a flow of energy from fuel to the load or consumer, with conversions in form in the boiler and in the turbogenerator. The furnace and boiler convert the chemical energy to high pressure steam; the turbogenerator converts it to electrical energy, which is transmitted and delivered to the load. Other storage forms are charged by extracting the energy as electricity. For TESS as shown, energy is extracted as steam, between boiler and turbogenerator.

The daily and weekly energy demand cycle of a "typical" electric utility over a summer week* is shown at the left of Figure 1-1. It

*Shown is Synthetic Utility D from EPRI Report EM-285, *Synthetic Electric Utility Systems for Evaluating Advanced Technologies* Reference 229).

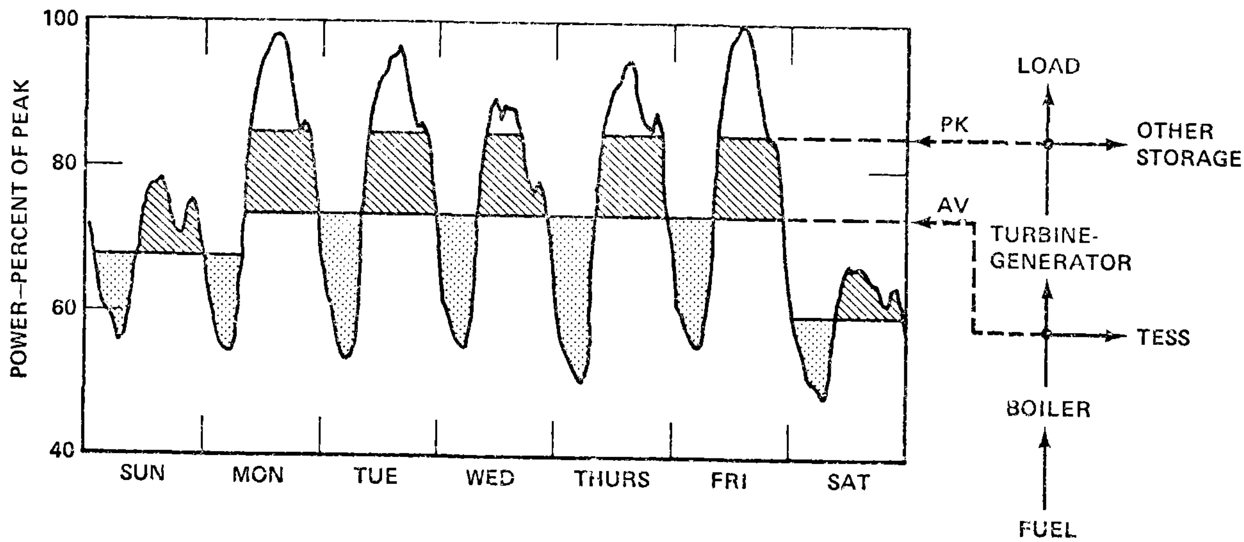


Figure 1-1. TESS and the weekly load curve.

varies from 100 percent down to 50 percent of the peak load. Base load generating plants, with lowest fuel costs, operate as many hours as they are available, supplying the full time load below 40 to 50 percent of peak demand. If base load capacity is increased to be larger than 50 percent of the peak load there would be unused capability during the troughs. By storing base load thermal energy during the troughs, to produce electricity for use in peak hours, fuller use can be made of efficient base load capacity, even if it is increased to 70 percent of peak demand.

With thermal storage the boiler can operate at a constant power level corresponding to the average power output of the base load plant. Turbine generator capacity must be provided to handle the peak-hour demands. The energy charged into TESS during off-peak hours is shown shaded. The energy converted to electricity during peak hours is shown crosshatched. There is a relationship between the two areas called the turnaround efficiency: the electric energy output from storage (cross-hatched area) divided by the electric energy not generated in order to charge the storage (shaded area). The fuel costs of any system are inversely proportional to efficiency, so high turnaround efficiency may

may be necessary to compete with other generation alternatives available during peak hours.

Constraints

In performing the system selection task, constraints on the scope are imposed by the work statement in order to focus attention on near-term commercialization by electric utilities.

The application of TESS is confined to new plants, planned and designed to incorporate the system. The new plants considered are conventional coal and nuclear fueled, which represent the large majority of expected electric utility capacity additions between now and AD 2000. As nuclear plants, only light water reactors (LWR) are considered; as coal-fired plants, only conventional types with flue gas desulfurization (FGD) when high-sulfur coal is to be burned are considered. All plants employ a steam driven turbogenerator for conversion to electricity and a fired boiler or nuclear reactor as a steam supply.

The requirement for near-term availability requires interpretation since the planning and construction cycle for large conventional plants is eight to twelve years. Concepts to be considered must be capable of demonstration before 1985 so that manufacturers can offer to supply, and utilities can plan and order with confidence over all or most of the period 1985-2000. By this criterion, penetration of the market will be small until the latter part of the period.

It is recognized that stringent electric utility requirements must be met, to match the standards set for performance and required of conventional plants. These include high reliability, flexibility and stability of operation, meeting environmental standards for emissions and for hazards to life and property, and low maintenance requirements. Not least of utility requirements is that a TESS plant be economic compared to the generating capacity alternatives available to the utility for comparable duty.

Another "constraint" is that the ensemble of concepts considered be comprehensive. Many concepts have been suggested, analyzed, or tested by many proponents both in the United States and abroad. In the screening process all those identified are to be considered.

Methodology

The sequence of subtasks for Task I, System Selection, is shown in the flow chart, Figure 1-2. Initial effort was on data gathering and structuring, performed in parallel as Taxonomy and Literature Survey. From the data gathered, a comprehensive listing and description of relevant concepts was derived and a preliminary screening performed on the basis of near-term availability, comparative economic viability, and suitability for utility operation.

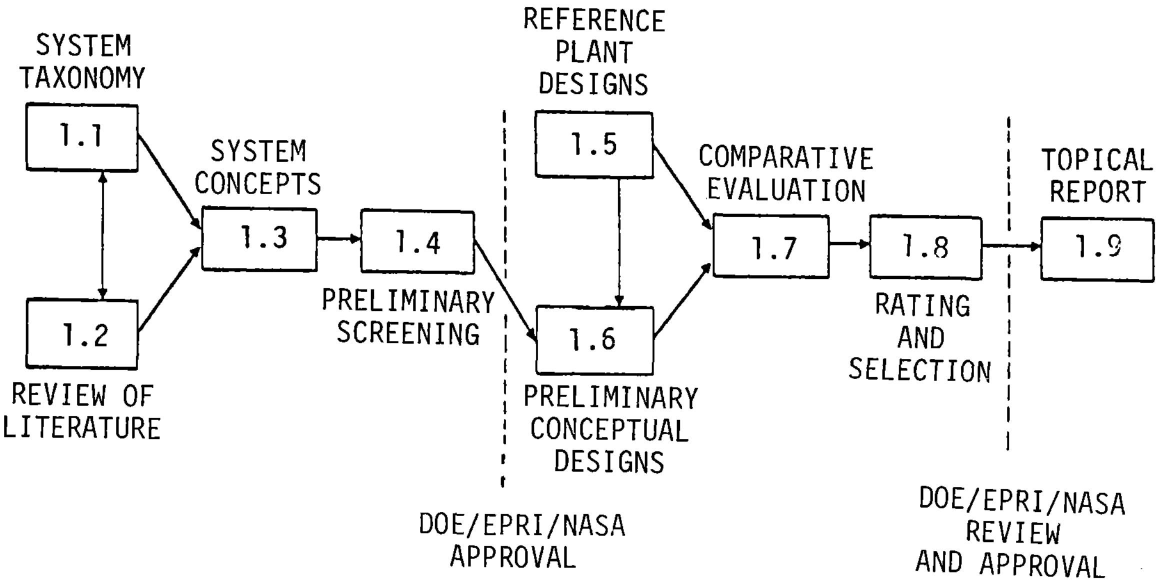


Figure 1-2. Flow diagram of Task 1.

In the second half of Task I, reference plants were selected and the problems of integrating the selected concepts with a conventional plant were addressed. The thermodynamic performance of the reference plants modified for TESS inclusion, and for the TESS systems, was computer modeled for comparative evaluation. Costs of storage materials, containment, other TESS components, and of the power conversion components of the reference plants were derived for economic comparisons.

Consultation with electric utilities and manufacturers of conventional plant components, TESS containment, and storage media provided information on other criteria for evaluation. The rating process in the comparative evaluation resulted in the selection of two concepts as best meeting all major criteria, with several others suggested as alternatives.

Subtask 1.9 is the preparation of this document, the Topical Report.

PLAN OF THIS VOLUME

The sections of this report follow the subtask pattern shown in Figure 1-2, with some combinations. An extensive data base was assembled for the preliminary screening. Much of this is presented in detail as Appendices A, B, and C, and is more briefly described in Sections 2 and 3. Sections 4 through 8 describe the procedure and results of the comparative evaluation, narrowing the concepts from twelve down to those recommended for approval, as presented to personnel of DOE/EPRI/NASA on May 22, 1978.

SECTION 2
IDENTIFICATION OF CONCEPTS

LITERATURE SEARCH

The initial source of literature references was recent project reports of ERDA, DOE, NASA, and EPRI that were relevant to thermal energy storage. Each of these, in its reference lists, provided additional sources that were obtained. Consultation with government agency program managers, industry project managers, and consultants provided additional sources.

A computer search was made, with relevant key-word combinations. The following data bases were searched from years as early as 1964 up to 1977: Science Abstracts, Energyline, Compendex (Engineering Index), NTIS, Nuclear Science Abstracts, ERDA Energy Data Base. The printout of abstracts from the selected key-word combinations were scanned, and about thirty-five references not previously identified were ordered.

The bibliography or literature references list continued to grow during the course of the project as information on particular materials, technologies, methodology, or concepts became of interest. Listed in Appendix A in Volume 2 are the 237 entries to date.

Each entry is assigned a number for ease in referencing in the concept descriptions (Appendix C) and in the text of this report. The numbers were assigned chronologically as references were received, but it will be noted that those in hand by the beginning of January 1978 were arranged in alphabetical order. For ready cross referencing, Appendix A contains a list of reference number versus author, and a full bibliographic reference list alphabetical by author or institutional source.

A limited cross reference by principal subjects is also provided, by number and author, for the convenience of the reader.

TAXONOMY

Taxonomy, the science or technique of identification, naming, and classification or ordering of a data base, is a useful method of structuring the many thermal energy storage systems that have been proposed, so that their common elements and their differences can be recognized. The basic structure connecting these elements is illustrated in Figure 2-1.

All of the thermal energy storage systems identified have one or more storage media, a form of containment for the storage media, a fluid for heat transfer and heat transport, a source of heat derived from the reference power plant, and a means for conversion of the stored thermal energy into electricity.

Major classifications are given within each box. A more extensive, numbered taxonomy was prepared to use in classification of the many concepts being collected from the literature. It aimed at being comprehensive, considering all possibilities. Many of the categories defined were found to be empty: no proposed concept used them, nor were they considered sufficiently attractive to warrant creation of a concept. This taxonomy may be found in Appendix B, and is used in Appendix C in characterizing concept definitions.

A summary of the alternative components can be presented as shown in Figure 2-1. For utility applications, the only thermal energy sources relevant to this project are steam and hot boiler feedwater. Some concepts identified from the literature used as sources hot gases: helium from gas-cooled reactors, or solar thermal towers; hot sulfur trioxide from solar towers; hot air from compressed air storage systems. Other components of these systems: containment, storage media, reconversion to electricity, were considered but non-steam-cycle thermal sources were discarded.

Sources

The steam source used can be at various pressures and temperatures. Live steam, the high pressure output from a coal-fired boiler, may have

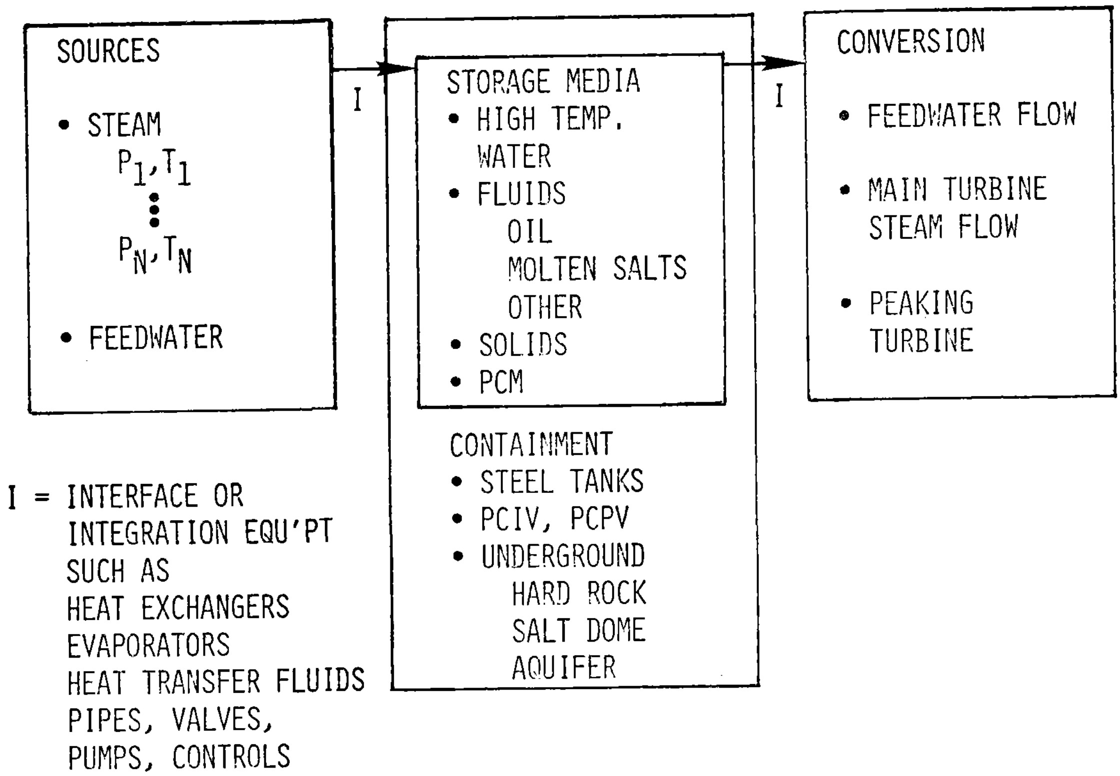


Figure 2-1. Basic taxonomy structure of thermal energy storage systems.

a pressure from 16 to 24 MPa (2400-3500 psig), at 540°C (1000°F). After passing through the high pressure turbine the pressure may be 4.8 MPa (700 psi) at 305°C (585°F). This is often called cold-reheat steam (CRH); after passing through the reheater tubes of the boiler, it again has a temperature of 540°C at a slightly reduced pressure and is called hot-reheat steam (HRH). From a LWR the steam pressure is 6.8 MPa (1000 psi) at 280°C (540°F).

Another possible source of steam is between the intermediate pressure (IP) turbine and the low pressure turbines. This point is called the crossover; the steam conditions here are 1.1 to 1.2 MPa (160-180 psi) at about 360°C (690°F) for the coal-fired plant, or 280°C for the LWR.

In addition, there are extraction points in the turbine generator sets for six or seven feedwater heaters, which permit limited withdrawal of steam at intermediate temperatures and pressures.

The condensate flow from the condenser is heated by the feedwater heaters (FWH) to successively higher temperatures, so in principle feedwater may be extracted, inserted, or stored at any of the temperatures between FWHs. After the highest temperature FWH, at the boiler inlet, feedwater temperatures are 215-225°C (420-440°F) for LWRs and up to 265°C (510°F) for fossil-fired plants.

Storage Media

The lowest cost storage medium is water. Even water purified to boiler feedwater quality has a cost of much less than $1 per Mg (90¢/ton). High temperature water (HTW), of adequate quality, also has the advantage of being usable directly in the boiler/turbogenerator cycle, without such interface equipment as heat exchangers. HTW has the disadvantage of requiring high pressure containment for temperatures much above 100°C (212°F). All the other common storage media considered can be stored at close to atmospheric pressure.

The penalty in cost of containment of HTW can be indicated by the temperature/pressure relationship of saturated water shown in Figure 2-2. The saturation pressure is roughly an exponential function of temperature as indicated by the curve fitting equation in the figure. Since the stored energy in HTW increases only linearly with temperature, storage as HTW is limited in maximum usable temperature unless very low cost pressure containment is available.

Alternatives to HTW as a storage medium are organic compounds such as aliphatic or aromatic petroleum compounds, and derivatives that may also contain chlorine, fluorine, silicon, or oxygen. Many of the major oil companies have trademarked lines of heat transfer fluids with the maximum temperature for operation with acceptable degradation rates varying from 310°C (600°F) for relatively low cost

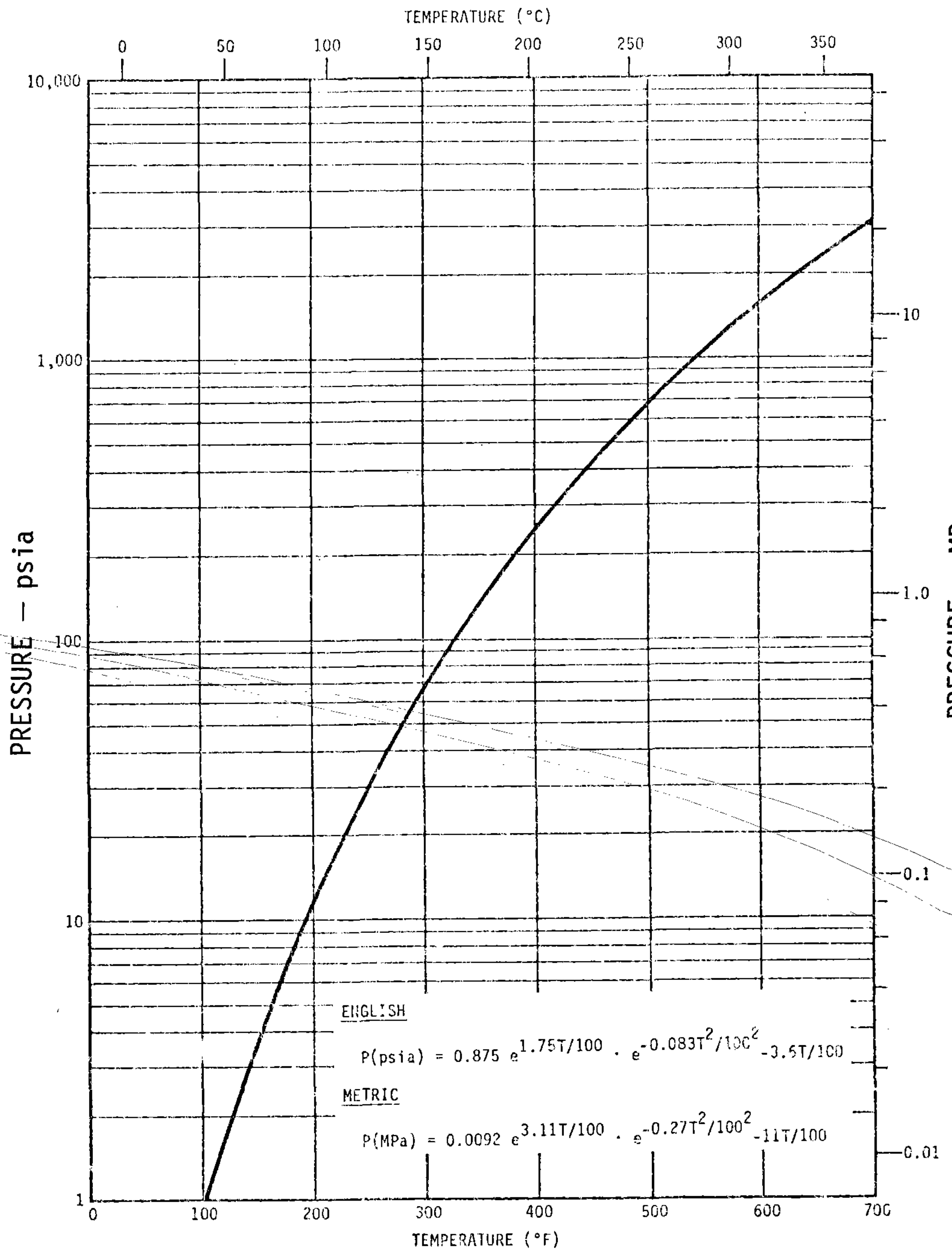


Figure 2-2. The pressure versus temperature relationship for saturated water.

media to as high as 400°C (750°F). Many of these fluids are low viscosity liquids, pumpable down to ambient temperature.

Also mixtures of inorganic salts are available whose melting point is below the lowest temperature in the range over which the storage medium is to be cycled, and is liquid and stable (low degradation rate) to very high temperatures. One example used in a number of the concepts proposed is the eutectic of sodium and potassium nitrates and nitrites (0.07 $NaNO_3$, 0.53 KNO_3, 0.40 $NaNO_2$). This salt has a melting point of 148°C (288°F) and has been used in industrial processes for over 20 years as a heat transfer fluid and as a quenching and annealing bath at temperatures up to 500°C with low degradation rates. It is offered by different companies by tradenames such as HITEC (duPont) and PARTHERM 290 (Park Chemical). Other salts are available with lower or higher melting points and with higher upper temperature limits and with lower cost materials. Selection must consider all the requirements.

Less expensive than the oils and molten salts are various solid materials. These range from crushed granite or other rock, through river-bed gravel, sand, pellets of sintered iron oxides such as taconite pebbles and Feolite, to ceramic spheres or bricks, cast iron balls and scrap steel. These can be used in stationary packed beds, with a heat transfer fluid passing through the bed for direct contact heat exchange to charge and discharge the bed. As the heat transfer fluid may be present in significant quantities to fill the voids in the packed beds, such a system concept is called a dual-media storage system. If the fluid and the solid are compatible at high temperatures, the lower cost of the solid can reduce the overall cost of storage.

Other sensible storage media suggested include molten metals and alloys, such as sodium, NaK (eutectic of sodium and potassium), lead, etc. Two of the industrial chemicals with the lowest cost in reasonably pure form are sulfur and sulfuric acid. Both are liquid in the temperature range of interest for thermal storage for utility applications. Sulfur has been proposed for utility applications and sulfuric acid for another application.

Another large class of storage media are phase change materials (PCM). These are materials which melt and freeze at a particular temperature of interest and have a large latent heat of fusion and crystallization. They have the advantage over sensible heat storage of a higher energy density of storage per degree of temperature change over the limited temperature range surrounding the fusion point.

Each of the above forms of storage media has good features and bad features, advantages and disadvantages. The weighing of these in the context of concepts is an important part of the preliminary screening.

Containment

For sensible heat storage in solids (eg packed beds of rock) and heat transfer liquids (eg oils and molten salts) at atmospheric pressure, steel tanks are adequate. Very large storage volumes are required so multiple tanks in modular sizes can be selected for cost and convenience. The American Petroleum Institute (API) provides specifications on a range of modular sizes suitable for estimating in preliminary conceptual designs. They are cylindrical with a height under 15 m (50 ft) and diameters from 6 m to 90 m (20-300 ft).

For pressure containment above one megapascal (1 MPa or 145 psi) the wall thickness of steel required in steel tanks increases proportionally with pressure and with diameter, so the thickness becomes excessive for welding and inspection at very high pressures and volumes. For assurance against reduced life and catastrophic failures, boilers and pressure vessels must comply with very detailed ASME codes. Modular sizes, small enough for rail transport which permit factory assembly, welding, test, and inspection, and with wall thicknesses under 0.15 m (6 inches) are often more cost effective than field assembled larger tanks. Because special steels, often in short supply are required by the codes, the costs and delivery times for steel pressure vessels encourage consideration of alternatives.

Prestressed concrete technology is over thirty-five years old. High tensile strength steel cables and "tendons" are incorporated in concrete beams and structures for bridges and buildings, and pretensioned to place all parts of the concrete in compression under all load conditions. Application of the technology to pressure vessel containment for nuclear reactors is roughly ten years old, but has undergone rapid development. None have as yet been built for pressures and temperatures that would be typical for thermal energy storage systems (eg 4-6 MPa, 260°C). Prestressed concrete pressure vessels (PCPV) would be almost completely field fabricated. For the nuclear reactor application ASME code specifications have been formulated, but not for the temperatures and pressures of interest.

A more recent concept is the prestressed cast iron pressure vessel (PCIV), conceived and under development by Siempelkamp Giesserei GmbH (Federal Republic of Germany). The concept uses factory-cast cast-iron arcs, six to a full circle, which can be quickly field-assembled into multiple cylindrical layers using key ways. External cable wrapping and vertical tendons are used to prestress the cast-iron to assure it is in compression. To contain boiler-quality feedwater or HTW a thin alloy steel liner would be welded in direct contact with the cast iron. An external thermal insulation is proposed.

While a small PCIV has been built, and conceptual design studies of the application of PCIV to HTW thermal storage have been done jointly by Professor P.V. Gilli of the University of Graz, Austria, and Siempelkamp, no full scale models for high pressure and temperature have been built.

An alternative to pressurized containment above ground is underground containment at depths where the overburden or hydrostatic pressure is compatible with the storage pressures required. Natural caverns, excavated caverns, solution mined caverns in salt domes, and aquifer storage have been proposed. Natural caverns with a depth, volume, and location suited to plant siting would be a rarity. Hard rock that is stable and competent and at suitable depths can be found in many parts of the United States.

To contain HTW in a hard rock cavern, without loss or contamination, requires a thin liner and means to transfer the pressure stresses from the HTW to the rock without danger of rupturing the liner. One means proposed is a poured layer of high temperature, high strength concrete between the liner and the rock. This permits heat conduction into the rock, with a significant steady state temperature gradient extending for many cavern diameters. For large caverns the annual fractional heat loss is low. An alternative to concrete stress transfer is the use of a free standing liner surrounded by compressed air that is in equilibrium with the HTW pressure. This permits insulation external to the liner that can reduce heat losses, and limit the temperature rise in the rock by continued cooling of the compressed air.

Salt domes and salt beds can be solution mined to form cavities at a lower cost per unit volume than hard rock excavation. However, suitable formations are very limited geographically, and no means of installing a liner to contain high quality water has been suggested. Storage of hot brine or hot oils in direct contact with the salt may require no liner but associated problems may be difficult to solve.

Confined aquifers, water laden porous layers contained above and below by impermeable layers, are common in sedimentary geographic areas which encompass much of the United States. Hot water can be injected and recovered, but of groundwater quality, not of boiler feedwater quality, so aboveground heat exchangers would be required. It is not currently known how high a temperature of injected water can be used without solution, precipitation, and other changes in the minerals of the aquifer over a reasonable life.

Conversion

The major conversion of interest is from expanding steam to electric energy. In some cases there are several intermediate conversions between the stored energy and the conversion to electric energy, eg, conversion from water to steam in evaporators or heat exchangers from a heat transfer fluid to boiling water.

The two major variants on the conversion of steam to electric energy are the use of an oversized version of the turbine generator which has been designed for base load plus peaking load flow rates, and the use of a separate peaking turbine for the increased capacity, leaving the main turbine essentially unchanged in size. In the latter case, the peaking turbine is designed for inlet steam at the temperature and pressure at which it can be derived from storage.

In the former case, steam derived from storage can only be inserted between turbine casings, ie between the high pressure (HP) turbine and the intermediate pressure (IP) turbine or between the IP and low pressure (LP) turbines. Since the process of storage degrades the quality of the steam available, the point of injection is at a lower pressure level than the source thermal energy.

With the oversized main turbine, another option is to pass a larger steam flow through the IP and LP turbines than normal by reducing the multiple steam extractions used to heat the condensate from the low temperature at the condenser output to the desired boiler inlet temperature. Manipulation of the water flow through the feedwater heaters (FWH) is known as feedwater storage. To charge storage, a greater steam extraction than normal is used to heat either additional HTW or another heat transfer fluid, which transfers the energy to storage. More steam extraction reduces the power output of the turbine. For peak output, steam extraction is reduced, increased power is derived from the greater steam flow, and needed additional energy for feedwater heating is discharged from storage. Combinations of deriving steam from storage and manipulating the FWH steam extraction are sometimes used in concepts.

The conversion of the stored thermal energy in pressurized HTW to steam may be done in several ways which are illustrated here because references to the terminology will occur repeatedly. In utility and industrial parlance, a pressure vessel containing HTW for steam generation is called a steam accumulator or just "accumulator."

VARIABLE PRESSURE ACCUMULATOR. The variable pressure mode of operation is shown in Figure 2-3. When fully charged, almost all the volume is filled with saturated HTW, with a small "cushion" of saturated steam (at the same temperature and pressure) above it. In this mode steam is drawn from the top; as the pressure in the steam cushion decreases, some of the water in the vessel will flash to steam. All evaporation or steam generation is internal to the vessel. As flashing to steam is continued the water will decrease in temperature, the saturation pressure will decrease and the water level will move downward by the amount of water converted to steam. If the useful range of temperature and pressure is limited, only a small fraction (15-25 percent) of the HTW volume may be flashed to steam. The remaining volume of water acts as a reservoir in which to store the thermal energy to produce steam. To recharge the accumulator, steam is injected. While, in discharging, flashing to steam occurs throughout the water volume and provides good mixing, during charging the water must be mixed with the steam to assure that the entire tank becomes heated and colder denser strata do not remain at the bottom and reduce the energy storage capacity.

EXPANSION ACCUMULATOR. This mode of operation is shown in Figure 2-4. When fully charged, the accumulator is almost full of HTW with a small steam cushion, as in the variable pressure mode. As hot water is drawn from the bottom during discharge, enough of the contained HTW flashes to steam to fill the tank volume. As indicated in the figure, this flashing reduces the pressure and temperature of the saturated water and steam slightly, but not nearly as sharply as in Figure 2-3. All of the water can be removed with a reduction in pressure of only about 30 percent. Alternatively, if it is thermodynamically valuable to keep the pressure and temperature uniform during discharge, a small amount of saturated steam from the source may be injected at the top as water is removed from the bottom.

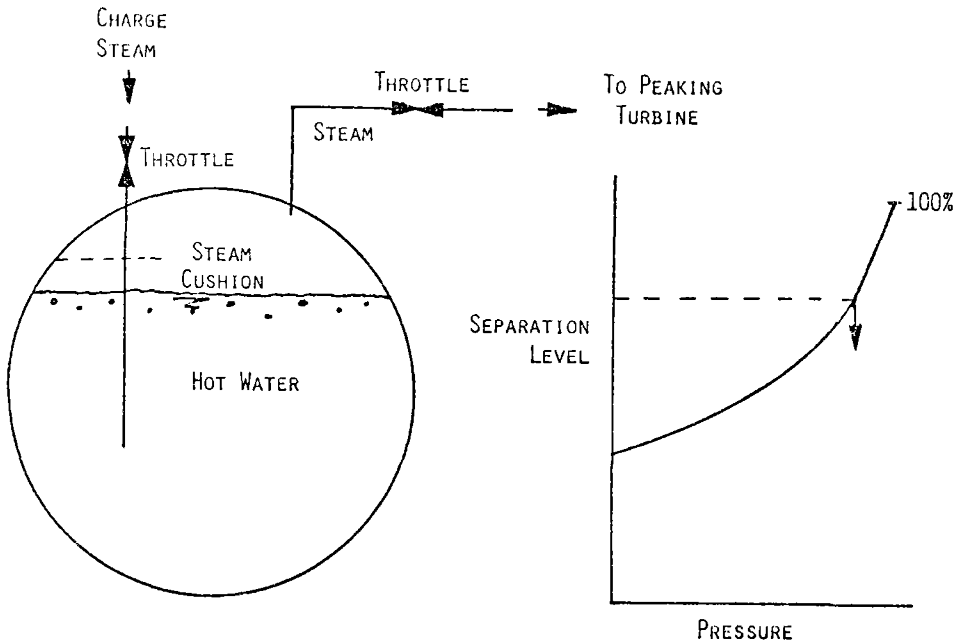


Figure 2-3. Variable pressure accumulator.

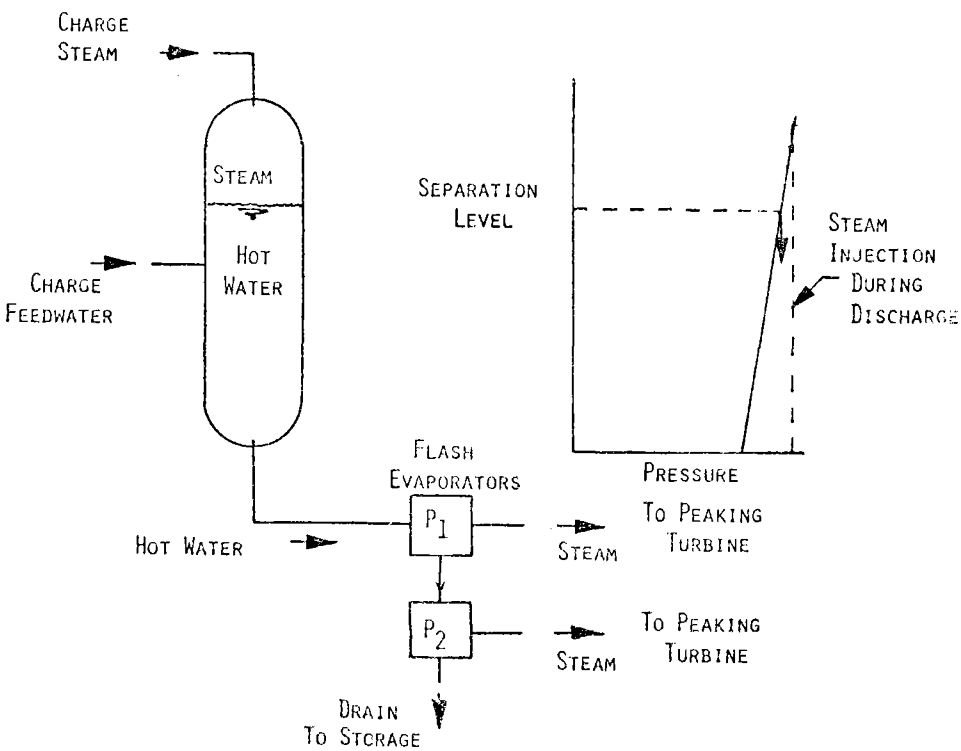


Figure 2-4. Expansion accumulator with flash evaporator.

The HTW removed must be flashed to steam in evaporators external to the expansion accumulator, as shown in Figure 2-4. The water is throttled to a pressure P_1 lower than the storage pressure, and the resulting steam and water are separated in a drum. The steam is dispatched to a turbine. The water may be throttled to a still lower pressure P_2 for generation of more steam at this pressure. This can be dispatched to a separate inlet on the same turbine or a separate peaking turbine. Additional stages of flash evaporators may be used similarly.

During discharge the water drained from the last flash evaporator must be collected and stored. Its volume will be more than half of the initial volume of HTW but it is at a low pressure and temperature so this "cold storage" is not costly. The variable pressure accumulator also required cold storage, but of a much smaller volume corresponding to just the volume of water flashed to steam.

To recharge the expansion accumulator requires simultaneous injection of hot water and saturated steam, until the whole volume except for the small steam cushion is refilled with saturated water at the desired pressure and temperature.

DISPLACEMENT ACCUMULATOR. In a third mode of use an accumulator is always completely filled with water. When fully charged with thermal energy, it is filled with HTW at the desired temperature; when fully discharged, the water contained is all cold. As shown in Figure 2-5, hot water is injected at the top during charge and removed from the top during discharge. Cold water leaves and enters at the bottom. Since hot water is less dense than cold, it will float at the top. A fairly sharp temperature gradient called the thermocline separates the hot and cold water. It remains stable and sharp if mixing currents are avoided, and is ultimately limited by the thermal conductivity of water.

During discharge, one or more flash evaporators are used to generate steam for the peaking turbine(s). The drain from the evaporators and the condensate from the turbines is returned to the vessel as cold water, so the large cold-storage described for the expansion mode is

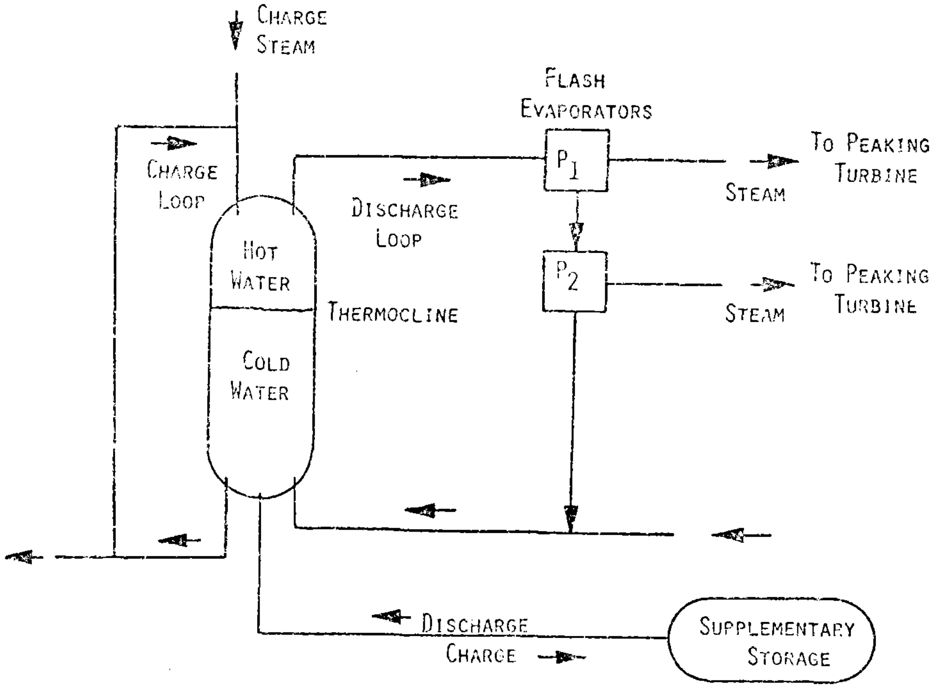


Figure 2-5. Displacement accumulator with flash evaporators.

not required. However, since hot water and cold water differ in density a small supplementary storage is needed for the net change in volume.

During charge, steam is mixed with cold water taken from the bottom of the tank to raise the water to the desired temperature. Cold water equivalent in mass to the steam is returned to the boiler inlet feedwater to generate more steam.

HEAT EXCHANGERS. When the storage medium is not HTW, the stored thermal energy must be transferred to water before conversion to steam can take place. This requires a heat exchanger. While direct contact heat exchangers are possible, in which the storage medium or input heat transfer fluid is in direct physical contact with the output heat transfer fluid, eg HTW, the water quality requirements for boiler and turbine operation make physical separation of the two fluids necessary. An example of the heat exchanger complement required when an atmospheric pressure sensible heat storage system is used is shown in Figure 2-6.

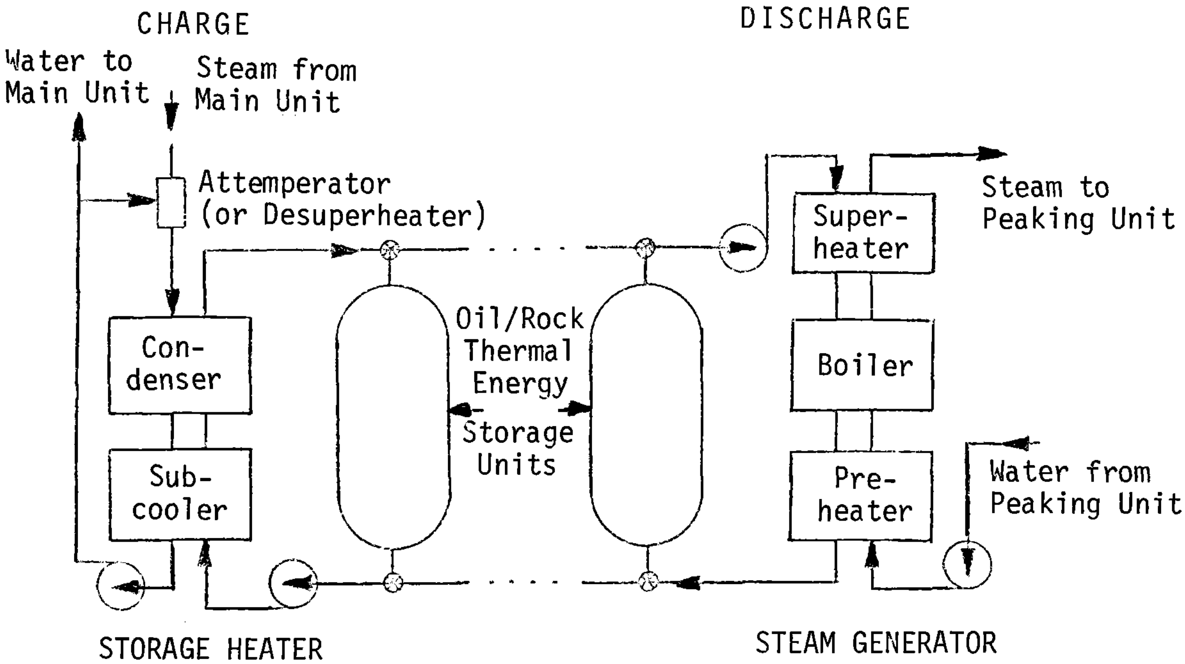


Figure 2-6. Heat exchangers for a sensible heat storage system.

Steam from the heat source chosen can go through three specialized heat exchangers in cascade. The entering steam may be superheated, ie at a temperature considerably higher than the saturation temperature for its pressure. The first heat exchanger or desuperheater removes the superheat producing saturated steam. The condenser then removes the latent heat of vaporization at constant temperature. The condensate water at saturation temperature may be subcooled in a third heat exchanger (HX) to further increase the thermal energy stored, and to match the temperature at which the output water is to be reintroduced into the source cycle.

In general tube-in-shell heat exchangers are used, in which one fluid is contained inside a bank of parallel closely spaced tubes, and the other fluid is exterior to the tubes but inside a containing shell. The heat transfer fluid to storage may be more likely to "foul" the heat exchange surface, ie produce deposited layers which impede heat transfer, than is the steam or HTW. The inner surface of the tubes is easier to clean than the outer surface, by means of access through the tube headers. This disposes designers toward using the interior of the tube for the storage heat transfer fluid. On the other hand, containing the high pressure steam and HTW in the shell requires a much thicker shell.

The different design problems for each temperature range makes it convenient to have the three heat exchangers physically separate. While the desuperheater can be designed as a shell-and-tube HX, a simpler, less expensive alternative is to spray just enough water into the superheated steam to remove the superheat. This is called an attemperator, and is shown in Figure 2-6.

On discharge of the storage, water (condensate) from the peaking turbine is heated successively in a preheater (to raise it to saturation temperature), in a boiler (to add latent heat at constant temperature to convert it to steam), and a superheater (to increase the steam temperature above saturation to the extent made possible by the maximum temperature available in storage).

The storage unit shown comprises multiple packed rock beds with hot oil as part of a dual-media system and as the heat transfer fluid. The storage tanks operate in the displacement mode with a thermocline separating hot and cold oil/rock, as described for HTW accumulators.

OTHER ANCILLARY EQUIPMENT. The need for pipes, pumps, valves, control systems, safety systems, and other ancillary equipment should not be forgotten nor treated lightly in considering concepts. These contribute a substantial but not major part of the capital costs, and for pumps particularly a required diversion of useful power output. For the preliminary screening of Task I, these are considered as lumped into the *installed* costs of the major components described.

PROPONENTS AND CONCEPTS

The literature collected represents the state-of-the-art, both in experimental data and in concept formulation. Many of the references contained useful data on the many elements in the taxonomy described, but did not describe a concept of a thermal energy storage system

(TESS) directly applicable to the objectives of this study: near-term utility applications for conventional coal and nuclear plants. Such references were considered source material.

However, a large number of references proposed and described thermal energy storage systems or major components thereof that could be considered relevant to the study. Either they were originated with this specific application in mind, or it was clear that some important and perhaps novel features of their proposed concepts should be considered in the preliminary screening process in order to explore a wide range of approaches.

These proposers or *proponents* of concepts were identified and their concept was defined in outline form as it might be applicable to this study. In Table 2-1, a list of proponents, the institution(s) and one or more individuals directly associated with the project or reference describing the concept, is given. It is not implied that said institutions or individuals are *advocates* or originators of the concepts, but only that they were named in the source material used.

The proponents listed on Table 2-1 are classed principally according to the storage medium used: HTW, other sensible heat materials, and phase-change materials. Within each class, some institutions and individuals are grouped as joint authors or as describing closely related concepts.

The numbers assigned to proponents refer to Volume 2, Appendix C, in which the outline concept definitions formulated are given. In some cases two or more concept variants will be found for the same proponents in that appendix.

CONTACTS

In the course of collecting, digesting, and using the references in defining the set of concepts, performing the preliminary screening, and the subsequent evaluation and concept selection, telephone and/or correspondence contacts were made with almost all of the institutions or individuals listed above. The cooperation received from proponents

Table 2-1. Proponents of concepts.

HTW Concepts			
1.	Graz University (Austria) Waagner Biro (Austria) Siempelkamp GmbH (FRG) Deutsche Babcock (FRG)	Paul V. Gilli Georg Beckmann F. Schilling, L. Gülicher E. Bitterlich	PCIV
2.	R&D Associates	J. Dooley, S. Ridgway	Concrete Stress Supported Hard Rock Cavern
3.	Ontario Hydro Atomenergi (Sweden)	A.G. Barnstaple, J.J. Kirby Peter Margen	Air Supported Hard Rock Cavern
4.	University of Houston Subsurface, Inc.	R.E. Collins K.E. Davis	Aquifers
5.	General Electric-TEMPO	C.F. Meyer	Aquifers
Other Sensible Heat Concepts			
21.	EXXON Corp.	R.P. Cahn, E.W. Nicholson	Hot Oil/Feedwater
22.	McDonnell Douglas Rocketdyne	G. Coleman J. Friefeld	Hot Oil/Packed Bed
23.	Martin Marietta	F. Blake	Oil/HITEC and All HITEC
24.	Honeywell, Inc.	J.C. Powell, R.T. LeFrois	Oil/HITEC
25.	Bechtel Corp.	William Stevens	Oil/Retrofit
26.	General Atomic ORNL	R.N. Quade, D. Vrable E. Fox, M. Silverman	HITEC/HTGR
27.	General Electric-Space Div.	E. Mehalick	Oil/Drained Bed
28.	University of Minnesota	M. Riaz, P. Blackshear	UG Rock Beds/Hot Air
30.	Jet Propulsion Laboratory	R.H. Turner	Steel Plates
31.	Energy Conversion Engrg.	Allen Selz	Molten Sulfur
32.	Boeing Company	J. Gintz	Refractory Brick/H_e
33.	University of Houston Subsurface, Inc.	R.E. Collins K.E. Davis	Oil in Salt Domes
Phase-Change Materials Concepts			
41.	Xerox Corp.	J.A. Carlson	HX Subsystem
42.	Naval Research Laboratory	T.A. Chubb	Salt/Terphenyl/Steam
43.	Comstock & Westcott, Inc.	B.M. Cohen	NaOH
44.	Inst. of Gas Technology	J. Dullea, H. Maru	Carbonates
45.	Clemson University	D.D. Edie	Immiscible Fluids HX
46.	Honeywell, Inc.	R.T. LeFrois	$NaNO_3$ Slurry/Scrapers
47.	Boeing Company	J. Gintz	Fluorides/Helium
48.	Grumman Corp.	A. Ferrara	HX Concepts
49.	General Electric-CR&D	H. Vakil, F. Bundy	Immiscible Fluids HX
50.	Rocket Research Corp.	E.C. Clark	H_2SO_4
51.	Swiss Federal Inst. for Reactor Research	M. Taube	Immiscible Fluids HX

was excellent, and the authors of this report wish to express their gratitude and thanks. Questions were answered, additional reference material supplied, and referrals made to experts in the subsystems and materials areas.

Many additional sources were consulted including authors of the references considered as sources rather than proponents. In most cases information was freely supplied. Where additional or special information or effort was required that was not part of a current or recent funded project, some respondents were compensated by small sub-contracts or consulting agreements. These included William Stevens of Bechtel National Corp., J. O'Hara of R.M. Parsons Inc., Professor Paul V. Gilli of Graz, Austria, and Professor G.J. Janz of the Rensselaer Polytechnic Institute Molten Salts Data Center.

SECTION 3
PRELIMINARY SCREENING

PURPOSE

The purpose of this preliminary screening is to compare the concepts defined and described, and to delete, combine, and integrate them into a set of twelve or less for more detailed study. Comparing and selection requires criteria to be defined and structured as to relative importance.

CRITERIA FOR SELECTION

A number of criteria were used. At this stage of screening, their use was largely qualitative, and their comparative use was largely based on the statements and data in the proponents' reference documents. The criteria were formulated as a check list for this screening, to be used more quantitatively in the second stage of screening, to concepts for analysis in Task II.

The major categories of criteria are that concepts should:

- Be Compatible with Near-Term Application
- Be Economically Viable in the Mid-Term
- Meet Utility Operational Requirements
- Be Environmentally Sound
- Have Conservation Potential
- Be Broadly Applicable
- Have Potential for Future Growth/Improvement
- Be Diverse in Type

Each of these will be described briefly with an indication of the major subcriteria therein. As listed above they are roughly in the order of importance for preliminary screening.

Compatible with Near Term Application

The phrase near-term has been considered to mean the present to 1985 in policy and planning documents. The result of this study might be an empty-set if significant commercial utilization were required to be in place by 1985, since the time required from initial order or electric utility decision to buy is over ten years for nuclear plants, several years less for large fossil fuel plants, and a large part of the seven years remaining until 1985 even for small, coal-fired, environmentally acceptable plants. The most feasible interpretation of this requirement is that the concept must be able to be demonstrated and operated before 1985 to the extent that in and after 1985 a utility can decide with confidence to order a plant incorporating thermal energy storage systems for load leveling.

The primary deterrent for near-term application is technical risk: the level of uncertainty in the technologies involved, and in the commitments of effort needed to resolve the uncertainties. "Confidence to order" will require resolution of problems in all the other named criteria, but the primary emphasis in this criterion is on the time scale of technologies to achieve the desired performance.

For judging the current status of concepts, a scale from best to worst would include:

- Complete system has been demonstrated at plant or pilot scale
- All subsystems have been so demonstrated
- All technologies required are mature in other applications
- All technologies are known and no major problems foreseen
- Problem areas are known but likely solutions have been proposed
- Serious problems recognized, solutions are speculative.

Quantitative measures of the above, not readily available for this level of screening, include:

- R.D&D time required
- R.D&D costs required
- Probability of development success
- Plant Construction Time after demonstration.

Economically Viable in the Mid-Term

Economic viability in the mid-term, 1985-2000, implies first the resolution of the technical problems and successful demonstration, then that fixed charges and variable costs attributable to the plant modifications required by a concept lead to an annual cost per kilowatt of incremental capacity that is less than or comparable to the alternative ways of achieving such incremental capacity and load leveling. That is, it must compete with the other thermal energy storage concepts considered in this project, as well as with other forms of storage and peaking capacity.

The last two, nonthermal storage and peaking capacity are not to be considered in Task I but must be ultimately addressed in recommendations concerning development of concepts in Task IV. Task I must consider the comparative economics of the concepts defined herein.

It is clear that there are costs primarily determined by the maximum increase in power desired, and costs which are determined by the amount of energy to be stored, which is related to the cycle and pattern of delivery of the increased power.

Capital costs of changes in the turbine generator, feedwater heating system, cooling and waste heat disposal, additional equipment for mass and heat transport and transfer are included in the first category, power related costs. Capital costs of the storage medium and of its containment are the principal part of the second category, energy related costs.

The power related costs depend very much on the details of the thermodynamic cycle chosen to implement a concept, including the state properties of the source of heat delivered to storage, and the state properties of heat delivered to the Rankine cycle steam turbogenerator system. They are roughly independent of the storage medium and containment except as these constrain the input and output state conditions and require more complex conversion equipment such as heat exchangers and evaporators.

The energy related costs for different storage media and different forms of containment can separately be listed in a rough order of increasing cost, but unfortunately the lowest cost storage media require the highest cost forms of containment. Table 3-1 is an approximation of relative costs, with media shown in the first column in order of increasing cost and containment means shown in the second column in order of decreasing costs.

Table 3-1. Approximate relative cost relationships.

Storage Media	Containment
Water	High Pressure Vessels
	• Welded Steel
Rock	• Prestressed Concrete
Sulfuric Acid	• Prestressed Cast Iron
Sulfur	• Lined Underground Caverns
Hydrocarbon Oils (eg, Caloria HT43)	• Unlined Salt Dome Cavities
	• Aquifers
Salts (eg, HITEC)	Ambient Pressure
Steel	• Stainless Steels
Other Metals	• Carbon Steel
Silicone Oils	

COSTS INCREASE ↓ (Storage Media)

COSTS INCREASE ↑ (Containment)

The approximate nature of this ranking must be emphasized. The bar in each column indicates the division between high pressure and ambient pressure. Water at high temperatures requires high pressure containment for storage. The containment forms above the bar – the most expensive – are required for this medium. Below the bar there will be a considerable price range for each of the media depending on the units of measurement and on the special requirements put on them.

The units which measure a storage medium's effectiveness vary from the cost per kilogram, the units in which it would usually be purchased, to cost per m^3, cost per kJ stored over a fixed small temperature range, cost per kJ over the working temperature range in a particular concept, and cost per kJ over the maximum possible working range of the medium. Beyond this, materials such as salts or eutectics of salts would differ widely according to the salts contained and to the purity required. There can be an order of magnitude difference between the cost of the technical grade and CP grade. Considerations of corrosion, stability, need for more expensive containment material, and environmental hazards may outweigh any advantage from using the cheaper grade.

Economic viability is also affected by the turnaround efficiency, defined as the ratio of the electricity actually produced from the energy delivered from storage to the electricity that could have been produced from the energy that was diverted to storage. The effect of low turnaround efficiency is to increase the fuel required per kilowatt hour to generate the electricity delivered during peak hours.

Other variable costs such as operating and maintenance costs may be critical to economic viability, for example if fouling or corrosion requires frequent attention in heat exchangers. If a storage medium used at high temperatures degrades, so that continual makeup or periodic replacement is needed, this adds an annual cost to be considered in levelized annual costs over the life cycle of the plant.

Utility Operational Requirements

Electric utilities have conventional methods of assuring the delivery of electricity reliably, to all customers, when needed, over their entire service area. To be considered, a new system must meet their needs in the various categories outlined briefly below.

- Site Flexibility

  To serve customers effectively there is need for plants distributed over their service area. The geologic needs of a concept, such as competent hard rock, salt domes, or aquifers may not be met in the desired load area. Water needs, land requirements, aesthetic acceptability of a conceptual design, or catastrophic risks to the community beyond the plant area may limit siting flexibility.

- Operating Flexibility

  Principles of dispatching plants to meet current and expected load fluctuations include lowest incremental cost, and ability to maintain high reliability. Some aspects of the thermal energy storage systems and the associated conversion equipment that will be of interest include the following:
  - Startup time of a cold plant or peaking turbine, the time for conversion from storage charging to discharging and vice versa, and the shutdown time
  - Capability for rapid load following over a range of demands
  - Part load efficiency as well as full load efficiency
  - Minimum load that can be safely met
  - Ability to maintain the boiler island (nuclear or fossil) at constant output, free from transient demands
  - Flexibility to provide load leveling according to the different daily and weekly load patterns of different seasons
  - Ease of control and transient stability.

- Reliability

  Reliability of a particular plant is measured in terms of its availability, ie the fraction of the year that it is available to produce its rated output. It may produce less than rated output for some hours of the year if the demand or the utility dispatch procedures warrant. Both scheduled or planned outages and forced outages reduce the availability. Planned outages for maintenance and minor repair can generally be scheduled to seasons when demand is low. Forced outage probability, found by experience, largely determines the amount of reserve capacity the utility must own or have on call to meet its overall standard of reliability, eg insufficient available capacity to meet peak demands should not exceed a probability of one day per ten years.

For the thermal energy storage load leveling systems, the technologies employed should be tested adequately to insure low forced outage rates. In selecting concepts, those which permit continual operation of the main turbine generator despite a forced outage of the peaking turbine or parts of the storage system have added value. Ability of the peaking turbine to operate when the main turbine is on forced outage has value. Ability of either or both turbines to meet some level of load from thermal storage when the boiler island output is reduced to zero also has value.

One of the significant although unquantified benefits expected from thermal energy storage load leveling systems is improved reliability and lifetime of the boiler island if its required output does not fluctuate. Currently 50 percent of forced outages of fossil plants larger than 600 MW are caused by problems in the boiler island (Reference 231).

- Operating Hazards

The addition of a thermal energy storage load leveling system adds operational flexibility, but may, if improperly designed, jeopardize the conventional system with which it operates. The reliability and life of the turbine generator system are critically dependent on a very carefully controlled quality of boiler feedwater. Unwanted solids, liquids, or gases in the feedwater can impair boiler heat exchange by scaling, can cause corrosion in the boiler or turbine, can cause erosion or even blade breakage if sizable pieces of scale enter the turbine. The steel used in the turbine, in heat exchanger tubes, and in pipes must have special properties. The liners used for HTW storage and the heat exchangers for other storage media must have these same properties.

When the sensible-heat storage or heat transfer fluids have properties which would cause major system damage if they leaked into the feedwater loop, due precautions must be taken that leakage is avoided, or is in the opposite direction and is quickly detected.

Some of the concepts of turbine operation require off-design-point operation of the turbine. Thermal stresses, transient stresses, different vibration modes and all other possible consequences of the deviations from conventional practice must be considered.

Environmentally Sound

In part the environmental constraints are subsumed in the above criteria in that siting flexibility, economic viability, and operational flexibility all are affected by the national and local environmental standards and requirements. As a summary in its own right, environmental effects to be evaluated in comparing thermal energy storage load leveling systems include:

- Normal operation must not be accompanied by unacceptable air or water emissions such as: conventional pollutants, NO_x, CO, particulates, hydrocarbons, radioactive material
- Aesthetics, water use, and land use must be locally acceptable
- Special emissions/waste disposal problems must be acceptable
  - Leakage of storage oils or salts
  - Fumes from degradation of materials
  - "Blowdown" products of periodic makeup or replacement
- Catastrophic risks must be demonstrably minimal or tolerable
  - Seismic damage
  - Storm or flood damage
  - Pressure vessel failure
  - Toxic material leakage into air, or surface or ground water
  - Fire or explosion danger from flammable materials.

Conservation Potential

As all thermal energy storage systems will suffer some losses and degradation of the energy through charging and discharging storage, more energy may be required than from operating a base load plant in a load following mode. However certain comparisons will show energy conservation, in the sense of conserving the scarcer and more critical resources, eg oil and gas.

To the extent that the concepts here considered replace the use of oil in gas turbine peaking capacity, they represent conservation of oil and progress toward reduction of imported oil. If the heat rate of the low-capital cost gas turbines is higher than the incremental heat rate of a thermal energy storage system, including its turnaround efficiency, there is a saving of net energy. If the thermal energy storage system replaces old, low-efficiency fossil plants that have been used for intermediate range duty, there may be a net savings in energy.

Finally, if the turnaround effiency is higher than that of an alternative storage system, such as pumped hydro storage, conservation of energy may be achieved.

Broadly Applicable

The commercialization of a system is easier if its range of applicability is large, both geographically and in size and type of heat source. All else being equal, a system that can be applied to nuclear plants and to small and large fossil plants has more market potential and is preferred to specialized types.

Potential for Future Growth/Improvement

Some systems can be synthesized from components that are considered near-term, but could be improved in performance or cost if technologies not yet demonstrated can be developed. (For example: molten salt alone is near-term, molten salt and compatible packed bed may not be near-term.)

Some storage materials may have a high current price because of low demand. The effects of large continued demand should be considered.

Some systems may be more sensitive than others to net escalation of the fuel used by the load leveling plant (coal or nuclear), or by the competing peaking options (oil or gas). Long-range as well as near-term economic relations should be considered.

Diversity

Another criterion that must be seriously considered is diversity of concepts. Even if it should appear that a dozen variants of one particular concept were superior on all criteria to all the other concepts, it would be unwise to so narrow the set to be considered in more detail in the second half of Task I. The preliminary nature of this first screening relies in part on proponents' data and analysis, and each analysis cannot be relied upon to be comparable in assumptions to that of other proponents and concepts.

To the extent possible within the limits of twelve or less surviving concepts, major components and concepts not clearly rejected by failure to meet important criteria should be retained. Closely related concepts and variants may be combined into a single concept to accomplish this objective.

THE SCREENING PROCESS

The screening of the many defined concepts (numbered as in Table 2-1) and their variants down to a maximum of twelve, without detailed analysis, required primary emphasis on just a few criteria:

- Suitability for the utility application as defined
- Near-term availability
- Higher economic ranking than similar concepts
- Retention of diversity.

At this stage other criteria, such as siting limitations, were considered but not used to reject a concept unless clearly overriding disadvantages were recognized. Considerable judgment was required, considering that the descriptions by proponents were often not of complete systems, or were described for another application such as solar-thermal storage.

On the other hand, the concepts and variants defined often have much in common, either in components or in system configuration, and do not require separate analysis. It was clear that a containment concept proposed to operate with one system configuration of conversion to steam and to electricity can work perhaps equally well with alternative conversion concepts, and similarly that each conversion concept can work with several alternative containment concepts. With sensible heat storage, the various alternatives of oils, salts, metal, rock, sulfur, etc, are virtually interchangeable within a configuration, with cost of storage medium, compatibility with other materials, stability at high temperatures, and characteristics that determine heat exchanger costs as the principal parameters to determine a relative ranking.

Each concept defined (Appendix C) contained a feature or features that are different. To meet the diversity criterion and reduce the set to twelve candidate concepts for further study, combinations of concepts that incorporate one or more of the unique features appeared to be necessary. Thus, the candidate concepts chosen are often an integration of the concepts of several proponents, and will be called Selections, or selected concepts.

The selected concepts are introduced by summary figures, with a brief textual amplification of the considerations involved in proposing each. The selected concepts have been grouped. The first seven selections emphasize different forms of HTW pressure containment. The remaining five emphasize low vapor pressure (LVP) storage media.

These twelve were presented to the NASA program manager and his review board for approval, in accordance with the subtask structure in Figure 1-2, and with the consent of DOE/NASA/EPRI were approved for further study in the second half of Task I.

SELECTED CONCEPTS

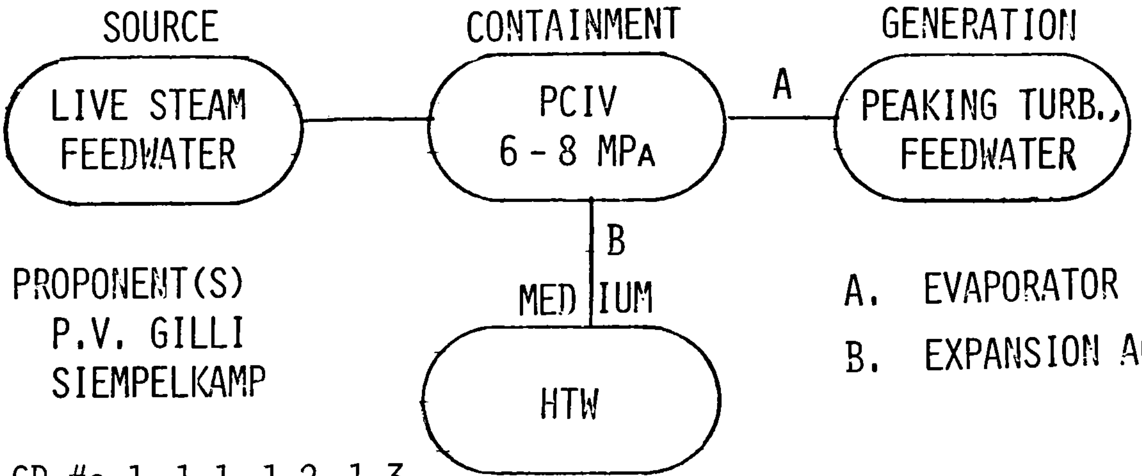
#1 – Prestressed Cast Iron Vessels (PCIV)

This selection features the prestressed cast iron vessel (PCIV) as the containment for high temperature water (HTW) under pressure. This is the first of seven selections using HTW as the storage medium and differing in the form of containment and the conversion to electricity. Reference 45, and Concept Definition #1 (CD-1) in Appendix C, describes different modes of use of the PCIV, that are almost equally applicable to the other forms of HTW pressure containment.

Professor Paul V. Gilli, now with the Graz University of Technology, Austria, has been prolific in descriptions of concepts for thermal energy storage systems for utility applications; with various coauthors, including G. Beckmann, K. Fritz, and F. Schilling, he has published over 15 papers in the field. Initial papers in the early 1970's used steel pressure vessels, as were used in the Berlin-Charlottenburg steam plant which has operated with storage since 1929. The PCIV, proposed by Siempelkamp Giesserei KG of Krefeld, FRG in the late 1960's, was adopted as a more satisfactory containment in recent papers. Both Gilli and Siempelkamp are listed as proponents in Figure 3-1 and in Table 2-1.

In a study of *Thermal Energy Storage Using Prestressed Cast Iron Vessels (PCIV)* for ERDA (Reference 45) Gilli and Schilling detail their ideas. For Selection #1 the variant described as CD-1.3 is selected as seemingly favored by Gilli for its high turnaround efficiency, high energy storage density, and ability to put out more power than a purely feedwater storage system. The source is both live steam and feedwater to fill an expansion mode accumulator. One stage of evaporator steam generation is used with the steam going to a peaking turbine and the water discharge of the evaporator being delivered to the boiler inlet as feedwater. The same configuration could be equally well used with prestressed concrete pressure vessels (PCPV) or steel vessels.

ADVANTAGES. The PCIV direct costs per m^3 of capacity as optimized by Gilli are lower than estimates by others on PCPV and steel vessels



PROPONENT(S)
P.V. GILLI
SIEMPELKAMP

A. EVAPORATOR
B. EXPANSION ACCUMULATOR

CD #s 1, 1.1, 1.2, 1.3

- CHARGE: STEAM FROM BOILER PLUS FEEDWATER FILLS ACCUMULATOR
- DISCHARGE: HTW FROM STORAGE THROUGH EVAPORATOR (STEAM AND WATER)
  STEAM TO COLD REHEAT POINT (CRH)
  PEAKING TURBINE FROM CRH AND CROSSOVER (CO) STEAM
  WATER TO BOILER INLET EFF. = 0.80 - 0.85
- PCIV
  PRESTRESSED CAST IRON VESSEL
  COST OF CONTAINMENT $1248/M^3
  SAFETY
  EASE OF ASSEMBLY

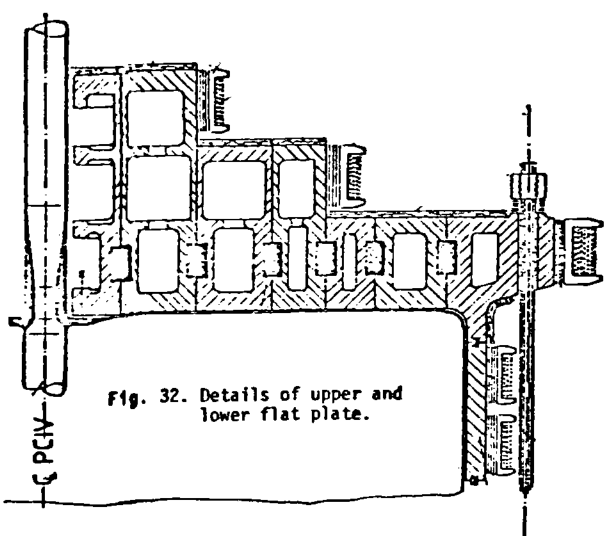
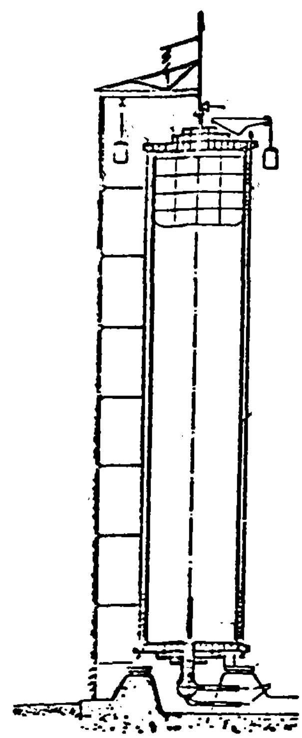


Fig. 32. Details of upper and lower flat plate.



(Reference 45)

Figure 3-1. Selection #1.

(respectively 1248, 1600, 4000 $/m^3). The cycle combines the merits of a feedwater storage system and a flash evaporator system. A turn-around efficiency of 0.80 to 0.85 is estimated. PCIV shares with PCPV a safety advantage over steel pressure vessels. PCIV can be easily site assembled from factory made castings.

DISADVANTAGES. Cost of containment is higher than underground containment concepts. While small sizes of PCIV at moderate pressures have been built and tested, nothing has yet been demonstrated at the size, temperature, and pressure levels required for this application (eg 6 MPa, 250°C). Current concept requires external thermal insulation, part of which, under the prestressed cable shoes, must be pressure resistant. The cast iron operates hot. Effects of thermal and pressure cycling on the prestressing system have not been tested. This is the reason Gilli chooses the expansion accumulator mode, as most constant in P and T. (Note: Siempelkamp indicates they are developing an insulation internal to the liner which would be compatible with boiler quality feedwater. No details available.) The technology resides in Siempelkamp; transportation costs to the U.S. would be large; alternatively, developing a comparable technology in the U.S. by license or independent development may not be "near-term available."

#2 – Prestressed Concrete Pressure Vessels (PCPV)

Prestressed concrete has been used in many applications, and as pressure vessels (PCPV) for nuclear reactor secondary containment for over 10 years. Bechtel Power Corp. lists 59 PCPV's they have engineered or constructed. There has been no specific proponent for a TES system using PCPV for thermal storage, but they can be considered for any HTW storage concept requiring pressure containment. None have been built or tested for the pressure and temperature range of interest (the reactor containment vessels were rated under 0.5 MPa (60 psi)).

The candidate concept selected is shown in Figure 3-2. The variable pressure accumulator mode is named, for diversity, although as indicated it can be considered with the steam cycle configurations of

LIVE STEAM — PRESTRESSED CONCRETE PV INSULATING CONCRETE + COOLING SYSTEM — PEAKING TURBINE

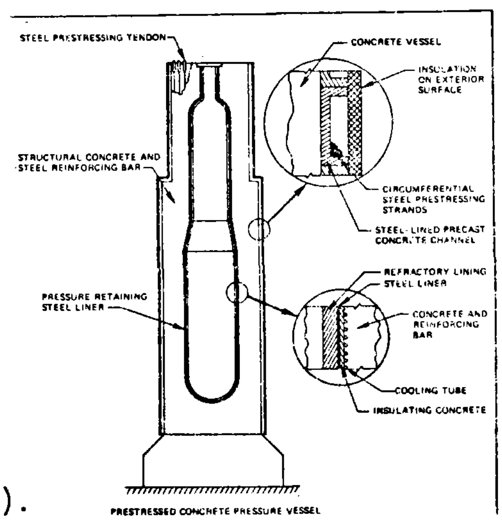
HTW TO 360°C

PROPONENTS
- CEGB
- ORNL
- T.Y. LIN INT.
- R.M. PARSONS/ J. O'HARA

CD #6

- CHARGE: LIVE STEAM THROTTLED TO REFILL VARIABLE PRESSURE ACCUMULATOR (OR USE EXPANSION MODE ACCUMULATOR WITH LIVE STEAM PLUS FEEDWATER).
- DISCHARGE: SAME AS #1 PCIV, OR #4 UG CAVITY
- STEEL LINE IS SURROUNDED BY THIN INSULATING CONCRETE AND COOLING SYSTEM. REINFORCED PRESTRESSED CONCRETE SURROUNDS THE LINER AND SYSTEM.

  COST OF CONTAINMENT, 1600 $\$/M^3$



Reference 200 (Greenstreet).

Fig. S.1. PCPV versus steel vessel (particular details for HYGAS gasifier).

Figure 3-2. Selection #2.

Selections #1 and #4 as well. Listed as proponents are groups interested in high pressure, high temperature PCPVs. These include Ian Glendenning of the Central Electricity Generating Board, UK, for thermal storage for compressed air energy storage systems; W.L. Greenstreet, et al, at ORNL for coal gasifier containment; and James O'Hara of R.M. Parsons Inc. and Philip Chow of T.Y. Lin, International who have completed a Department of Energy study of conceptual designs of PCPV for four coal gasifier process components.

ADVANTAGES. PCPV is considerably cheaper per m^3 contained than steel vessels for comparable duty, according to reports both by O'Hara and Glendenning. It can be built on site in large unit sizes. The redundancy of prestressing cables and tendons reduces the chances of catastrophic failure by cracking. There is a high level of confidence in the technology through experience (but not for P and T of interest). ASME Code Section III Div. 2 applies to Concrete Reactor Vessels, and would be a start toward code approvals of a higher pressure and temperature PCPV.

DISADVANTAGES. Not built and tested for temperatures and pressures of interest. More costly than PCIV (if the cost assumptions by the several estimators are comparable). Must be site assembled, labor intensive, long construction time. Bulkier than PCIV or steel, external size much bigger than internal capacity; possible aesthetic/land-use objections. PCPVs require cooling to protect the concrete and reinforcing bars from high temperatures; the cooling systems are expensive and imply thermal energy losses.

#3 – Steel Vessels

The use of thick wall steel tanks as pressure vessels has been referred to in Selections #1 and #2. They have long been used. Experience in construction, inspection, test, and use of them is long standing; they are a mature technology. At high temperatures and pressures the cost of containment in them is high compared to the estimates made for PCPV and PCIV. However, steel pressure vessels definitely qualify as near-term available; the others may not, and the cost estimates on the undeveloped systems may prove to be overly optimistic.

In a recently completed contract, the Jet Propulsion Laboratory explored the use of steel as a thermal storage medium and containment means. A number of concepts were proposed and explored sequentially. Initially, emphasis was put on steel as the storage medium; thick bars or slabs contained passages for HTW which would heat the steel. Recognizing that steel was far more expensive as a storage medium than water, the emphasis shifted to the configuration shown in Figure 3-3.

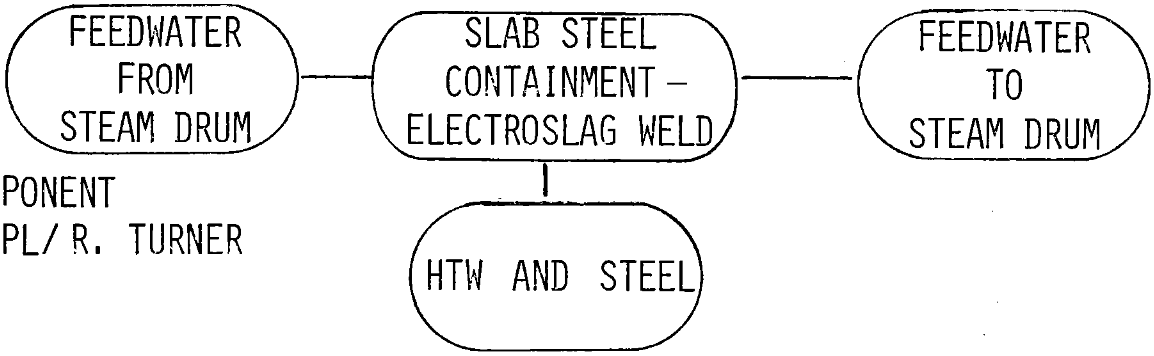
In this configuration, thick slabs of common steel are electroslag welded to form a square channel to contain HTW. As shown the steel is 60 percent of the area, 90 percent of the weight, and stores 40 percent of the thermal energy. Stacking such units crosswise as shown was postulated to make a compact, stable storage system.

A distinctive feature proposed in Reference 181 is deriving the HTW from the steam drum inside the fossil-fired steam supply. Water here can be at over 375°C (700°F) and at 17 MPa (2500 psia). Interfacing charging and discharging at this point would require major design changes in the steam supply, as discussed in Section 4. However, the containment concept can be applied to many other TESS cycles using HTW storage.

Later concepts abandoned the large slabs of 6" steel and proposed many small diameter tubes with a wall thickness designed for the pressure, and with sand packed between tubes as the storage medium.

Estimated TESS costs and containment component costs were not made available for these concepts by JPL.

A well-known constructor of steel pressure vessels, Chicago Bridge and Iron, was asked to provide cost estimates as an added check on the estimates made by non-proponents of steel tanks. The cost estimates were not received during the performance period of this task.



PROPONENT
JPL/ R. TURNER

- CHARGE: WATER AT SATURATION T EXTRACTED FROM BOILER/STEAM DRUM, PUMPED INTO STEEL PIPE/TANKS MADE OF WELDED STEEL SLABS.
- DISCHARGE: HTW OUT OF STORAGE DELIVERED TO STEAM DRUM.
- COMBINED STEEL AND HTW STORAGE
  40% OF TES IN STEEL; VARIANT USES SAND AS STORAGE MEDIUM.
- COST: > 4000 $/$M^3$

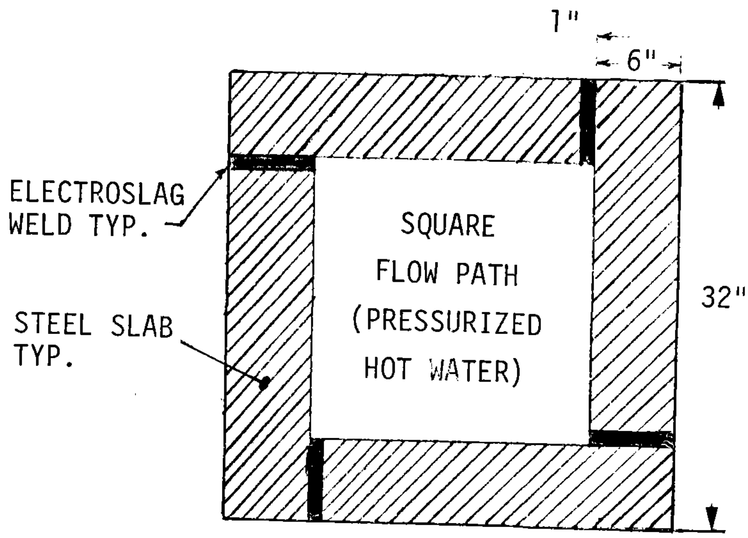


FIGURE A

FOUR SIDED STEEL SLAB THERMAL ENERGY STORAGE UNIT

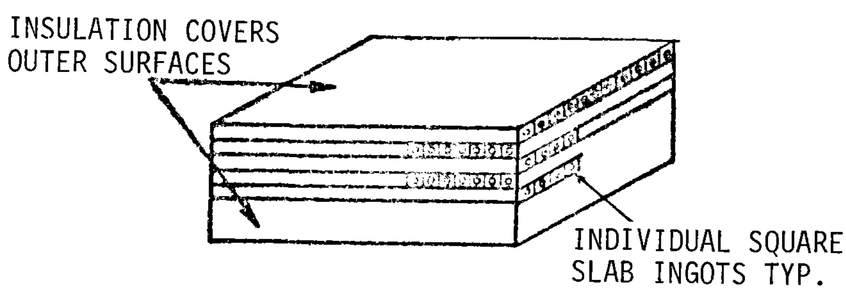


FIGURE C

OVERALL SQUARE SLAB THERMAL ENERGY STORAGE UNIT

(CD #30.3, References 180 and 181)

Figure 3-3. Selection #3.

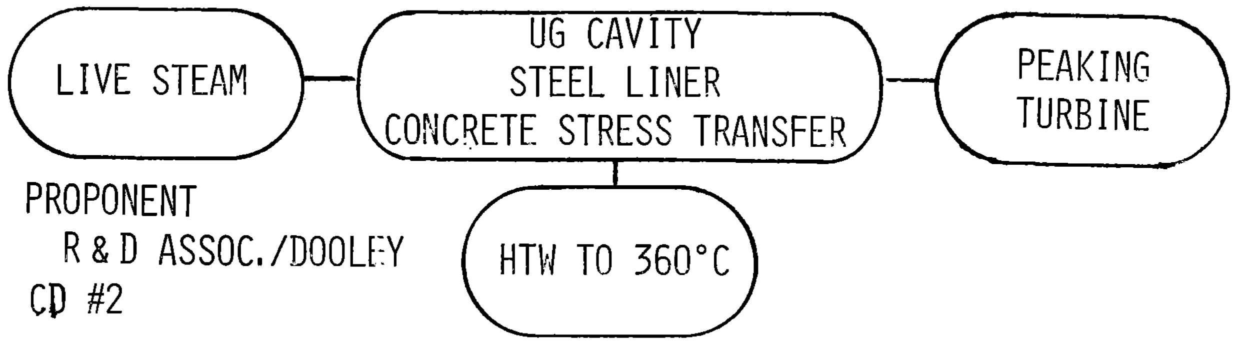
ADVANTAGES. Steel pressure vessels are near-term available with years of design and operating experience at pressures and/or temperatures over those required for thermal storage. Made in modular sizes they can be factory built, inspected and tested, and transported by available rail cars. ASME codes spell out in detail the requirements on materials, methods of construction, inspection, test, and use for the protection of the user and the public. Steel pressure vessels will be used for other components of TESS (eg evaporators, heat exchangers) and of the utility plant.

DISADVANTAGES. Cost is a major disadvantage. Any emphasis on steel as storage is probably even more expensive than steel as containment. The volume to be contained for thermal storage may be in the hundreds of thousands of cubic meters, a far larger volume than most pressure vessel applications. Although building and testing to code should minimize the danger of catastrophic failure, the large number of modules at risk may prove unacceptable.

#4 – Underground Cavity - Concrete Stress Transfer

This is the first of three candidate concepts featuring underground storage of high temperature water (HTW). Selection #4 as summarized in Figure 3-4 features an excavated cavity 30 meters or more in diameter, in competent hard rock, with a steel liner fabricated within the cavity and high-temperature high-strength concrete poured between liner and rock for stress transfer. The means of stress transfer distinguishes this candidate concept from Selections #5 and #6.

The proponents are James Dooley and his colleagues at R&D Associates, Marina del Rey, CA (Reference 28). In an excellent section on *Cavity Considerations,* the procedures and precautions for excavation of cavities are explained. A shaft is excavated to a depth where the overburden will sustain the pressure of storage. Upper and lower horizontal tunnels at this depth provide access to the planned locations for one or more cavities. A small shaft is drilled between the



PROPONENT
R & D ASSOC./DOOLEY
CD #2

- CHARGE: LIVE STEAM THROTTLED TO REFILL VARIABLE-PRESSURE ACCUMULATOR
- DISCHARGE: VARIABLE PRESSURE FLASHED STEAM GOES TO PEAKING TURBINE EFFICIENCY = 90 - 95 PERCENT
- SPHERICAL CAVITY IN COMPETENT ROCK CONCRETE FOR STRESS TRANSFER AND INSULATION

COST OF CONTAINMENT
150 - 300 $/M^3$

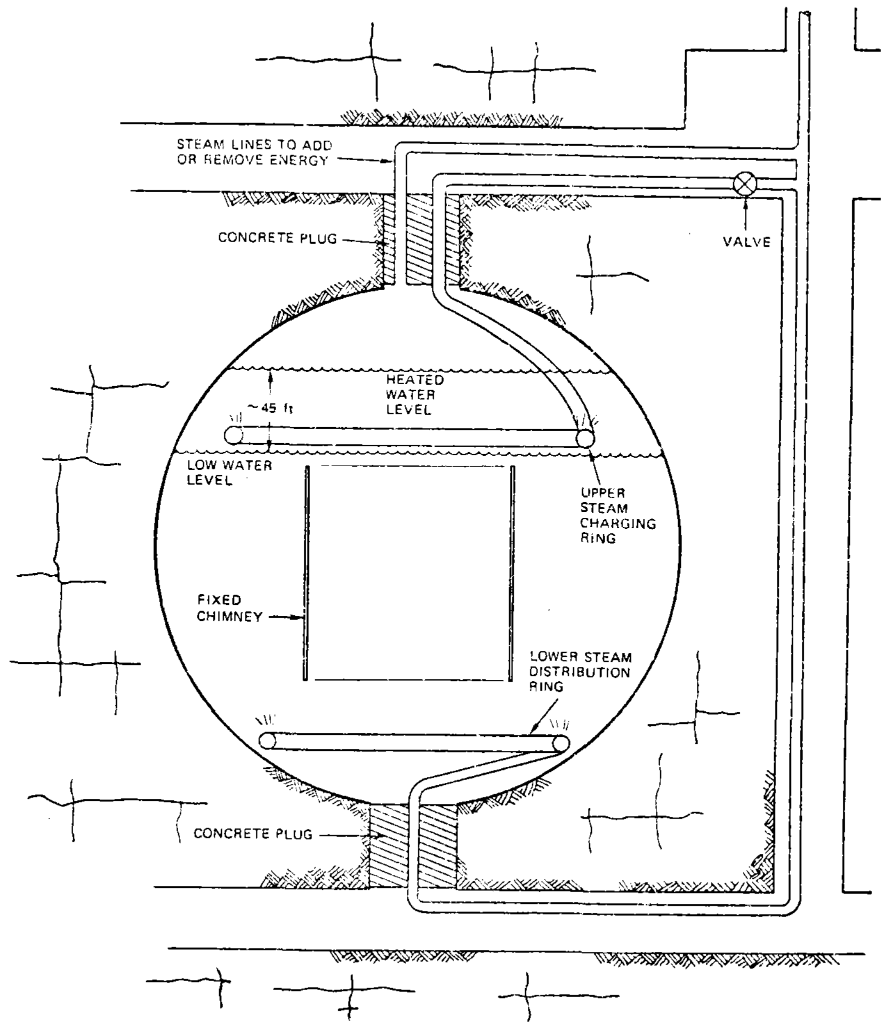


Figure 3-6. Schematic of Equipment in Cavity (Alternate B)
(Reference 28)

Figure 3-4. Selection #4.

upper and lower tunnel and the rubble or muck from all subsequent excavations is removed via the lower tunnel and the main shaft. Spherical cavities from 30-100 m (100-300 ft) in diameter are described as a baseline concept but it is indicated that shape of cavity may be of secondary importance.

In excavating the cavity from the top down, by drilling, blasting, and removal of muck, additional operations are needed such as rock bolting to reduce slippage of rock along natural weaknesses; grouting. and shot-creting to control water flow and reinforce weak areas; and mounting panels of the steel liner to rock-bolts. After welding and X-ray inspections, the high strength concrete is injected between liner and rock.

The use of the lined cavity proposed is as a variable pressure accumulator. Live steam charges the water in the cavity to saturation temperature. For storage discharge the pressure is reduced and a fraction of the water flashes to steam. This mode requires piping only steam through the vertical shaft; expansion mode or displacement mode accumulators would require pumping HTW, to and from the surface against a head of 300-600 m while maintaining saturation pressure in the HTW in all pipes.

Including both the estimated direct costs for a 60 m (200 ft) diameter cavity and for the vertical shaft, the estimated cost of storage is about 250 $\$/m^3$, considerably less than the aboveground pressure containment. By restricting the fraction of the water flashed to steam, hence the change in pressure and temperature of the steam, a turnaround efficiency of 90-95 percent was estimated by the proponent.

ADVANTAGES. Low cost of storage per unit volume. This permits reduced demands on pressure swing for high turnaround efficiency. Unit size of storage volume can be quite large; multiple storage volumes can share a common shaft for further cost reductions. Low insulation cost, and low "equilibrium" thermal losses. Low visibility of storage system; low hazards to personnel and public. Excavation technology is near-term available where precedent exists.

DISADVANTAGES. Underground cavities in competent rock are limited in siting. Map estimates in Reference 28 show about 30 percent of the area of the U.S. as likely sites; these areas probably touch utility areas serving over half the U.S. population. Excavation technology at the larger sizes (100 m diameter) stretches current technology and may be more costly than estimated. Systems exposing the rock to high temperature and periodic pressure cycling have not been built and demonstrated.

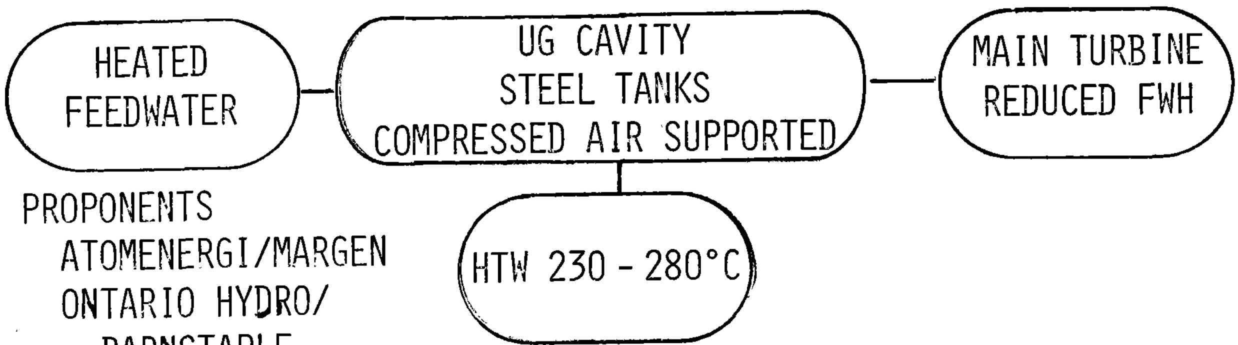
#5 — UG Cavity - Air Supported

Following a concept described by Peter Margen of Studsvik Energiteknik AB Sweden (formerly AB Atomenergi Sweden), Ontario Hydro of Toronto, Canada, proposed and explored an underground cavity for HTW storage in which the stress in a thin steel liner is minimized by use of compressed air between liner and rock (Figure 3-5). Stress transfer is by compressed air at or above the saturation pressure, rather than by concrete as in Selection #4. An equalizing tank connected to both HTW and air limits pressure differences to that caused by the head of water in the tank. Excavation, shaft, and piping costs are to a first approximation much the same as for Selection #4.

The power conversion concept used in CD #3 and CD #8 is feedwater storage. To charge storage, extra HTW is generated by excess steam extraction. To discharge storage, HTW is withdrawn from storage and delivered to the nuclear steam supply system inlet, and an oversized main turbine produces more power because of reduced steam extraction.

Ontario Hydro proposes a limited size of tank, of domed cylindrical shape, but postulates that the excavation can be a gallery 30 m wide and as much as ten times as long, so multiple tanks can be placed within the gallery.

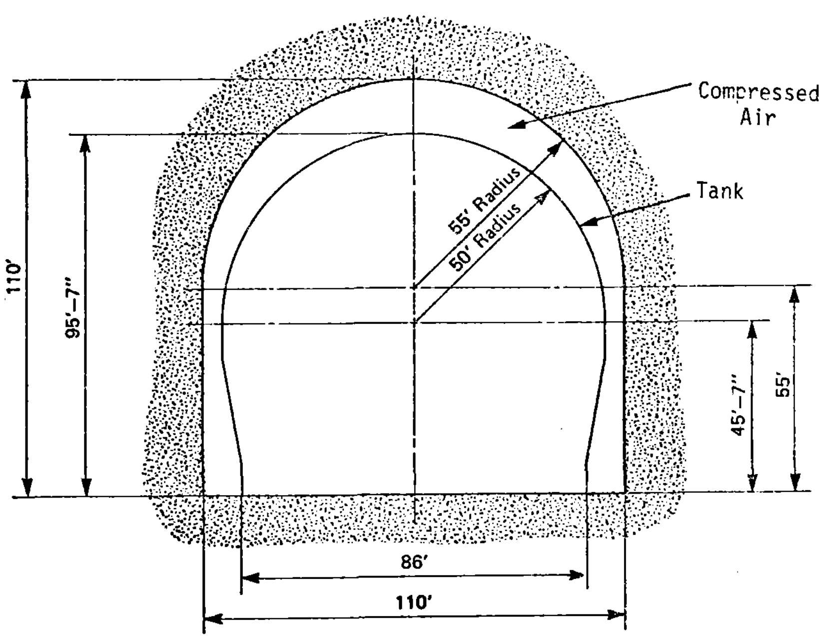
ADVANTAGES. The same advantages for underground cavities apply as for the previous selection. Compressed air stress transfer permits



PROPONENTS
ATOMENERGI/MARGEN
ONTARIO HYDRO/
BARNSTAPLE

CD #3

- CHARGE: EXCESS STEAM EXTRACTION PRODUCES EXTRA HOT-FEEDWATER FOR STORAGE. UG TANK OPERATES IN DISPLACEMENT MODE WITH A THERMOCLINE.
- DISCHARGE: HOT WATER FROM TOP OF TANK REPLACES NORMAL FEEDWATER SO SPECIALLY DESIGNED MAIN TURBINE CAN PRODUCE 25 PERCENT MORE POWER - EFFICIENCY 80 PERCENT
- AIR-SUPPORTED TANKS
  THIN WALLED TANKS ADEQUATE IF PRESSURE EQUALIZATION IS ASSURED
- FEEDWATER LOOP STORAGE
  BASICALLY MORE EFFICIENT THAN MOST STEAM STORAGE BUT LIMITED IN CAPACITY SWING



TANK SHAPE AND CAVERN CROSS SECTION
(Reference 3)

Figure 3-5. Selection #5.

external thermal insulation on the tanks the compressed air is cooled so that rock temperatures are near ambient. Feedwater storage gives a high energy density in kWh/m^3 and a high turnaround efficiency.

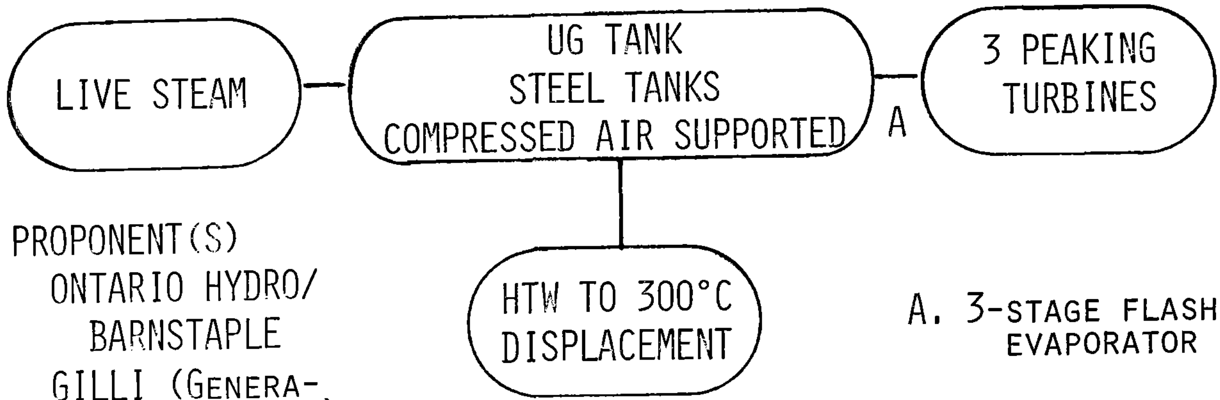
DISADVANTAGES. Many of the disadvantages for the previous selection also apply. Site selection is limited by geology. Leakage of compressed air out of, or of groundwater into, the cavity may be hard to control by grounting or shot-creting. Has not been demonstrated. Use of displacement mode of storage with a thermocline imposes thermal stresses on the steel tank. HTW must be pumped down and up again without flashing to steam; extra pumping may be costly. A purely feedwater storage system can provide only a limited amount of peaking capacity. Without major changes in the nuclear steam supply, peaking is limited to about 15 percent of rated reference plant capacity; to attain even this much requires turbine modifications and redesign that may not be near-term available for large nuclear plants.

#6 – UG Cavity - Evaporators

This candidate concept uses the underground cavity technology with compressed air stress transfer as described in Selection #5. The unique feature is a three-stage steam generator using flash evaporators at 2.6, 0.9, and 0.2 MPa (Figure 3-6). A larger power swing (ratio of peaking capacity to rated capacity) is achievable than with pure feedwater storage. The displacement mode with thermocline is still utilized in the underground cavity.

ADVANTAGES. The principal advantages of selections #4 and #5 apply. Use of the three-stage evaporator permits a larger power swing. The peaking turbines are available technology, using modules, eg 2 two-flow LP turbines, to stay within the capabilities of available sizes.

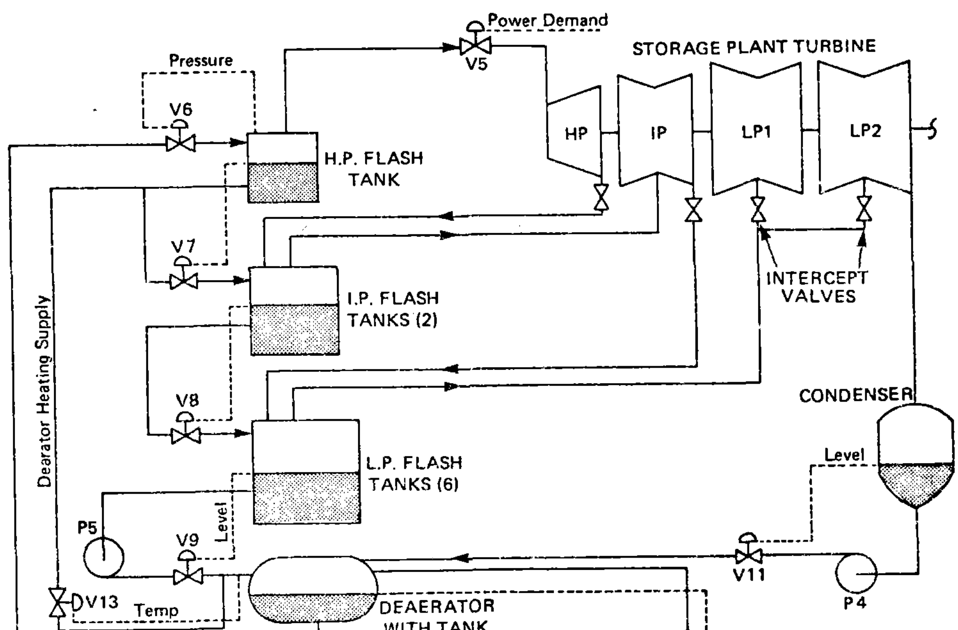
DISADVANTAGES. These are as listed for the preceding underground cavity concepts.



PROPONENT(S)
ONTARIO HYDRO/
BARNSTAPLE
GILLI (GENERATION CONCEPT)
CD #3.1

A. 3-STAGE FLASH EVAPORATOR

- CHARGE: COOL WATER FROM BOTTOM OF STORAGE IS MIXED WITH LIVE STEAM TO MAKE HTW AT 300°C, WHICH IS INJECTED AT TOP OF STORAGE
- DISCHARGE: HTW FROM TOP OF STORAGE GOES THROUGH THREE FLASH EVAPORATORS. WATER FROM LAST EVAPORATOR IS RETURNED TO BOTTOM OF STORAGE.
- FLASH EVAPORATORS
  CONVERTS HTW TO CONSTANT PRESSURE STEAM AT 3 LEVELS.



(Reference 3)

Figure 3-6. Selection #6.

#7 – Aquifer Storage

Storage of HTW in aquifers, ie porous layers of water-saturated gravel, sand, or sandstone confined between impermeable layers, as illustrated in Figure 3-7, can have an extremely low energy related cost. The aquifer is available over a wide range of sedimentary geologic areas without excavation or modification. However, the power related costs are significant for they include the cost of drilling and casing the wells, the cost of pumps and pumping energy, and the cost of heat exchangers. The doublet well concept illustrated permits recycling hot and cold (or warm) water to and from the same aquifer to minimize resource usage. The temperature range over which aquifer storage can be effective is unknown, experiments or demonstrations have not been made except at nearly ambient temperatures.

CD #4 (References 26, 169) proposed to store very high temperatures at great depths for containment (350°C and 1500 m) so that withdrawn HTW could be used to generate steam for power production. At these temperatures, using aquifer storage, increased solution of minerals and/or chemical changes occur so that cycling of the water temperature could soon cause precipitation and scaling, plugging the aquifer and the heat exchanger. Dr. Collins from the University of Houston no longer favors this approach.

A lower temperature range, 100-200°C is believed usable by the proponents of CD #5 (Reference 125). This range may be usable for feedwater storage (if up to 200°C is feasible) or for district heating to supply space heating, residential hot water, and industrial heat loads in this temperature range. This use of storage may be an adjunct to some of the other candidate concepts for storage, in that a daily cycle of storing thermal energy during off-peak hours, thus modifying the electric output supply, can be combined with seasonal withdrawal from aquifer storage for space heating.

ADVANTAGES. Very low cost of storage per kWh (essentially zero: only losses and maintenance are energy related). Capacity for very large amounts of energy storage for weekly and seasonal cycles as well as smaller daily cycles.

HOT FEEDWATER, EXTRACTION STEAM — CONFINED AQUIFERS TO > 300 M — MAIN TURBINE REDUCED STEAM EXTRACTION

HTW TO 200°C

PROPONENT(S)
GE-SB/MEYER
U OF H/COLLINS

CD #4,5

- CHARGE: GROUNDWATER WITHDRAWN FROM RIGHT WELL IS HEATED BY EXTRACTION STEAM AS FEEDWATER AND STORED IN LEFT WELL.
- DISCHARGE: REVERSE FLOW FROM LEFT TO RIGHT WELL HEATS FEEDWATER WHICH BYPASSES NORMAL FW HEATERS.
- AQUIFER STORAGE

  VERY LOW $/KWH PERMITS CONSIDERING LONG-TERM STORAGE AND VERY LARGE VOLUMES.

  COLLINS SUGGEST USE TO OVER 300°C BUT GEOCHEMICAL EFFECTS ARE UNCERTAIN.

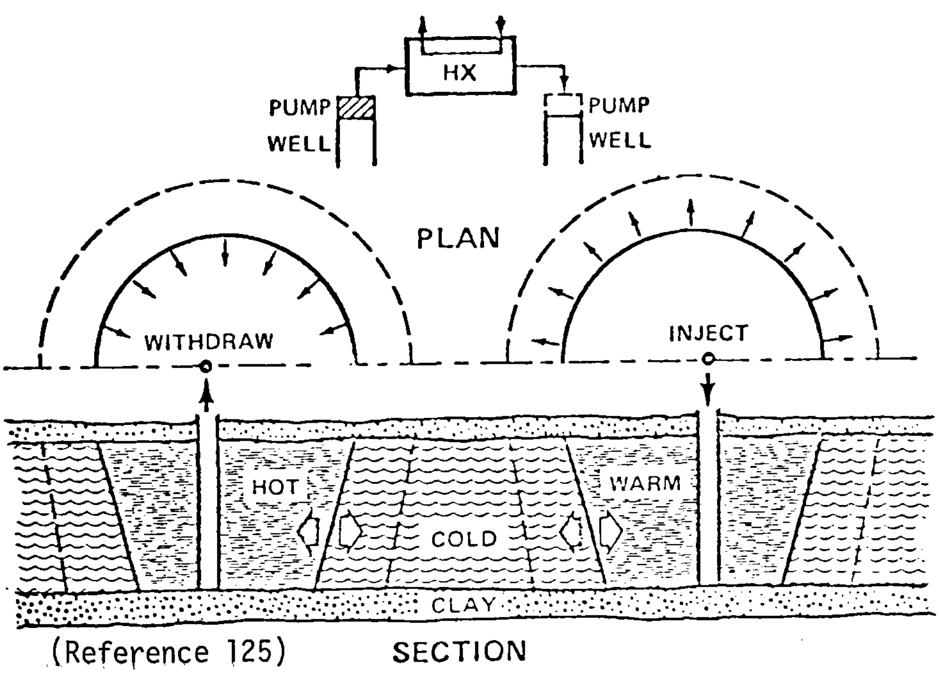


Figure 3-7. Selection #7.

DISADVANTAGES. While aquifers are widely available, their usability will be site-specific. Some areas not suitable. There will be constraints against using or endangering aquifers containing potable water. Geochemistry effects versus temperature not understood or fully explored. Not near-term available in that tests or demonstrations of significant size and useful temperatures have not been made.

#8 – Oil Storage of Feedwater Heat

The next four candidate concepts selected use sensible heat storage in media other than HTW. This selection features the main turbine/feedwater storage approach. Two proponents are the Bechtel Corp. which studied the possibility of retrofitting existing plants with thermal storage for ERDA (W. Stevens, Reference 6); and the EXXON Research and Engineering Co., which made an in-house study of the application of their high temperature oils to thermal storage applied to the Pressurized Water Reactor (R. Cahn, References 16,17,66). General Electric's Large Steam Turbine Division assisted in the latter on Turbine Island performance and costs. CD #25 and #21, respectively, describe their proposed configurations.

As shown in Figure 3-8, extraction steam from the various accessible extraction points is used as a source, with some live steam used to *trim* the heat exchange to oil, ie raise the temperature enough so that on discharge the feedwater produced is at the desired inlet temperature. For retrofit configurations, unless a derated boiler is available, the accessible extraction points for steam are more limited; without major modification of the main turbine, extraction cannot be increased greatly except at the cross-over (LP turbine inlet), cold reheat, and main steam points.

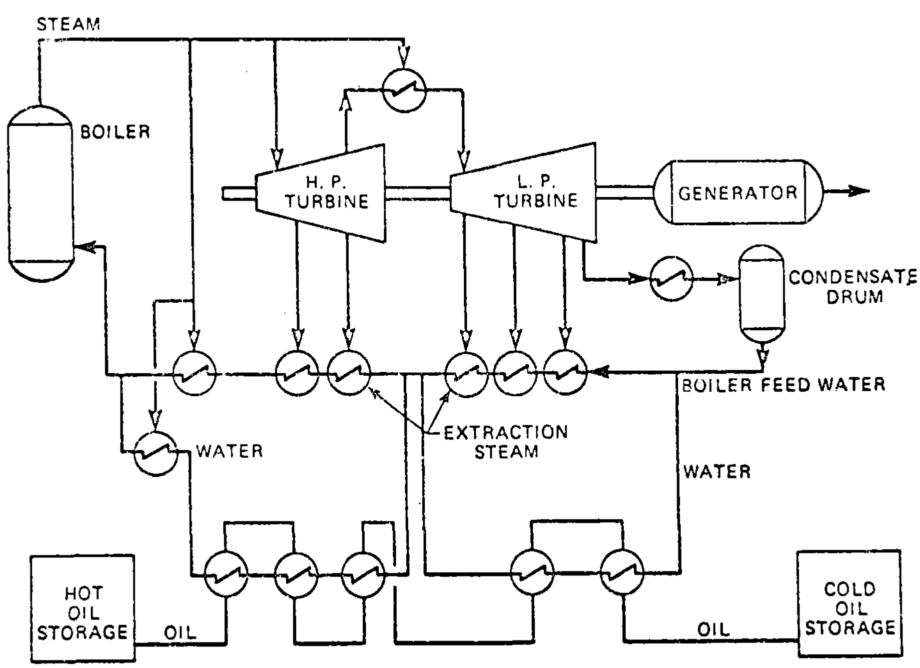
As with HTW feedwater storage concepts, to charge the storage excess steam extraction is condensed to produce an extra mass flow of a fluid, HTW or hot oil,which goes to storage. During storage discharge, the hot fluid transfers its thermal energy to heat condensate water to boiler inlet temperature; steam extraction for feedwater heat is reduced so the steam flow can produce more electricity. Heat exchang-

EXTRACTION STEAM, LIVE STEAM — LOW PRESSURE STEEL TANKS — MODIFIED MAIN TURBINE W/O STEAM EXTRACTION

HOT OIL (CALORIA HT 43) 38-270°C

PROPONENTS
BECHTEL CORP/ STEVENS
EXXON R & E CO/CAHN
CD #21,25

- CHARGE: EXTRACTION STEAM + LIVE STEAM HEAT OIL MOVING FROM COLD TANK TO HOT TANK.
- DISCHARGE: SEPARATE FEEDWATER/OIL HEAT EXCHANGER TRAIN HEATS FEEDWATER. MAIN TURBINE HAS INCREASED OUTPUT THROUGH REDUCED STEAM EXTRACTION.
- HEAT EXCHANGERS
  NON-HTW SENSIBLE HEAT STORAGE SYSTEMS REQUIRE ONE-TO-MANY HEAT EXCHANGES
  COST/EFFICIENCY/ MAINTENANCE



(Reference 66)
Figure 3-8. Selection #8.

ers are required to separate hot oil and or other sensible heat fluids from boiler quality feedwater. The heat exchanger can transfer heat from condensing steam to heat the oil directly, or an intermediate heat exchanger, ie added feedwater heater capacity, can produce HTW which is used in a heat exchanger to heat the oil. The latter course was used by both proponents. It provides some added security against oil entering the feedwater loop but imposes added capital costs.

For the retrofit case, if the plant has a steam supply at full rated output, a separate peaking turbine can be used for the added peaking capacity, since main turbine modification for oversizing is impractical as a retrofit measure.

The two proponents differ slightly in the proposed storage of oil. Exxon uses separate hot and cold tanks fully sized to each contain the full quantity of oil. Bechtel proposes more than two tanks, at least one of which is empty. By switching, when one tank's contents have been fully transferred to the empty one, it in turn becomes the empty one for a continuance of the transfer.

ADVANTAGES. Atmospheric pressure containment is a major advantage: roughly it is 35 $\$/m^3$ compared to the range from 250 to 4000 $\$/m^3$ for pressure containment. The hazards of catastrophic failure of the container are less. Pumping pressures and costs are less. Oils similar to Caloria HT43 are near-term available; they have been used as heat transfer fluids for many years.

DISADVANTAGES. Oil is more expensive than HTW. It takes about twice as many cubic meters of oil as water to store the same energy over the same temperature range. Heat exchangers required are added power related costs. Fouling of heat exchangers by degradation products of oil is a potential problem, so that periodic maintenance will be required. Oil is flammable and degrades slowly at high temperature; an inert gas cover must be provided for the oil. Leakage of oil can be a fire hazard and a pollution hazard.

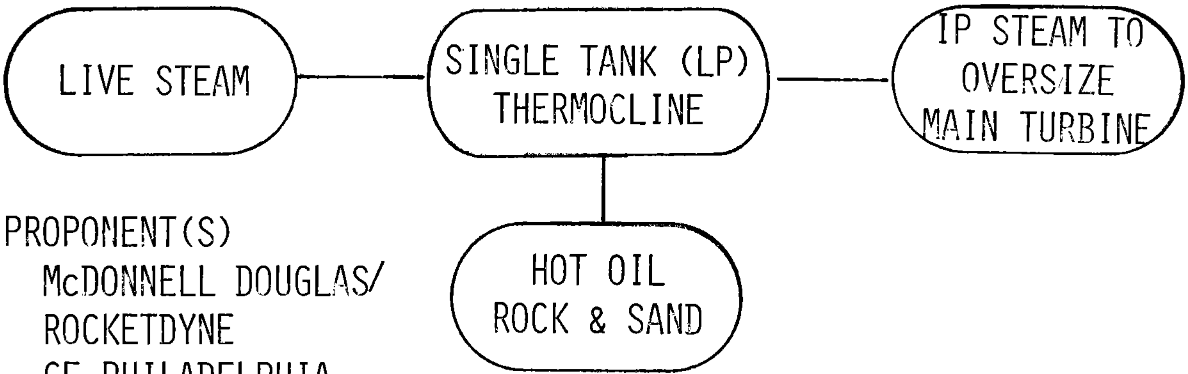
#9 — Oil and Packed Bed/Thermocline

Use of thermal storage for solar thermal applications, to condense steam from the solar receiver and to reconvert to steam for electricity generation has been examined by a number of contractors in parallel procurements. The concept proposed by the McDonnell Douglas/Rocketdyne team (CD #22; Reference 62), as well as by others, reduces the quantity of oil needed by filling the storage tank with rock and sand (Figure 3-9). Oil need only fill the voids and be the heat transfer fluid between heat exchangers and storage tanks. The tank is used in the displacement mode, ie hot oil floats on top of cold oil; in charging storage cold oil is withdrawn from the bottom and heated oil is returned at the top. A fairly sharp horizontal discontinuity, a thermocline, separates the hot oil and rock from the cold oil and rock. As the tank is charging the thermocline moves down; in discharging it moves up.

The heat exchanger configuration for charging must be designed to first remove the superheat, then condense the steam, then subcool the condensate. In discharging again three steps are to preheat the condensate, boil it (convert to steam), and then superheat the steam. Usually for design convenience each of these functions is packaged separately. In some configurations some functions are combined or deleted.

ADVANTAGES. The thermocline tank (compared to oil alone) saves tankage. The dual media storage, rock and oil, reduces the storage cost per kWh stored, as rock is much cheaper than oil per unit of energy stored. Steam generation for use in a peaking turbine avoids the maximum peaking capacity limitation of feedwater storage. Higher pressure sources (live steam and cold reheat) can be used as sources; higher pressure steam can be generated for electric production, subject to the temperature limits on the oil. Pilot size demonstrations have been made giving some confidence in near-term availability.

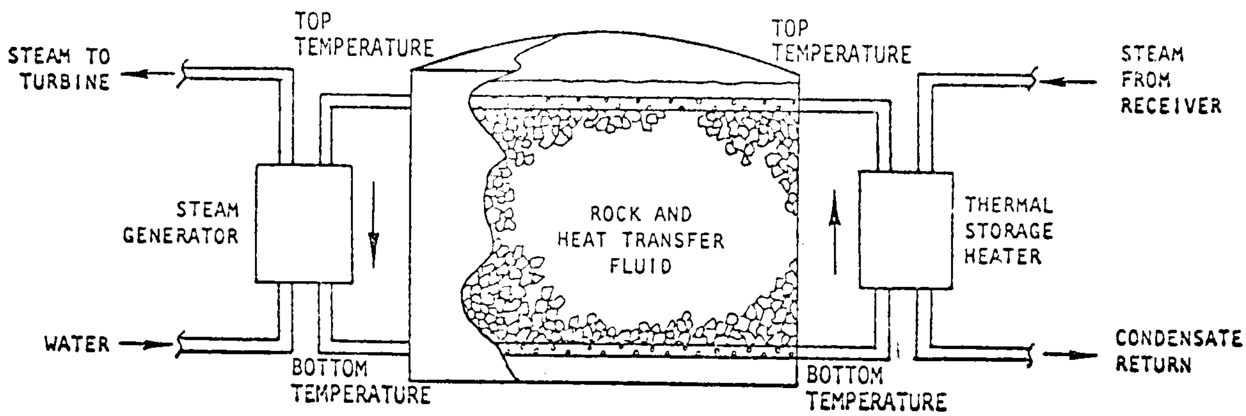
DISADVANTAGES. Some previously mentioned still apply. Heat exchanger fouling is still of concern because of reduced performance and the increased maintenance required. Flammability of oil requires precautions. Tests and demonstrations have not yet been adequate for assur-



PROPONENT(S)
McDONNELL DOUGLAS/
ROCKETDYNE
GE-PHILADELPHIA

CD #22,27

- CHARGE: LIVE STEAM, OR COLD REHEAT, DESUPERHEATED, CONDENSED IN OIL-STEAM HX, COLD OIL FROM BOTTOM OF TANK IS HEATED AND RETURNED TO TOP. IT HEATS ROCK AS IT DESCENDS, PRODUCING DESCENDING THERMOCLINE.
- DISCHARGE: REVERSE FLOW OF OIL PRODUCES RISING THERMOCLINE AS ROCK HEATS OIL. HOT OIL CHANGES FEEDWATER TO SUPERHEATED STEAM IN HX. IP STEAM TO OVERSIZED MAIN TURBINE.
- DUAL MEDIA
  USE OF LOW COST ROCK SAVES OIL. THERMOCLINE TANK SAVES TANKAGE.



(Reference 62)

Figure 3-9. Selection #9.

ance of long-term (10 to 20 years) degradation rate of the oil (requiring replacement or refurbishing), compatibility of oil with rocks of various chemical compositions, sizes, and shapes, and stresses that may be put on the tankage by the thermal cycling. This is an effect called *ratcheting*, hypothesized but not yet experienced, in which, when the tank expands more than the rock, the rock bed will settle, but not move upwards again when the tank shrinks during the next half cycle.

#10 – Oil and Salt Storage

In this concept, illustrated in Figure 3-10 and described in CD #23 and #24 (References 51,61), both hot oil and molten salt are used as storage media for different temperature ranges. Caloria HT43 is usable up to 315°C (600°F) which is adequate for the HTW subcooling and preheating, and for the condensing and boiling heat exchangers. To retain the high quality of the steam expected from the Solar Thermal receiver, both Martin Marietta and Honeywell chose to use a molten salt loop for desuperheating and superheating. HITEC (a Dupont trademark) or PARTHERM 290 (the equivalent trademark of Park Chemical Co.) is a eutectic of sodium and potassium nitrates and nitrites with a melting point of 142°C (248°F), and which is reasonably stable to temperatures over 500°C (900°F).

Figure 3-10 shows the Martin Marietta configuration for the steam generator. It comprises multiple oil tanks (seven with one empty to store hot or cold oil and to transfer between tanks during charging and discharging by use of the extra tank. The hot oil is used to preheat the feedwater to the saturation temperature, and then to convert it to steam by use of a boiler and steam drum. Saturated steam passes through a superheater to raise its temperature to 422°C at 3 MPa. A pair of molten salt tanks, one hot and one cold, supply the thermal energy for superheat. A similar set of heat exchangers is used for desuperheating, transferring heat to the molten salt, and a condenser and subcooler, transferring heat to the oil.

LIVE STEAM — HOT AND COLD TANKS FOR 2 STAGES — LIVE STEAM FOR PEAKING TURBINE

OIL/BOILER HITEC/SUPERHEAT

PROPONENTS
HONEYWELL, INC/POWELL
MARTIN MARIETTA/BLAKE

CD #51,61

- CHARGE: HITEC PASSES FROM COLD TO HOT TANK AND DESUPERHEATS LIVE STEAM. OIL PASSES FROM COLD TO HOT TANKS THROUGH SUBCOOLER AND CONDENSER. STEAM IS CONDENSED THEN SUB-COOLED BY COUNTERFLOW THROUGH THESE HXs. CONDENSATE RETURNS TO FEEDWATER.
- HITEC: THIS MOLTEN SALT HAS WIDE WORKING RANGE 200 - 500°C.
- TWO-STAGE STORAGE
  THE HIGH WORKING RANGE PERMITS USE OF SUPERHEAT FOR BETTER EFFICIENCY.

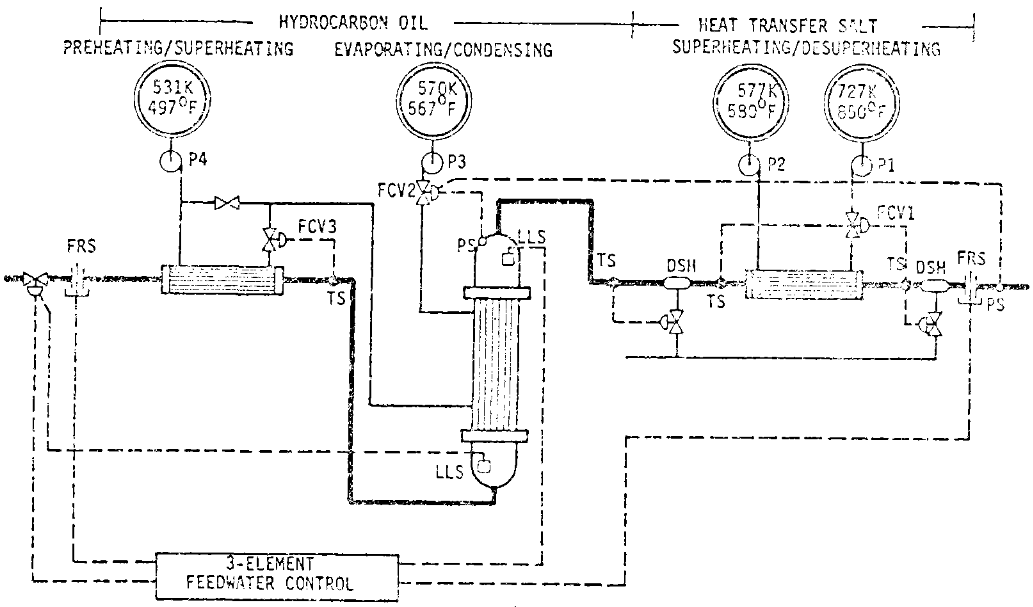


Figure V.A-1. Simplified Research Experiment Schematic.

(Reference 61)

Figure 3-10. Selection #10.

The principal difference in the Honeywell, Inc. configuration, which also uses Caloria HT43 and HITEC loops, is that the oil storage is a rock packed bed with thermocline as described for Selection #9.

ADVANTAGES. The distinctive feature, the addition of HITEC storage for superheating, can potentially improve the turnaround efficiency and improve the performance and cost of the peaking turbine system. This must be traded off against the added cost of salts, tankage, and superheater heat exchanger. Molten salts, particularly HITEC (and its other trade names) are definitely near-term available. They have been used for over 20 years as a quenching bath for heat treating, and as a heat transfer fluid in many industries. The nitrates passivate carbon steel so corrosion is not a problem below 500°C, and they can be used up to 600°C with special steels. There is little or no fouling problem below 500°C and the heat transfer coefficient is much higher than that of oil.

DISADVANTAGES. For the oil and oil/rock storage media in this concept, advantages and disadvantages are as previously described. The molten salt subsystem has its own disadvantages. While not flammable, molten nitrates are a powerful oxidizer and must not be exposed to flammable material. There is slow degradation of HITEC above 500°C that requires the maintenance of makeup, replacement, or other processing. HITEC is considerably more costly per unit of energy stored than oil (lower specific heat, higher cost per pound). One proposed way to mitigate the cost is to use HITEC and rock in a thermocline mode. While tried, there is not yet sufficient data on long-term effects of the molten salt on the rock or of rock on the molten salt to assure they are compatible. Another disadvantage of molten salts as a heat transfer fluid is the high melting point. In case of shutdown, provision must be made to trace all pipes and tanks with steam pipes or electric heaters to reestablish a flow path. One source, American Hydrotherm (Reference 1), has licensed a technology to facilitate shutdown and startup of a HITEC system by adding water at an appropriate rate during the cooling period to assure that the medium stays liquid. DuPont has technical data sheets on the use of HITEC/ water mixtures to give any desired melting point and a corresponding upper limit at which the vapor pressure exceeds one bar. It is claimed that none of these mixtures will corrode carbon steel.

#11 – All Molten Salt

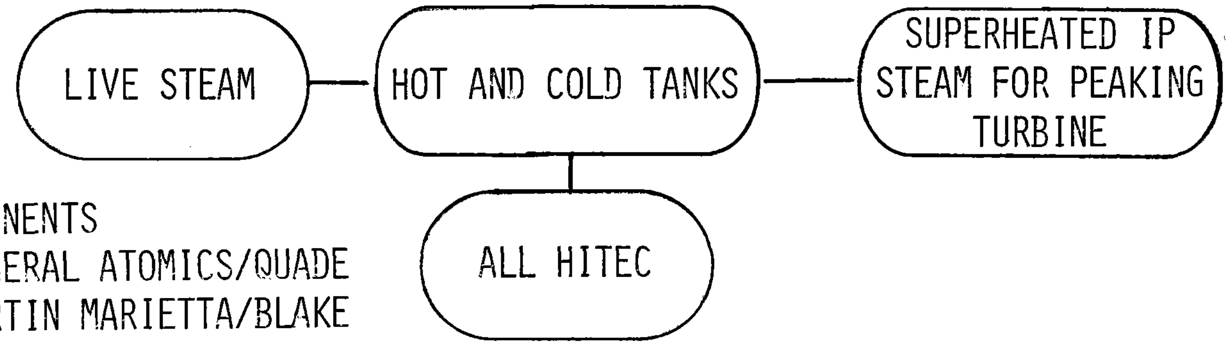
In a variant, CD #23.1 (Reference 61), Martin Marietta Corp., and its subcontractor The Georgia Institute of Technology, propose that only one medium be used: molten HITEC. Three storage tanks would be used instead of four, with the salt temperatures 238°C, 294°C, and 482°C. The configuration of tanks and heat exchangers is shown in Figure 3-11.

The lower temperature tanks are larger and use a small temperature drop for effective heat exchange between a sensible heat medium and a condenser or boiler. A fraction of the salt from the middle tank is further heated in the desuperheater, and is later used to provide superheat.

The General Atomic Co. and ORNL have also proposed all molten-salt concepts (CD #26; Reference 95). Their application is the high-temperature gas-cooled reactor, so the heat exchange to charge storage is from helium to molten salt. For storage discharge, a HITEC to water/steam steam generator is used to produce live steam and hot reheat steam.

ADVANTAGES. The basic motivation for all-salt rather than two media is simplicity. The complexity of two separate storage systems is avoided, tankage requirements are reduced, some of the salt is effectively used for the full temperature range from 238°C to 482°C, and the possible hazards from having flammable material (oil) in close proximity to strong oxidizers (nitrates) are avoided.

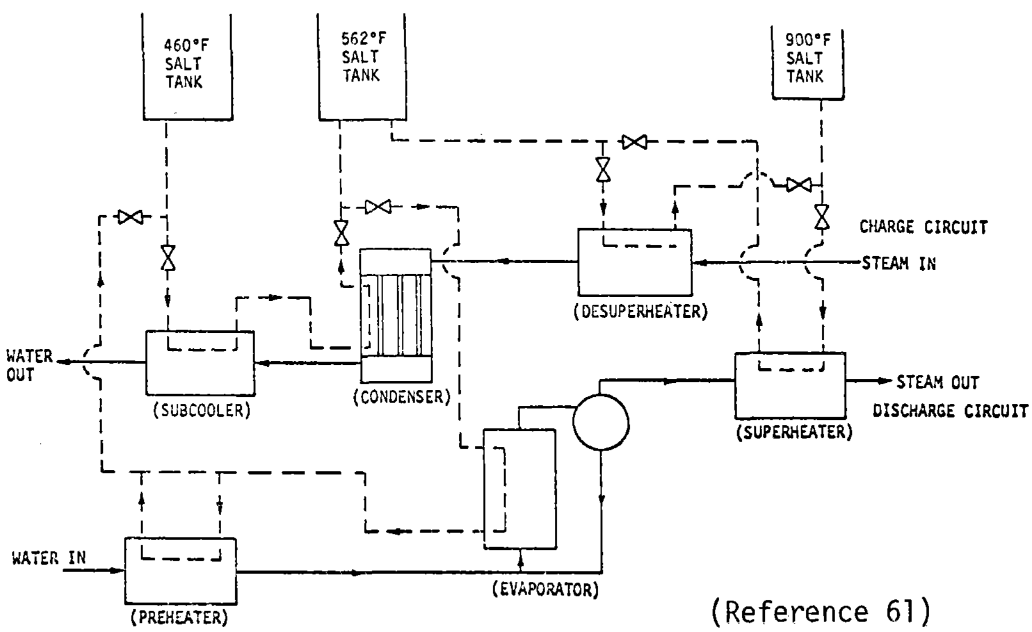
DISADVANTAGES. HITEC and Partherm 290 cost more than Caloria and far more than rock. One can conceive of salt and packed rock bed configurations with thermoclines, either to cover the full range from



PROPONENTS
GENERAL ATOMICS/QUADE
MARTIN MARIETTA/BLAKE

CD #23.1,26

- CHARGE: HITEC MOVES FROM COLD TO HOT TANKS WHILE STEAM MOVES COUNTERFLOW THROUGH TRAIN OF HX FOR DESUPERHEAT, CONDENSATION AND SUBCOOLING, TO JOIN FEEDWATER.
- DISCHARGE: REVERSE FLOW FOR HITEC FROM HOT TO COLD TANKS WHILE FEEDWATER IS PREHEATED, VAPORIZED, AND SUPERHEATED THROUGH TRAIN OF HX.
- WIDE RANGE USE
  SOME OF THE HITEC
  IS USED OVER FULL RANGE
  FROM PREHEAT TO SUPERHEAT.



(Reference 61)

Figure 3-11. Selection #11.

234 to 432°C or a large tank covering 238 to 294°C plus a smaller tank covering 294 to 432°C, but compatibility of rock and molten salts has not yet been adequately demonstrated. Other disadvantages previously listed for oil and for salt also apply.

#12 – Phase Change Materials (PCM)

Many of the references are concerned with phase change materials. CDs #41 through #51 (see Table 3-2, p 3-43) describe concepts using PCM, with various distinctive features such as the salt or other material used, and the method of heat exchange. The beneficial effect sought from PCM is either: a high energy storage density per cubic meter, because of the large heat of fusion as well as sensible heat capacity over the working temperature range; or a gain in thermodynamic efficiency by heat exchange to and from a boiling or condensing fluid (eg water) at almost constant temperature hence with high heat exchanger effectiveness and a minimum ΔT.

The latter advantage has proven difficult to achieve, not in the melting or storage-charging phase but in the freezing or storage-discharging phase. In conventional heat exchangers, the freezing material tends to build up on the heat exchange surface, so that heat exchange must include conduction through a solid layer of low thermal conductivity. In fluid to fluid heat transfer, the heat exchanger design assures adequately turbulent flow to make the film thickness limiting heat transfer very thin. A buildup of several millimeters or more of PCM reduces heat transfer by an order of magnitude, and consequently increases required area and costs.

A number of ingenious ways to minimize this problem have been proposed. Some are illustrated in Figure 3-12. CD #46 (Reference 176) by R. LeFrois of Honeywell, Inc. describes a mechanical scraper system to keep solid material from adhering to the heat exchanger tubes. Shown at the lower left of the figure is a tube surrounded by a number of scraper blades with an elliptical hole as shown by the projection. The blades are fastened to two strips; rings, one of which is a sprocket for a chain drive, hold the blades to the tube with

LIVE STEAM — A — LOW PRESSURE TANKS — A — LIVE STEAM FOR PEAKING TURBINE

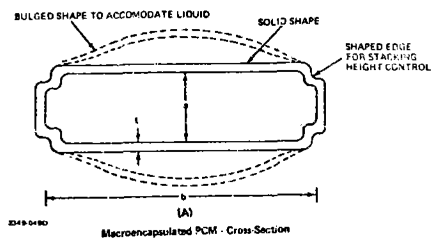
LOW PRESSURE TANKS — A — PHASE CHANGE MATERIAL $NaNO_3$ OR NaOH

A: HEAT TRANSFER FLUIDS AND HX

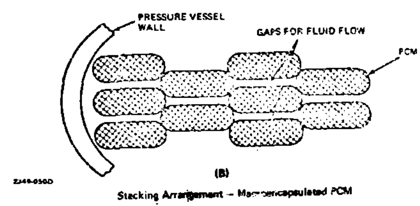
PROPONENTS
- HONEYWELL/LEFROIS
- EGFR/TAUBE
- NRL/CHUBB
- GRUMMAN/FERRARA

CD #41 THROUGH 51

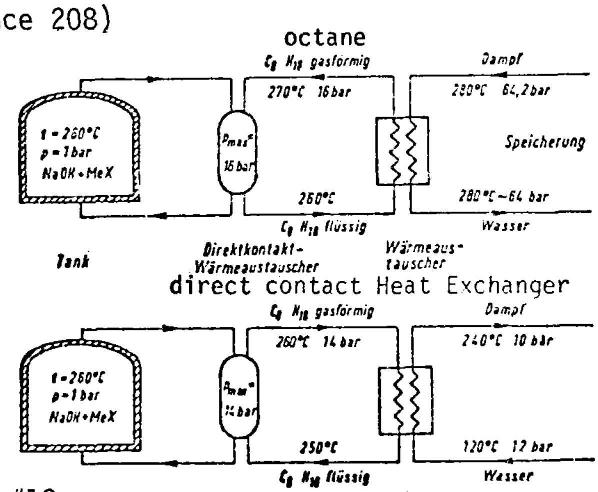
- CHARGE: LIVE STEAM IS DESUPERHEATED WITH HITEC HX. LATENT HEAT IN STEAM IS TRANSFERRED TO LATENT HEAT OF FUSION OF PCM. SUBCOOL WITH SAME HITEC MATERIAL.
- DISCHARGE: REVERSE FLOW USES HITEC FOR PREHEAT AND SUPERHEAT AND PCM FOR EVAPORATION. RESULTING STEAM CAN BE HIGHER IN PRESSURE, TEMPERATURE, AND AVAILABILITY THAN WITH ONLY SENSIBLE HEAT EXCHANGE.
- PCM – LATENT/LATENT HEAT EXCHANGE CAN IMPROVE STORAGE EFFICIENCY BUT HAS SEVERE HEAT EXCHANGER PROBLEMS. THREE PROMISING SOLUTIONS WILL BE EXAMINED FOR ECONOMIC MERIT AND TECHNICAL RISK

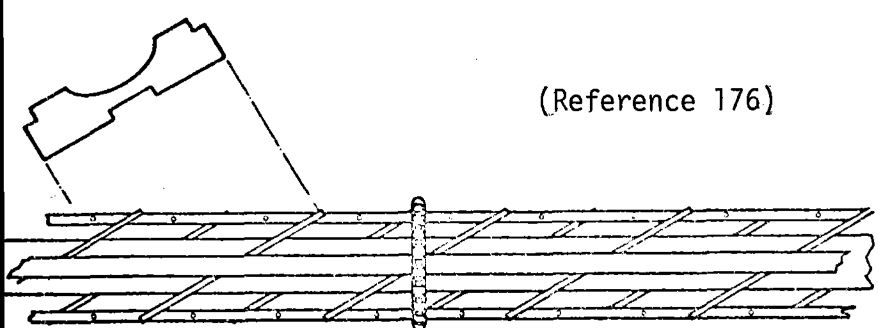


(Reference 132)



(Reference 208)





(Reference 176)

Figure 3-12. Selection #12.

a close clearance. Performance tests have been successful in showing a high heat transfer coefficient. An off-eutectic mixture of $NaNO_3$ plus a small percentage of NaOH forms a slurry which is kept circulating by the chain drive systems.

Several other approaches try to greatly increase the heat transfer area by encapsulation of the PCM. CD #48.1 (Reference 132) by Grumman Aerospace Corp. describes a macro-encapsulation of PCM, illustrated at the upper right of Figure 3-12. The PCM is contained in long plank-shaped containers of very thin wall steel. The upper sketch indicates that the horizontal faces can bulge to accommodate changes in volume with freezing and melting. The shaped notches on the sides facilitate stacking with passage space for the heat transfer fluid, as shown in the lower sketch. Grumman also mentions microencapsulation of PCM as 100 μ particles, without details on a technology to coat such particles of salt with a metal or plastic coating.

Another approach, essentially increasing the area of heat transfer, is use of a direct contact heat exchanger. The PCM, as a slurry containing 10 to 90 percent of solidified material, is placed in direct contact with an immiscible fluid. This is usually an intermediate heat transfer fluid. Prof. Taube at the Swiss Federal Institute for Reactor Research (Eidgenössischen Instituts für Reaktorforschung) has proposed a system illustrated at the lower right and described in CD #51 (Reference 208). It has been analyzed but not built. The intermediate heat transfer fluid is octane. Condensing steam boils the octane (maintained at a suitable pressure); the octane vapor is bubbled through a slurry of off-eutectic NaOH, reducing the percentage of solids in the NaOH returned to storage. For storage discharge, reduction of the octane pressure allows the slurry to vaporize octane, which condenses, generating steam in the right-hand heat exchanger.

T.A. Chubb, et al at the Naval Research Laboratory combine the use of an intermediate working fluid and macroencapsulation in the CD #42 (References 213,103). The eutectic salt is contained in small metal cans, hung or stacked in a large pressurized tank. For solar applications the solar receiver delivers hot gases which heat-exchange in the bottom of the tank with a reservoir of the heat transfer fluid, eg terphenyl, boiling it. The vapor condenses on the cans, melting the PCM, then dripping back to the reservoir. Again, by reducing the pressure and passing water through a heat exchanger at the top of the tank, terphenyl condenses, boiling the water and dropping onto the cans, where it is vaporized as the PCM solidifies.

This variety of PCM concepts is combined into one selection as a means of retaining flexibility to determine in the final selection process whether any of these concepts can be called near-term available, economically competitive with the other candidate concepts, or strongly indicated by improved turnaround efficiency or utility operating advantages.

It should be noted that the heat transfer between oil or salt and rock in a packed bed involves similar thermal conduction through a solid. The solution here is that very large heat transfer areas are achieved at low cost. The use of sand and gravel with a size not much over a centimeter in diameter, plus a very large cross section (5 to 15 m diameter) at the thermocline, and a very slow motion of a finite thickness thermocline, leads to a negligible ΔT between outside and inside of the individual particles.

ADVANTAGES. The thermodynamic loss of availability is reduced by latent-heat to latent-heat heat transfer, as compared to sensible heat transfer to boiling and condensing steam. Direct contact heat exchangers combined with latent-latent heat exchangers may be less costly than the sensible heat transfer systems previously described.

DISADVANTAGES. Because of problems of solid phase PCM either settling or freezing on heat exchange surfaces there are strong reservations that any of the concepts are near-term available. While energy storage density per kg or per m^3 may be higher than competing materials for some applications, there is great doubt that any PCM could compete in energy stored per dollar, if rock beds are found to be compatible with either oils or salts.

DISPOSITION OF OTHER CONCEPTS

The foregoing listing of twelve selected concepts for further analysis subsumes more than twelve of the listed Concept Definitions and variants in Appendix C. Some of the selections described indicate Concept Definition numbers included as variants. There are others that can be considered as minor variations subsumed by one of the twelve, or potential growth directions when they become near-term available. Some are rejected as not being directly applicable to conventional fossil and nuclear plants. Some are rejected as not as *near-term available* as those chosen. A brief review is in order to show the disposition of the Concept Definitions by inclusion in Selections 1 to 12 or by rejection. This is shown in Table 3-2.

SUMMARY

A summary table of the approved candidate concepts concludes this section. Table 3-3 indicates the distinctive feature(s) of each, and the selected basic configuration.

Table 3-3. Twelve candidate concepts — summary.

Selection Number	Feature(s)	Other Data
1	PCIV	Expansion Accumulator, 1 Evaporator
2	PCPV	Variable Pressure Accumulator, etc.
3	Steel Tanks	Displacement Accumulator, etc.
4	UG - Concrete Stress Transfer	Variable Pressure Accumulator
5	UG - Comp. Air Stress Transfer	Displacement/Feedwater Storage
6	UG - Comp. Air Stress Transfer	Displacement/3 Evaporators
7	Aquifer	Feedwater Storage
8	Oil/Feedwater Storage	Hot and Cold Tanks
9	Oil/Packed Rock Bed/Thermocline	Steam Generator, Peaking Turbine
10	Oil and Salt Loops	Steam Generator, Peaking Turbine
11	All Molten Salt	Steam Generator, Peaking Turbine
12	PCM Materials	Various Heat Exchanger Concepts

Table 3-2. Disposition of the Concept Definitions.

Concept Definition Number	Proponent	Selection Number
1 & Variants	P. Gilli - Graz Univ. of Tech.	1
2	J. Dooley - R&D Associates	4
3	A. Barnstaple - Ontario Hydro	5
3.1	A. Barnstaple - Ontario Hydro	6
4	R. Collins - Univ. of Houston	7
5	C. Meyer - General Elec. Co.	7
6	J. O'Hara - R.M. Parsons Int.	2
8	P. Margen - AB Atomenergi	5
21 & Variants	R. Cahn - EXXON R&D Co.	8
22	G. Coleman - McDonnell Douglas	9
23	F. Blake - Martin Marietta	10
23.1	F. Blake - Martin Marietta	11
24	J. Powell - Honeywell	10
25	W. Stevens - Bechtel National	8
26	R. Quade - General Atomic Co.	11
The HTGR application is not considered.		
27	E. Mehalick - General Elec. Co.	9
Trickle charge is a growth direction if shown to be superior.		
28	M. Riaz - Univ. of Minnesota	
Air heat transfer to rock beds not applicable for fossil and nuclear steam.		
30 & Variants	R. Turner - Jet Propulsion Lab.	3
31	A. Selz - Energy Conv. Eng. Co.	9,11
Sulfur not proven near-term available but may be growth potential to replace oil or salt.		
32	J. Gintz - Boeing Eng. & Const.	
Hot helium to refractory brick not applicable for fossil and nuclear steam.		
33	R. Collins - Univ. of Houston	
Hot oil in salt dome not near-term available. Problems of salt plasticity, heat exchanger fouling, workable pumping concept.		
35	W. Hausz - General Elec. Co.	9
Drained bed systems are a growth direction if shown to be superior.		
41	J. Carlson - Xerox EOS Lab.	12
Presents only merits of PCM systems, no concept.		
42 & Variants	T. Chubb - Naval Research Lab.	12
Concepts for solar applications.		
43	R. Cohen - Comstock & Westcott	12
Preliminary concept for solar application.		
45	D. Edie - Clemson University	12
Immiscible fluids HX for low temperature application.		
46	R. LeFrois - Honeywell	12
47	J. Gintz - Boeing Eng. & Const.	
PCM chosen ($T_m = 640°C$) and hot helium from solar receiver not applicable to this project.		
48	A. Ferrara - Grumman Aerospace	
Multiple PCMs at different temperatures is a combination of sensible and latent heat storage. Not believed to be near-term available or economically viable.		
48.1	A. Ferrara - Grumman Aerospace	12
Thin-walled macroencapsulation.		
49	H. Vakil - General Elec. Co.	12
Another variant of immiscible fluid HX. Application is only the HTGR.		
50	E. Clark - Rocket Research Co.	
Proposes use of sulfuric acid and water heat of reaction. Sulfuric acid is low cost sensible heat storage and heat transfer fluid if containment problems are solvable.		
51	M. Taube - Swiss Federal Inst.	

SECTION 4
REFERENCE PLANTS

SELECTION

The context for comparison of the twelve TES concepts selected during the preliminary screening includes the baseload plants — nuclear and fossil fueled — into which they are to be integrated. Selection and description of new-capacity plants for installation in the period of interest, 1985-2000, will provide a frame of reference for comparing economic, technical, environmental, and operational advantages and disadvantages of the various TESS.

The major additions to capacity during the period are expected to be a mix of LWR nuclear plants and coal-fired plants with flue gas desulfurization (FGD). The fraction of each is uncertain, as both suffer site approval, fuel escalation, and intervenor problems. Roughly equal shares is most likely. Where low sulfur coal is accessible, minimal or no FGD may be needed; this affects the economics of the fuel supply, processing, and waste disposal and has only a minor impact on the comparison of TESS. There will also be additions of gas turbine plants, combined cycles, advanced nuclear reactors, and alternative forms of storage. These are not considered as reference plants for TESS installations.

Utility planned purchases of LWR plants are mostly in the 1000-1500 MW capacity range. Planned coal-fired plants range up to 1200 MW, but most units planned by large utilities are in the 600 to 800 MW range. Smaller utilities will have need for units in the 100 to 400 MW range.

To cover this range of sizes, three reference plants on which suitable data are available have been selected. Basic data on these are given in Table 4-1. To be most useful as reference plants, not

Table 4-1. Key plant parameters – reference plants.

	Plant Number 1	2	3
Rated Output - MW_e	800	1140	225
Fuel Type	Hi Sulfur Coal	PWR	HSC
Steam Pressure at Turbine - MPa (psia)			
Superheater	24.2 (3512)	6.72 (975)	16.6 (2415)
Reheater	4.4 (637)	1.13 (164)	3.2 (491)
Steam Temperature at Turbine - °C (°F)			
Superheater	538 (1000)	284 (544)	538 (1000)
Reheater	538 (1000)	284 (544)	538 (1000)
Steam Flow Rate per Hour - 10^6 Kg (10^6 lbs)			
HP	2.64 (5.81)	6.23 (13.72)	0.73 (1.60)
IP	2.36 (5.19)	RH*.65 (1.42)	0.65 (1.44)
Net Station Heat Rate-J thermal/J electric (Btu/kWh)			
HR	2.78 (9482)	3.0 (10224)	2.86 (9750)
Thermal efficiency-percent	36	33.4	35
Condenser Pressure-kPa	5.8/8.5	8.5	11.9
(in. HgA)	(1.7/2.5)	(2.5)	(3.5)

*The reheater flow from the PWR.

only the technical data and thermodynamic performance, but also a detailed and consistent data base on the cost elements of the standard cost accounts should be available. Recent ERDA/DoE and EPRI studies by United Engineers and Constructors, Inc., Bechtel National Corp., and others have been used by these agencies as data base for computer codes (CONCEPT) and cost scenarios for utility planning purposes.

The first reference plant selected is an 800 MW high sulfur coal-burning plant as documented in NUREG-0244, Volume 3, produced by

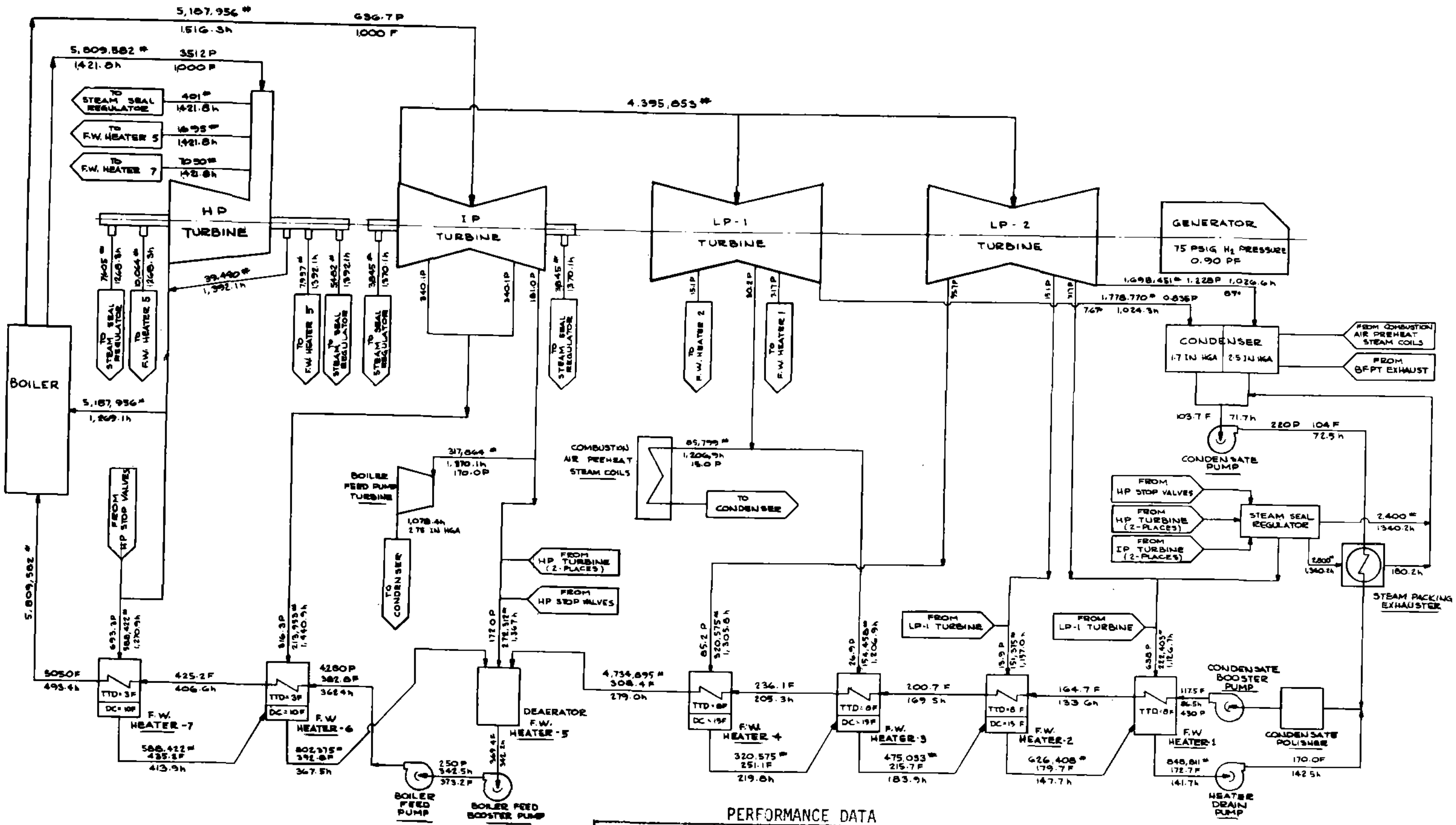
United Engineers and Constructors (Reference 212). The second is a PWR nuclear plant as documented by NUREG-0241, Volumes 1 and 2, by the same authors (Reference 93). To cover the lower end of the size range, for which no similar documentation was available, a 225-MW coal plant, for which technical data was available, was selected, and the costing was derived using the scaling laws built into the CONCEPT IV code.

Figures 4-1, 4-2, and 4-3 picture the configuration and the flows. They suffer in reproduction by reduction from the foldout size in the original documents but are adequate for a general picture. The many details of flow and heat balance of these reference plants are not relevant to what will be used in comparing TESS concepts.

ECONOMICS

The reference plants are base load plants that can produce electricity at the lowest possible cost in the time frame stated, if they are operated at their rated output power for as many hours per year as their reliability permits. They are the starting point for system concepts that modify these plants by the addition of thermal energy storage systems (TESS) and other cycle modifications as needed to give improved and economic load-following (mid-range and peaking loads).

A number of assumptions must be made, and terms and methodology defined, for understandable and consistent economic analysis of different plants and different storage system concepts in different future years. The Electric Power Research Institute (EPRI) has issued a *Technical Assessment Guide* (TAG) (Reference 172) as an aid to comparative evaluations. Its intent is to supply a consistent set of assumptions, organized in an economic methodology familiar to and accepted by electric utilities, so that studies made by different groups and contractors can be more easily compared. To the greatest extent possible the methodology and the recommended numerical parameters in this guide (TAG), revised to August 1977, are used.



PERFORMANCE DATA

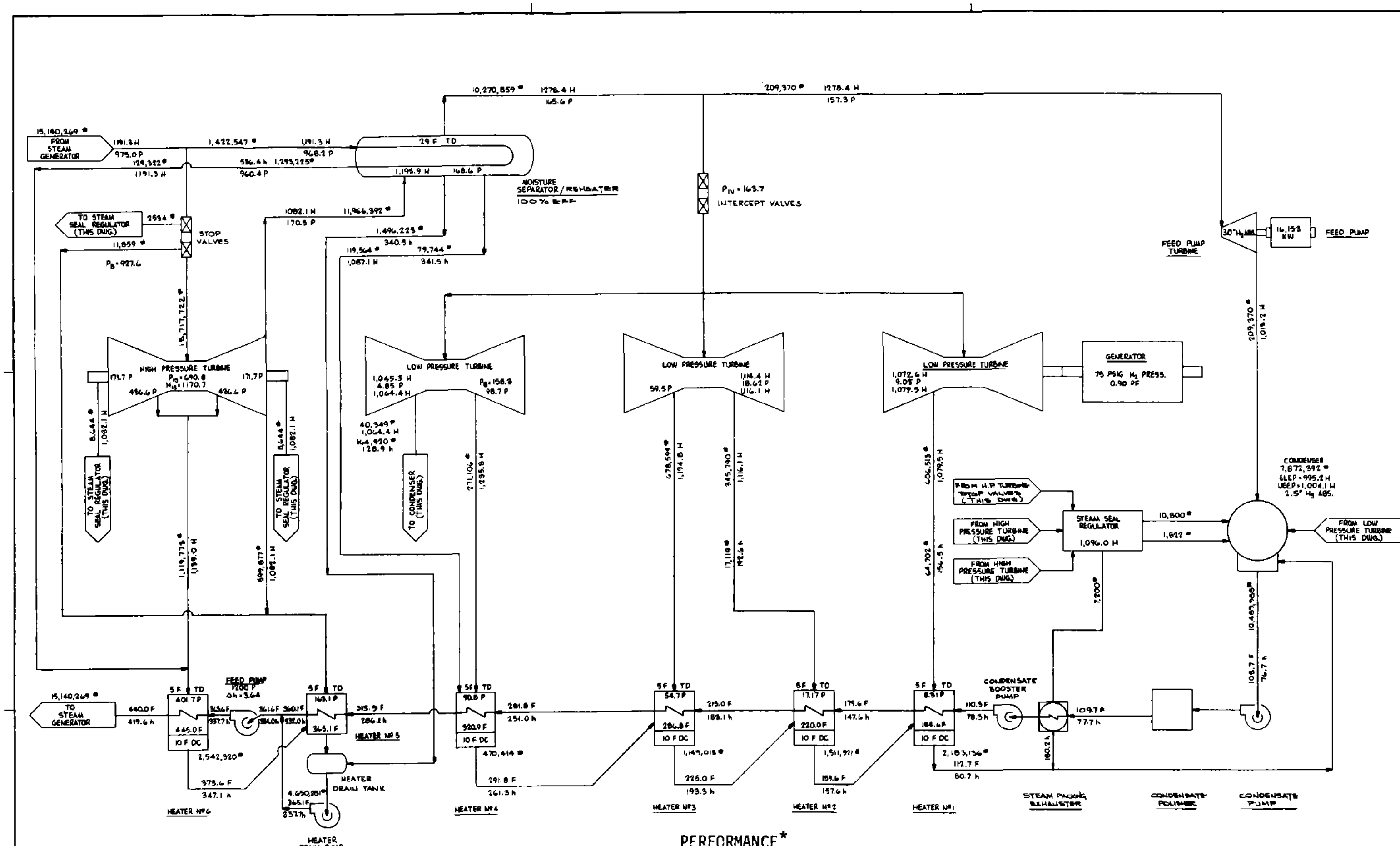
Total Turbine Output	868,867	kW
Fixed Losses	2,779	kW
Generator Losses	11,373	kW
Generator Output	854,715	kW
Auxiliary Power	60,300	kW
Net Power to Transformer	794,415	kW
Net Station Steam Rate	7.31	#/kW
Net Station Heat Rate	9,482	Btu/kW-hr
Thermal Efficiency	35.97	%

LEGEND

P - Pressure - psia
F - Temperature - Degrees/Fahrenheit
h - Enthalpy - Btu/lb
- Flow - lb/hr
% - Percent Moisture

STEAM HEAT BALANCE DIAGRAM (max. guarantee)
794 MWe FOSSIL PLANT
Middletown Hypothetical Site
UNITED ENGINEERS & CONSTRUCTORS INC.
6509.002-HSC-6

Figure 4-1. Steam heat balance diagram, 800 MW_e fossil plant.



LEGEND

H,h - Enthalpy - Btu/lb
\# - Flow - lb/hr
P - Pressure - psia
F - Temperature - Degrees Fahrenheit

PERFORMANCE*

Total Turbine Output	1,211,682 kW
Fixed Losses	5,396 kW
Generator Losses	13,886 kW
Generator Output	1,192,400 kW
Auxiliary Power	53,790 kw
Net Power to Transformer	1,138,610 kW
Net Station Steam Rate	13.3 #/kW
† Net Station Heat Rate	10,224 Btu/kW-hr
† Thermal Efficiency	33.38 %

* Based on warranted steam flow.
† Based on core thermal output of 3411 MW_t

STEAM HEAT BALANCE DIAGRAM
1139 MWe PWR PLANT
Middletown Hypothetical Site
UNITED ENGINEERS & CONSTRUCTORS INC.
6509.003-P-6

Figure 4-2. Steam heat balance diagram, 1139 MW_e PWR plant.

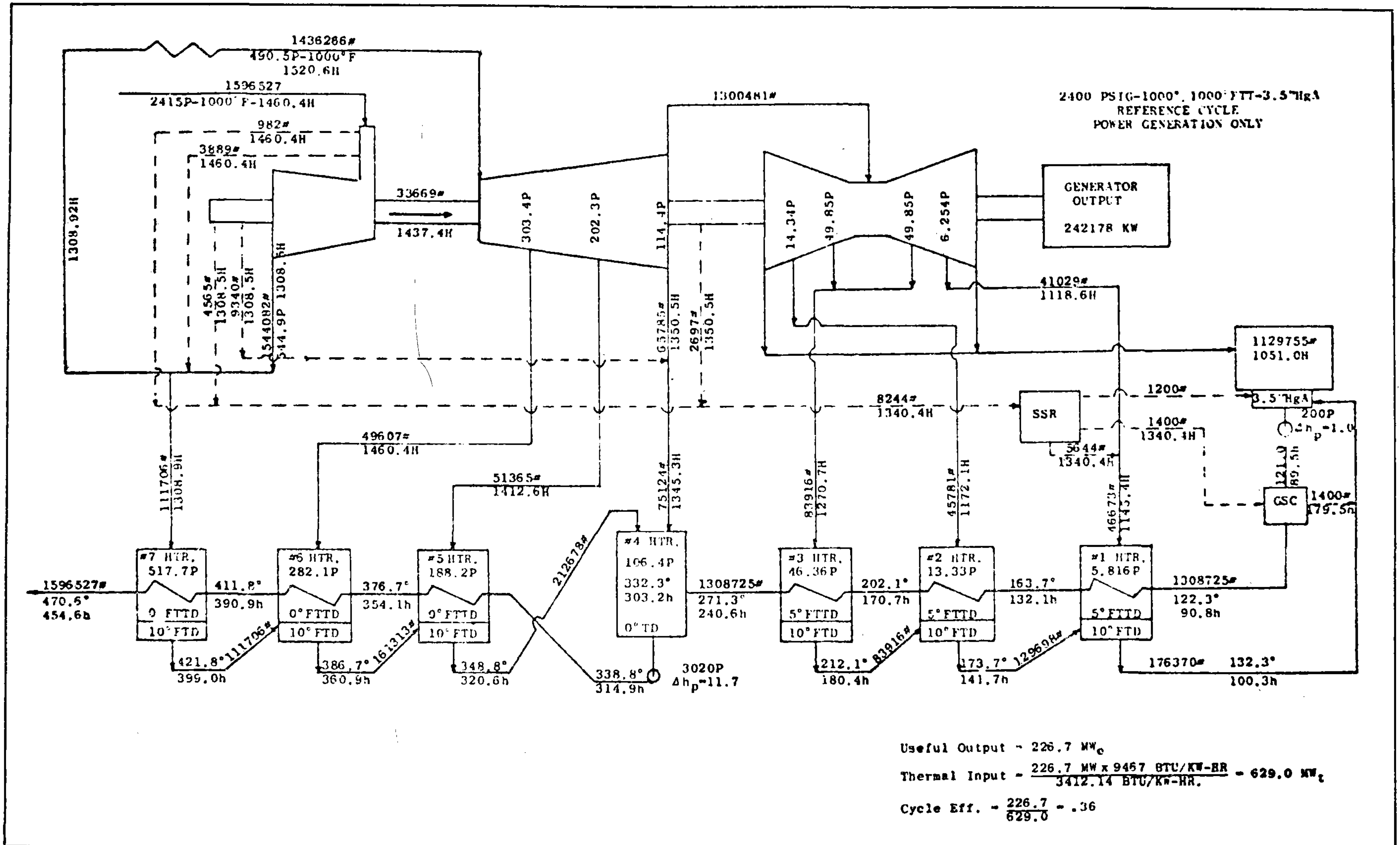


Figure 4-3. Steam heat balance, 226 MW coal-fired plant.

Some key assumptions:

- All dollar values are given in mid-1976 dollars. Future costs are expressed in 1976 dollars.
- All capital costs are assumed to escalate at a constant general inflation rate of 6 percent/annum. Compatible with this is a fixed charge rate (FCR) of 18 percent to convert capital costs into uniform annual fixed charges over a 30-year life of plant. For other equipment lifetimes an adjustment in FCR must be made.
- Fuel costs are expressed in 1976 dollars but are assumed to escalate faster than general inflation at net rates given in TAG. The fuel costs over a time period, reduced to 1976 dollars, will be higher for later dates of initial plant operation. For simplicity in this analysis 1990 is assumed as the initial operation date for all analyses.
- Single unit plants are assumed, of the capacity given in Table 4-1. The TAG prefers to give specific costs (dollars per kilowatt - $/kW) for twin units at one site, but gives relationships to find the cost of the first unit and the cost variation with plant capacity.
- As there are regional differences in costs, plants located in the East Central region are assumed, as suggested in the TAG, as roughly average for the nation.

As implied in these assumptions, the TOTAL cost of each reference plant is made compatible with the TAG by using the scaling laws given therein to convert twin-unit to single-unit costs (factor 1/0.92), and to convert the TAG reference plants at 1000 MW capacity to 800 MW for Reference Plant #1 (HSC Coal), and to 1140 MW for Reference Plant #2 (PWR). The scaling law used is exponential:

$C_{\#1} = C_{1000}\ (800/1000)^{0.85}$ for the coal-fired plant,

$C_{\#2} = C_{1000}\ (1140/1000)^{0.7}$ for the PWR. (In terms of $/kW rather than total cost in M$, the exponents would be -0.15 and -0.3 respectively.)

Cost Components of Reference Plants

Table 4-2 compares the costs of the three reference plants and illustrates the various components of the cost and levels of cost. All figures are in millions of dollars (M$) except the $/kW summary at the bottom.

Table 4-2. Cost accounts of reference plants.

#1 — HSC Coal 800 MW per UE (NUREG 0244 V3) and EPRI (TAG)
#2 — PWR 1140 MW per UE (NUREG 0241) and EPRI (TAG)
#3 — HSC 225 MW

Grouped Cost Accounts	#1 - 800 MW	#2 - 1140 MW	#3 - 225 MW
	Millions of Dollars		
20 Land	2.0	2.0	1.4
21 Structures	38.0	101.4	14.6
25 Misc. Plant	8.7	11.8	5.6
22 Steam Gen. Plant	120.1	133.4	38.3
23 Turbine Plant	65.2	111.3	20.8
24 Electric Plant	28.9	39.4	15.3
26 Heat Rej. System	12.0	21.6	4.9
A Total Direct	275.0	421.0	100.9
	x 1.22 =	x 1.35 =	x 1.3 =
B Base Cost	335.2	568.8	131.0
	x 1.77 =	x 1.57 =	x 1.5 =
C TOTAL Cost	594	894	197
direct to TOTAL	x 2.16	x 2.12	x 1.95
$/kW			
Direct Cost	343	370	448
Base Cost	419	500	583
TOTAL Cost	743	785	874

The several sources use cost accounts to indicate at a two-digit level the major cost elements or subsystems, and at a level of three or more digits the elements of the subsystems down to individual parts (eg pumps, motors, tanks) and construction materials (eg pipes, concrete, reinforcing steel). At the two-digit level, Table 4-2 presents the account numbers, the account title, and the "direct cost."

It is important to note and understand some of the terminology used in the cost accounting system. There are many echelons of costs, and serious errors in comparing concepts or systems can be made by not assuring that the costs of each are at the same echelon, with the same assumptions.

For example, plant #2 has at the lowest subaccount echelons the costs of factory equipment, the onsite labor costs, and the onsite material costs. The sum of these three is the direct cost, also often called the installed cost. Some illustrative examples of the echelons of cost accounts from Reference 93 are shown in Table 4-3.

Table 4-3. Illustrative cost breakdown of cost accounts (millions of dollars – 1976$).

Account Number		Factory	Labor	Materials	Direct Cost
231.11	Turbine Factory Cost	53.22	--	--	53.22
231.1	Turbine & Accessories	53.22	2.57	0.24	56.03
231.2	Foundations	--	1.34	0.83	2.17
231.	Turbine Generator	54.87	5.19	1.29	61.36
23.	Turbine Plant Equipt.	82.63	23.34	5.32	111.28
2.	Total Direct Costs	229.10	133.14	66.72	420.96
9.	Indirect Costs	95.92	19.45	32.50	147.87
Total Base Costs		317.02	152.59	99.22	568.83

It can be seen that some 4-digit accounts are all factory equipment cost, some are all onsite costs. The sum of all turbine and accessory accounts give a 3-digit Turbine Generator Account. To this must be added the condenser, feedwater heating equipment, and other parts of the Account 23 Turbine Plant Equipment. Adding the reactor equipment, electrical accounts, land and construction accounts, and miscellaneous gives the Account 2 Total Direct Costs. Yet to be added are the indirect costs such as home office and onsite overhead costs. Including these gives the echelon called Total Base Costs.

Sometimes a multiplier is used on factory equipment costs to give a rough estimate of direct or installed costs. It can be seen that this can vary from 1.0 to infinity. On the level of aggregation of Total Direct and Total Base Costs the multiplier of Factory Equipment Costs is 1.84 and 1.79, respectively. Similarly, since overhead costs are not directly allocable to the specific direct cost accounts, a multiplier can be used to convert a direct cost to a base cost. In this case it is 1.35, as is indicated in Table 4-2 in the 1140 MW PWR column.

Not included in the base cost as estimated by United Engineers and Constructors (References 93 and 212) are a number of cost elements that must be included to form a proper estimate of the investment required by a utility to make a plant operational. Reference 93 indicates some of these as:

- Owner's costs for consultants, site selection, etc
- Fees, permits, State and local taxes
- Spare parts
- Interest during construction (or AFDC - allowance for funds during construction)
- Contingency allowance.

The EPRI *Technical Assessment Guide*, in order to provide a complete cost estimate acceptable to utilities, and to be useful in comparing the plants they describe and other energy options being studied, include the above cost elements, but exclude certain components such as switchyards, which are common to all plants. From the TAG total cost in $/kW times the capacity in kW, a TOTAL cost in millions of dollars is found, which included the above cost elements. To couple these TOTAL *cost* estimates from EPRI to the detailed data base on the *direct cost* of plant subaccounts, a multiplier on the total direct cost is derived. It can be seen that for the three plants this multiplier does not vary widely; it is 2.16, 2.12, and 1.95, or may be conveniently called 2.1. As our interest is in converting direct costs to TOTAL* costs, base costs are not further used in this report.

*TOTAL is used to emphasize *this* sense where ambiguity with other uses of Total, meaning *sum*, is to be avoided.

FIXED PLANT. The basic concept of thermal energy storage for electric utility load-following applications is that the steam generating plant, Account 22, will operate to the maximum extent of its availability. During part of the daily or weekly demand cycle, the steam may be fully used in a turbine generator at the rated electric output of the plant. During off-peak hours, electric output is reduced and the thermal energy generated is used to charge a storage reservoir. During peak hours this stored energy is discharged to produce additional electric power in a supplemental turbine. The parts of a plant may be allocated into those not affected by the storage cycle and those which must be modified or changed in size for the load-following application. This is developed in Table 4-4. In the former category are the steam generator, land, miscellaneous equipment, and much of the structures and improvements: respectively, Accounts 22, 20, 25, and 21. The accounts which vary in proportion to the peak electric output are the Turbine Plant, Electric Plant, and Heat Rejection System (cooling towers), Accounts 23, 24, and 26. This will be called the *Turbine Island.*

A portion of 21, Structures and Improvements, represents the Turbine Generator building, housing for electrical switchgear and controls, etc, and will vary in cost with the addition of storage. An amount to account for this is subtracted from Fixed Plant (often called Boiler Island) and added to the Turbine Island.

The total of direct costs of the elements shown as Fixed Plant is 159.4, 230.7, and 56.6 million dollars for the three reference plants. By dividing by the kilowatt capacity of the plants, the direct costs in \$/kW electric are found as 199, 203, and 252. Since the Fixed Plant is dominated by the Boiler/Reactor costs it is interesting to also divide the direct cost by the rate at which fuel is consumed, in kilowatts thermal, to get the cost in $\$/kW_{th}$ shown as 70, 68, and 88.

Table 4-4. Direct cost allocation to fixed and load-following subsystems.

Accounts	#1 - 800 MW	#2 - 1140 MW	#3 - 225 MW
Fixed Plant			
20, 25, 21, 22	168.8	248.7	59.9
Less 213, 218H,J,K,etc.	- 9.4	- 18.0	- 3.3
	159.4	230.7	56.6
Turbine Island			
23, 24, 26	106.1	172.3	41.0
Plus 213, etc.	9.4	18.0	3.3
Less 25 percent of 231, 234, 24, 213	- 17.0	--	- 6.6
	98.5	190.3	37.7
23A HP Turbine Account	17.0	--	6.6
Total Direct Cost	275	421	101
TESS Plant			
Peaking Turbine	98.5 · P	190 · P	37.7 · P
$/kW (electric)			
Fixed Plant	199 (70)*	203 (68)*	252 (88)*
HP Turbine	21 (85)	--	29 (116)
Turbine Island / Peaking Turbine	123 (164)	167	167 (223)
Total Plant	343	370	448

* $/kW of thermal output.

TURBINE ISLAND. The parts of the reference plants that must match the load-following demand by drawing on thermal storage as well as the Fixed Plant steam supply include the Turbine Plant Equipment, the Electrical Plant Equipment, and the Main Condenser Heat Rejection Equipment (ie cooling towers), Accounts 23, 24, and 26. As indicated above, certain parts of Structures and Improvements (21) were deducted from Fixed Plant and are added to the Turbine Island, in Table 4-4.

What is considered as fixed in output and what is considered as load-following will of course depend on the source of heat for TES. In the case of the supercritical (24 MPa, 3500 psia) 800 MW plant and the similarly high pressured 225 MW plant (16 MPa, 2400 psia) there can be great cost in storing at or near these pressures, and considerable penalties in thermodynamic efficiency in degrading the steam to a much lower pressure for storage. An alternative source is between the high pressure and intermediate pressure turbines, where work has been obtained by passing the steam through the HP turbine before diverting some to storage. At the output of the HP turbine the pressure is 4.9 MPa (700 psia) in the 800 MW plant and 3.8 MPa (545 psi) in the 225 MW plant.

With IP turbine inlet steam as the TES source the HP turbine will be running at rated load whenever the Fixed Plant is operating. In order to be able to separate the HP turbine from the remainder of the Turbine Island, cost accounts relevant to the HP turbine were allocated in proportion to the kW (electric) output of the HP turbine. The HP turbine supplies about 25 percent of the electric output in reference plants 1 and 3. This percentage of the turbine generator (231), the feedheating system (234), the electric plant equipment (24), and the turbine bay (213) were subtracted from the Turbine Island and made a separate account: 23A HP Turbine. The condenser and heat rejection accounts are considered only related to the LP turbine.

The nuclear plant has only two turbines, considered as the equivalent of the IP and LP turbines, so no HP turbine account is separated out.

For the 800 MW plant the combined cost of the Turbine Island and the HP Turbine Account is 115.5 M$, which leads to 144 $/kW of total electric output (115.5/0.800). For the separated accounts this amounts to 123 and 21 $/kW of *total* electric output. However, since the HP turbine outputs 200 MW and the Turbine Island 600 MW, a better estimate of the HP turbine cost per unit output is 85 $/kW (17/0.200) and of the Turbine Island is 164 $/kW (98.5/0.600).

The estimate of the specific cost of the Turbine Island is useful in estimating the cost of a supplementary or peaking turbine system operating at similar temperatures and pressures, hence the bracketing of Peaking Turbine with Turbine Island in Table 4-4. If a supplementary Turbine Island is added to Plant #1 of comparable size to the original (600 MW), a first-order approximation of its cost would be 164 $\$/kW_e$. The weak assumptions in this estimate are many but are at least partially self-cancelling. If a completely separate Turbine Island were used, and the turbine used the same quality of steam as the original, the estimate would be good. However, the discharge from storage will usually be degraded in steam pressure by about 2:1, which would require higher specific costs for the condenser, cooling system, feedwater heater, and the turbine. On the other hand, if some parts of the two turbine islands are shared, gaining the economies of scale of a factor of two, the specific costs would be reduced. The electric plant equipment, the structures and improvements, and perhaps the condenser and cooling system would benefit from this effect.

The specific costs of Plant #3 are considerably higher because of its smaller size. Rather than being linear (exponent 1.0), there are "economies of scale" for the different components of cost that have exponents x from 0.3 to 0.9; the specific costs, \$/kW, then decrease with size with exponents (x - 1) from -0.7 to -0.1. The exponent for the combination of turbine plant equipment, electric plant equipment, and the heat rejection system is about 0.75, or (x - 1) is -0.25.

Annual Costs of Reference Plants

Economic comparison of plants is usually done by comparing the sum of all costs converted to uniform annual costs over the life of the equipments. Capital or investment costs are discounted forward (eg AFDC) or backward (eg replacement costs) to the date of initial operation and multiplied by a fixed charge rate (FCR) that considers the required return on equity and debt, taxes, insurance, allowable depreciation, and other factors that are capital dependent. Since practices of assigning and using the FCR differ among utilities, the recommendations in the EPRI TAG will be used. TAG recommends a FCR of 0.18 as compatible with a 6 percent annual inflation rate, 30-year plant life, and other assumptions listed therein. There are operating and maintenance costs that are fixed (independent of annual energy output) that are usually expressed as an annual amount per kW, but which can be expressed as a multiplier to the FCR or capital costs.

Table 4-5 develops the components of the annual costs for the reference plants using the recommended values. Values are taken directly or derived from the EPRI TAG, August 1977 revision.

Table 4-5. Annual costs for reference plants.

	Plant		
	#1 – 800 MW	#2 – 1140 MW	#3 – 225 MW
Capital Cost - Direct - M$	275	421	101
TOTAL - M$	594	894	197
Fixed Charge Rate	0.18	0.18	0.18
Annual Fixed O&M - M$	3.4	5.5	1.1
Annual Fixed Cost - M$	110.3	166.4	36.6
Fuel Used	HSC*	Nuclear	HSC*
1990 Price (1976$)			
$/MBtu	1.06	0.70	1.06
$/MWh	3.62	2.39	3.62
Levelizing Factor	1.959	2.482	1.959
Level - $/MWh	7.09	5.93	7.09
Availability	0.723	0.723	0.82
Heat Rate (Efficiency)	9482 (0.36)	10224 (0.334)	9750 (0.35)
Annual Fuel Cost - M$	99.9	128.3	31.9
Annual Variable O&M - M$	15.8	16.8	4.5
Annual Variable Costs - M$	115.7	145.1	36.4
Total Annual Costs - M$	226.0	311.5	73.0
Annual Cost - $/kW	283	273	324
Energy Cost - $/MWh	44.60	43.14	45.14

*4.0% S Eastern Bituminous Coal

The echelons of *costs* that were described in connection with Tables 4-2 and 4-3, from factory equipment costs to TOTAL costs must be kept in mind. Reference sources that do not clearly state their assumptions on the type of costs that are given and the basis of dollars used (eg 1976$) are difficult to compare, and can be misleading by factors of two or more. While direct costs will be used in this report in combining and comparing costs at the component and sub-account level, the analysis of investment costs and annual costs must include all the adders required to give TOTAL costs. The factors derived in Table 4-2 are used for the three plants. The annual capital charges are the total capital cost multiplied by the fixed charge rate. To this is added the annual fixed operation and maintenance cost, given in TAG in \$/kW/a, and levelized as described below. The sum is the annual fixed cost in millions of dollars. For future use on other capital costs (eg storage), fixed O&M can be expressed as a multiplier to the fixed charge rate ($594 \cdot 0.18 + 3.4 = 110.3 = 594 \cdot 0.18 \cdot 1.032$), the factor 1.032 in this case.

The other major cost components are the variable costs, principally the cost of fuel. The amount of fuel used is related to the annual output of electric energy by the heat rate (or the thermal efficiency). The cost of fuel can be expressed in metric or English units (\$/GJ or \$/MBtu) but is best stated in \$/MWh (thermal) for convenience in combining power and energy costs. The TAG gives price scenarios for nuclear fuel and coal over the time period 1975 to 2000. Coal in the East Central Region is postulated to escalate in cost at 6.8 percent/a from 0.95 \$/million Btu in 1976. With a general inflation of 6 percent assumed, this is a *net* escalation of 0.8 percent. The 1990 price in 1976\$, which would be unchanged if there were no *net* escalation, requires correction for 14 years and 0.8 percent (1.008^{14}), which gives 1.06 \$/million Btu. Conversion gives 3.62 \$/MWh; similarly, a net escalation of 2.3 percent/a for nuclear fuel gives 2.39 \$/MWh.

A coal price of 3.62 \$/MWh in 1990 and a total escalation rate of 6.8 percent/a gives a fuel cost at the end of the assumed 30-year plant life of 26 \$/MWh in 2020. Converting this escalating stream of annual fuel costs into an equivalent uniform or *levelized* stream of payments requires finding a fuel cost intermediate between the extremes that has the same present worth as the escalating stream. The August 1977 revision of TAG gives levelizing factors as a function of the total or gross escalation rate for the assumed values of 10 percent discount rate, 6 percent general inflation, and 30-year life (Reference 172, p VI-11). GE's levelizing method is lower than EPRI by one year's escalation of the quantity being levelized. Incorporating this correction, the proper levelizing factor for coal with 6.8 percent escalation (0.8 percent net escalation) from 1990 to 2020 is 1.959. For the higher net escalation rate of 2.65 percent for nuclear fuel (assumed to continue to 2020 the rate given by TAG for 1990-2000) the levelizing factor is 2.482. The levelized cost of fuels over the period is 7.09 and 5.93 $/MWh for plants #1 and #2 respectively.

The capability of each plant to produce electric energy is limited by periods of reduced output or zero output caused by scheduled mainteance or forced outages. The fraction of the maximum theoretical output that can be obtained is called the availability. Again, TAG provides recommended values based on current experience, eg 0.723 for both the 800 and 1140 MW plants. Currently, plants over 600 MW_e have significantly lower availability than small plants, in part because of immaturity of the technology.

Combining these factors with the thermal efficiency leads to the annual fuel costs to produce maximum output as limited by the availability. Variable O&M costs are given in TAG in $/MWh in 1976$. Escalating to 1990 in 1976$ by the net escalation rate for fuel and applying the same levelizing factors used for fuel gives the annual variable O&M costs shown. These plus annual fuel costs give Annual Variable Costs.

Combining Fixed and Variable Costs gives Total Annual Costs. Dividing by the number of MWh produced annually gives the specific cost of energy in $/MWh (the same as mills per kWh). The 225 MW coal plant, with a higher capital cost and a slightly lower efficiency, was credited with a higher availability, so its specific costs are almost as low as for the 800 MW coal plant. The higher capital cost per kilowatt of nuclear plants plus the higher net escalation rate for nuclear fuel gives a specific energy cost close to that for coal derived power. The net escalation factor used assumed no reprocessing.

With reprocessing and plutonium recycle, the specific energy cost for the nuclear plant would fall to 36$/MWh.

Load Following by Reference Plants

The data of Table 4-5 are for a specified capacity factor, ie the maximum availability of the plant. For lower capacity factors (ratios of actual annual energy output to the energy output if operated at rated power 8760 hours/year), there will be lower annual fuel costs but the same annual capital costs; the specific energy cost will be higher. The relationship between total annual cost and capacity factor or hours per year is linear, as shown on Figure 4-4. Data on advanced gas turbines burning oil are also shown, taken from the data in TAG. The reference plants described are intended for base load operation, that is they would be operated for close to their maximum availability, over 6000 hours per year, at rated load. Other older plants, oil/steam plants and less efficient fossil plants, have higher production costs than base load plants, and would normally be reduced in load or shut down to follow daily and seasonal load variations, rather than load-following with the reference plants.

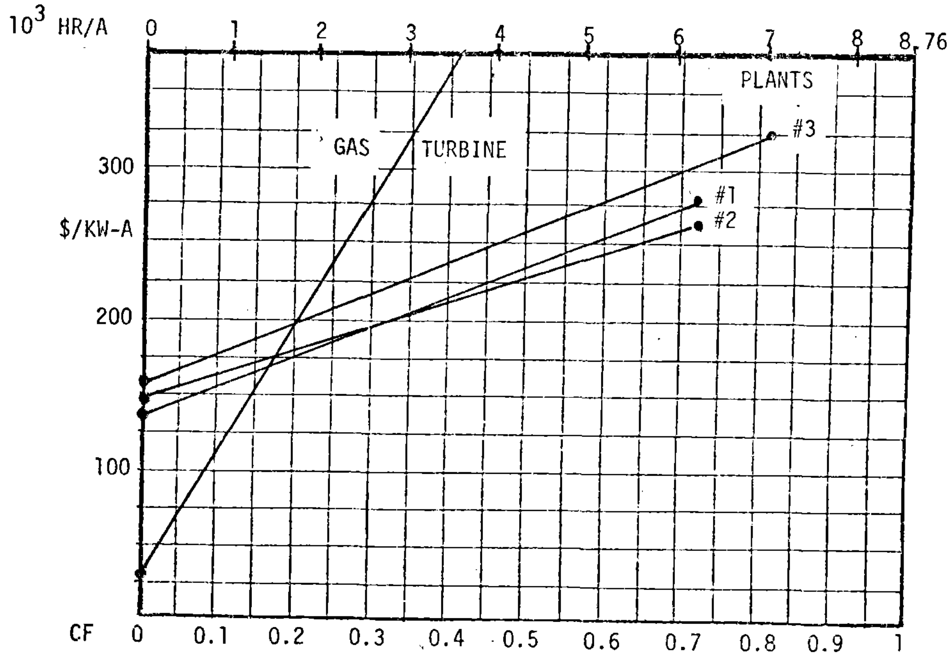


Figure 4-4. Screening curve — annual costs per kilowatt vs capacity factor or hours per year.

However, a reference plant can load-follow by throttling the flow rate and pressure into the turbine generator, hence the output of the boiler. There are economic penalties. Less annual output than the maximum available means less revenues to liquidate the annual fixed costs, hence a higher specific cost in $/MWh. Also, although the heat rate and efficiency do not change much between rated load and 80 percent load, the efficiency declines rapidly below 50 percent load. Figure 4-5 is illustrative of the change of heat rate with load for a plant similar to reference plant #1.

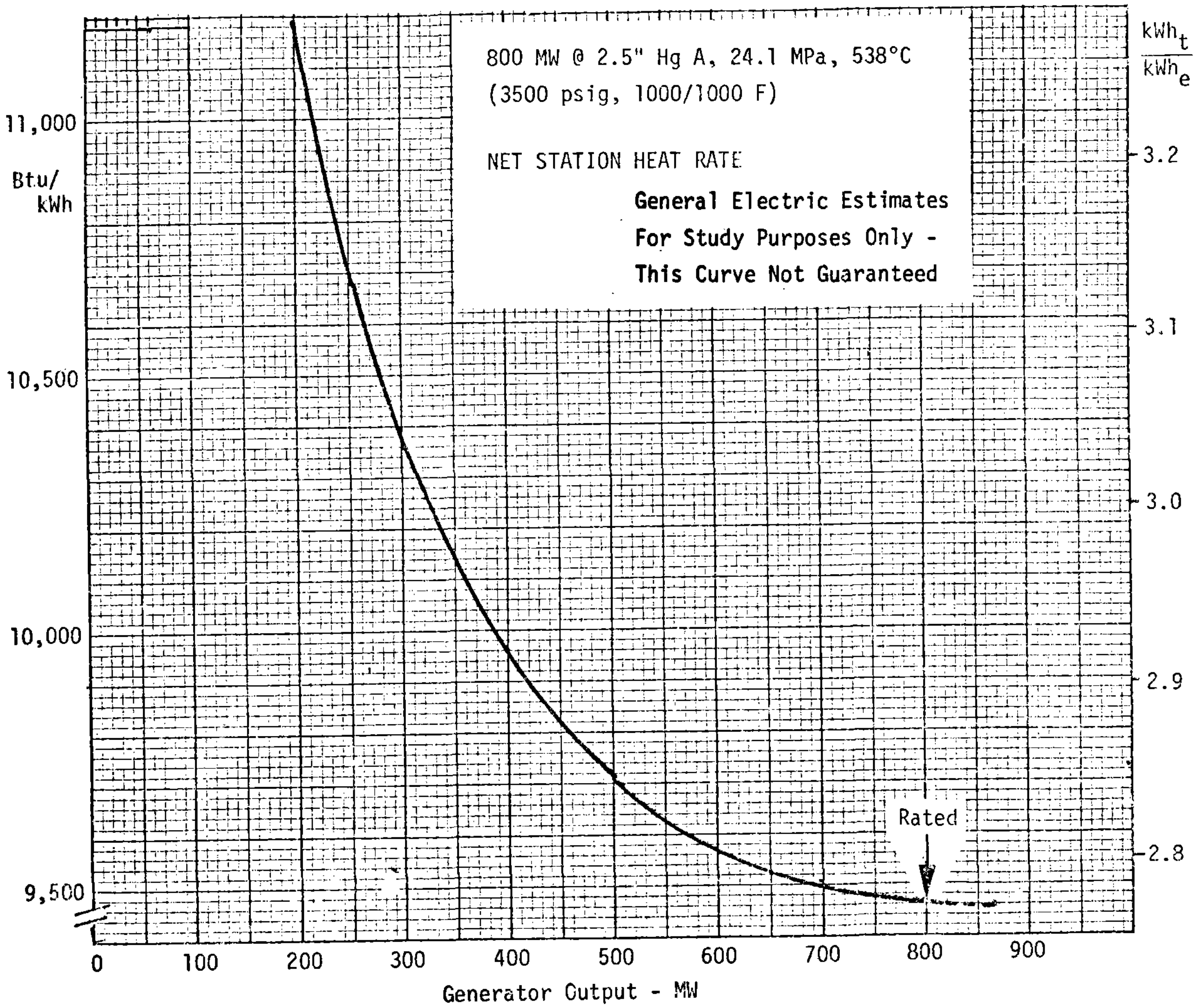


Figure 4-5. Net station heat rate versus load.

As an example of the impact of load following on efficiency, a load-following pattern that produced full, two-thirds, and one-third load for equal times (8 hours each per day) would have efficiencies relative to full load efficiency of 1.0, 0.98, and 0.89. The average daily efficiency would be 0.975, or about 2.5 percent more fuel would be needed for load following than for the same energy output at full load.

In addition to economic penalties, it has been suggested that there are less quantitative penalties associated with load following with a base load plant. These are effects on reliability and on operational flexibility. This is well founded in that excessive rates of change of temperature in the turbine can cause severe damage, and plants that are completely shut down and started up frequently have poorer reliability records than those operated at rated load. Prolonged operation at very low loads, ie below 20 percent can cause problems for nuclear reactors and for turbines. However, utilities consulted* did not seem to feel that limited load following impaired reliability if done properly, ie from say 50 percent load to full load, with temperature, pressure, and flow limited in rates of change by manufacturers' specifications and by experience. Apparently both steam supplies and turbines can change output over this range in minutes if temperature changes are not required, but it will take 10 to 24 hours to bring a large turbine up to full load conditions from a cold start.

Operational flexibility limits include these rate-of-change constraints, which may be more severe on a large base load plant than on smaller units specifically designed for cycling operation. Also supercritical plants (eg reference plant #1) and plants with flue gas desulfurization may be less amenable to load following than older subcritical plants without FGD. A number of parallel trains of FGD equipment process the stack gases, five modules in reference plant 1.

* Commonwealth Edison, Southern California Edison, Public Service Electric and Gas Co. of New Jersey, and Niagara Mohawk.

Rather than operating any of these partly loaded, one or more are shut down if plant output is reduced, and the operating problems of shut-down and startup are encountered for each major load swing.

The principal basis for comparison of thermal energy storage concepts in this study is against each other: on near-term availability, on economic criteria, and on other criteria. The economic basis for comparison is the cost of electricity produced by the TESS, in association with a reference plant, and following a specified pattern of charging and discharging the storage to match a daily load pattern. This cost comparison is dependent on both the capital cost of the components added for TESS operation and on the turnaround efficiency of the integrated system.

While the reliability and operational problems with load following reference plants must be given some weight, interest in thermal energy storage to keep the boiler or nuclear reactor at rated load while the Turbine Island load follows will depend principally on the economic advantage such storage may have. One reference value against which to compare thermal energy storage concepts is the base load reference plant used in a load-following mode.

There is one additional aspect of load-following base load plants that must be addressed before the incorporation of TESS is discussed. By definition, base load plants have low production costs through use of the lower cost fuels (nuclear and coal) and through higher efficiency than older plants. Utility dispatchers are motivated to use such base load plants to the limit of their availability. Only when there is more base load capacity in a utility system than the minimum daily load is there a motive to seek other applications for unused "off-peak power." Many or most U.S. utilities do not now have excess base load capacity. Many will not have excess base load capacity within the next 15 years. However, some utilities, eg Commonwealth Edison, have a significant fraction of their capacity in nuclear plants and find it currently worthwhile to add cycling coal-fired plants and to consider storage alternatives.

MODIFIED PLANT DESIGNS FOR TESS

The plant designs shown in Figures 4-1 to 4-3 are quite complex, including many small flows of steam from bearing and stop-valve steam seals, and to auxiliaries such as turbine driven pumps. For computer modeling there is no disadvantage to eliminating or neglecting these flows appropriately. Other simplifying changes in plant design were also made. Figure 4-6 shows the configuration used for reference plant #1.

In making changes for ease of comparison or for ease of integration of the plant with TES systems, it is desirable that:

- Changes should not affect the rank ordering of TES concepts on economic or other criteria. The changes may alter absolute values of the criteria, or modify relative values slightly.
- Changes should be generally favorable to storage, or not unfavorable.
- Changes should improve, or not handicap the near-term availability of the plant modifications required to integrate with TESS.

One of the major changes made is the elimination of the reheater between HP and IP turbines. If the source of energy for storage is to be either live steam (24.2 MPa, 538°C) or cold reheat steam (4.9 MPa, 307°C), the steam flow to the boiler reheater tubes will be decreased while the flow through the main boiler and superheater tubes remains unchanged. Operating the boiler as designed in this mode, variable flow ratio between superheater and reheater, can cause serious problems of excess reheater tube temperature, and increased forced outages. The alternatives to avoid this seem to be

- Redesign the boiler for variable flow ratios.
- Use hot reheat steam (output from the reheater) for storage instead of live steam or cold reheat.
- Eliminate the reheater, so that cold reheat or live steam can be used.

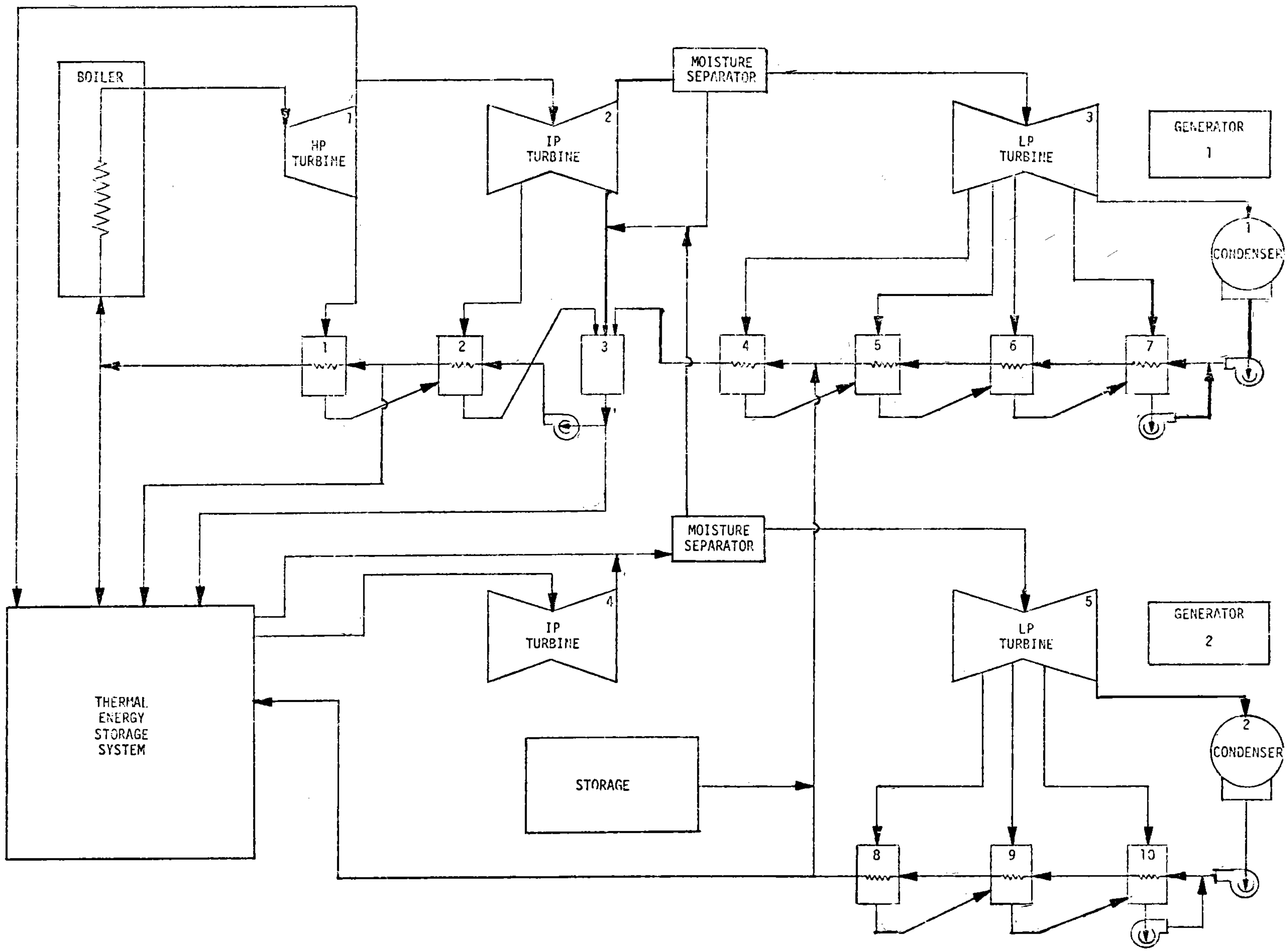


Figure 4-6. Cycle configuration for TESS fossil plant.

A telephone conversation with a leading boiler manufacturer* indicated that a conventional boiler could not tolerate more than small variations in flow ratio without danger of increased reheater tube failures, and that a new boiler could be designed to accept changes in the reheater flow by some means of damper controls to change the relative flow of hot gases and redirect energy to reheater and superheater. The total boiler thermal output would be reduced during the charging of storage with live or cold reheat steam, unless the superheater, boiler, and economizer tubes were increased in the design revision.

For the second alternative the relative effectiveness of live steam, cold reheat, and hot reheat steam as a source for storage were compared as described in Section 5. For a given swing in the initial temperature and pressure of storage to the temperature and pressure at the end of storage it was found that the turnaround efficiency ranked highest for cold reheat, next for live steam, and lowest for hot reheat. The second alternative thus does not appear attractive.

The third alternative, eliminating the reheater tubes in the steam generator has the disadvantage of also being a major change in the steam generator design. However, it is in the direction of simplicity, reduced heat, higher reliability, and known technology. It is a reversion to practices before reheat cycles were common. Per unit of heat transferred, the reheater is more expensive than the superheater and boiler tubes and more sensitive to hot spots and failures if inadequately controlled and maintained. Within the ground rules of this study, the third alternative appears most satisfactory. It is achievable in the near-term, retains flexibility to use in this study live steam or cold reheat steam, preferred for turnaround efficiency, and provides a less costly, more reliable boiler.

Elimination of reheat will increase the required flow for the same thermal output from the boiler, and will reduce the quality (increase

* Telecon with Walter Gorzegno, Foster-Wheeler Corp., 17 March 1978.

the wetness) of steam in various stages of the IP and LP turbine. Moisture separation is desirable and necessary to minimize turbine efficiency reduction and the danger of blade erosion. A moisture separator is shown between LP and IP turbine and increased moisture separation at the extraction points for feedwater heating will occur. The absence of reheat will increase the heat rate by about 5 percent, and the increase in required "back end" steam flow of almost 20 percent for the same power will increase proportionately the cost of condenser, cooling system, and feedwater heaters. The turbine cost will roughly increase in this proportion but generator and electrical costs will not increase since the output is still 800 MW_e. Simplification of the boiler by reheater omission should reduce its cost to partially cancel the added Turbine Island costs.

It may be decided that the loss in efficiency is not acceptable to utilities or that redesigning the steam generator for a variable reheater flow is simple enough to be considered near-term available, or this improvement can be considered a growth potential to be incorporated later. In any case, the changes in flow, heat rate, and costs are not sufficient to adversely affect the comparative merits of TESS concepts or of their comparison with a base load plant, providing that the cost and performance data used for the TESS comparison are both for the modified reference plant cycle.

Reference plant #2, the 1140 MW PWR, does not have three turbines in tandem, so is not considered to have a HP section. Although the reference plant diverts part of the live steam to a moisture separator/reheater in order to superheat the steam to the LP turbine section, it was decided for convenience in modeling to retain the moisture separator but eliminate the reheater. This makes the configuration of reference plant #2 the same as that for #1 except for the elimination of the HP turbine.

GE-Steam Turbine Division personnel suggested that omission of the nuclear reheat would not change the heat rate much, and that for rapid load-following the required variation of the reheat flow could present added problems of control and reliability. Within the accuracy limits of our simplified model, the heat rate is unchanged but the mass flows through the turbine and back end components are increased by 5 to 12 percent, implying some cost increase.

As with reference plant #1, the simplifying modifications should not impair comparative ranking of TESS concepts. The discussion in this section applies to the main turbine. The parallel peaking turbine and storage system modeling are covered in Section 5.

The reference plant #3 (225 MW HSC) is in general similar to reference plant #1 except in size. It is assumed to be modified in the same way: elimination of reheat, inclusion of a moisture separator, elimination of minor flows to seals and auxiliaries. Performance was not separately modeled as the principal difference expected is in the specific costs of the system because of its smaller size.

What have in this section been called reference plants, then modified plant designs, will in subsequent sections be simply called:

- Plant #1, or 800 MW coal plant
- Plant #2, or 1140 MW nuclear plant
- Plant #3, or 225 MW coal plant,

or generically, a baseline plant when a plant modified to interface with a TESS is meant.

SECTION 5
MODELING TES SYSTEMS

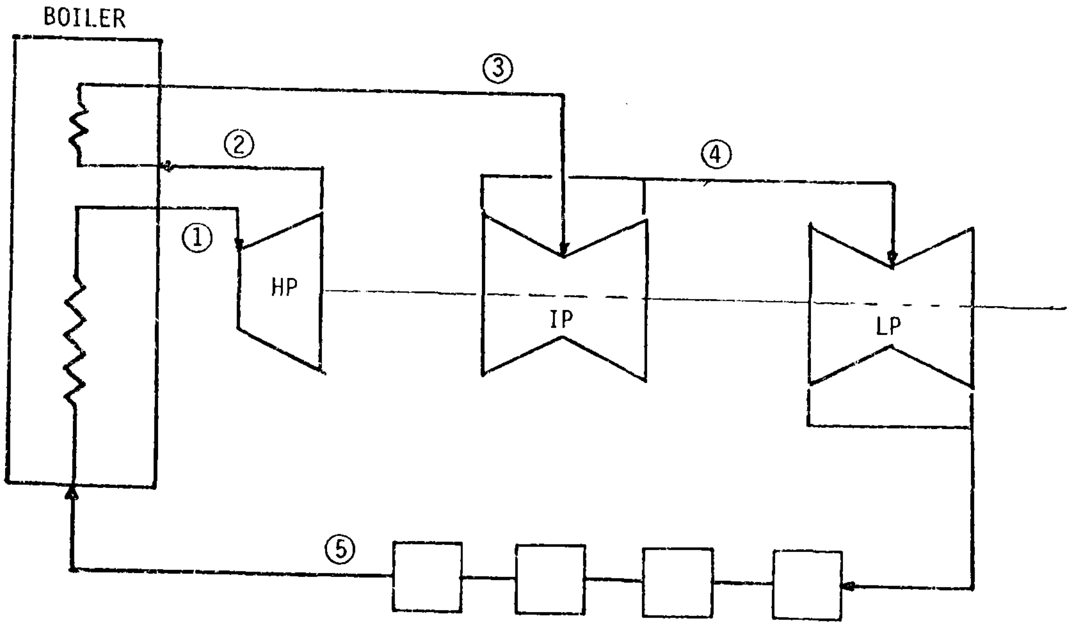
This section describes the modeling of the various TES systems necessary to provide data for the comparative evaluation. It begins by describing the thermodynamic modeling of the steam turbines, and of the individual TES concepts integrated into the baseline plants, and concludes with a discussion of the economic modeling.

TURBINE ISLAND MODELING

Plant #1 – 800 MW HSC

The two key considerations in attaching a TES system to a power plant are the source of thermal energy to charge the system and the use of the energy during discharge. Figure 5-1 shows a schematic representation of a fossil-fired plant indicating the various sources of thermal energy for charging the TES system. Any one, or a combination, of these sources may be used. During discharge the stored energy can generate steam, which provides an additional source for an oversized main turbine or powers a separate peaking turbine, or it can be used to supply heated feedwater to the boiler, thereby reducing the steam extracted from the turbine for feedwater heating.

For modeling steam generation from storage, the use of an oversize main turbine was not considered. The capacity which could be added in this way, within the current state of the art on large turbines is limited. Assuming a parallel peaking turbine permits sizing the capacity addition at anything from zero to a very large peaking swing. A small peaking turbine would simulate adequately any oversizing of the main turbine in all respects but cost, which can be addressed separately. A separate peaking turbine permits much greater operational flexibility and offers improved availability if the peaking unit can be powered directly from the boiler as well as from the TES system.



1 - Live Steam
2 - Cold Reheat Steam
3 - Hot Reheat Steam
4 - Crossover Steam
5 - Feedwater

Figure 5-1. Sources of thermal energy for charging the storage system.

Steam Generating TES Systems

From Figure 5-1 it is clear that diverting either live steam or cold reheat steam to charge the TES system reduces the flow through the reheater tubes. This creates an imbalance of flows through the boiler and requires extensive modification of the boiler and additional control equipment.

TES systems using High Temperature Water (HTW) as the storage medium store hot water at saturation pressure (or higher). Thus, charging steam must be desuperheated and condensed, generally with a spray condenser, before storing. If the steam has significant superheat, this process results in a loss of availability and, consequently, a reduction in system turnaround efficiency. For example, a crude calculation of the turnaround efficiency for reference plant #1 indicates that charging with hot reheat steam (280°C superheat) results in a turnaround efficiency about 5 percentage points lower than charging with cold reheat steam (45°C superheat). TES systems using other storage media can make beneficial use of the superheat, but generally the cost of a desuperheater heat exchanger more than offsets the benefits gained.

As related in Section 4, pp 22-26, the reheater was eliminated in plant #1 so that live steam or cold reheat (the output of the HP turbine) can be used to charge storage.

Figure 5-2 shows a simplified flow diagram of plant #1 as modeled. The main unit has 3600 rpm HP and IP turbines on one shaft and an 1800 rpm LP turbine on a separate shaft. The IP turbine is coupled to the small, high-speed, HP turbine to provide inertia and simplify over-speed control. The lower speed LP turbine is necessary to minimize bucket erosion with wet steam. In order to simplify modeling, the steam seal regulator and stop valve flows are neglected. The steam flows to the combustion air preheat coils and the boiler feed pump turbine are omitted and electrically driven feed pumps assumed.

The peaking unit, powered by steam from the TES system, uses 1800 rpm IP and LP turbines on the same shaft. Three feedwater heaters are provided, primarily to permit moisture removal from the LP turbine.

Feedwater Heating TES Systems

The simplest form of feedwater heating TES systems simply heats extra feedwater for storage as HTW during the charge cycle. During the discharge cycle the stored feedwater is pumped to the boiler inlet and an equal mass of cold feedwater removed from the feedwater heater train. Figure 5-3 shows the flow diagram appropriate for these HTW systems. In order to provide a valid basis for comparison, the basic plant layout is identical to that used for the steam generating TES systems, with the deletion of the peaking turbine and some lines.

Some of the sensible heat storage systems circulate feedwater through heat exchangers to heat an intermediate heat transfer fluid, such as oil or molten salt. During discharge the flow through the heat exchangers is reversed and the feedwater is heated by the inter-

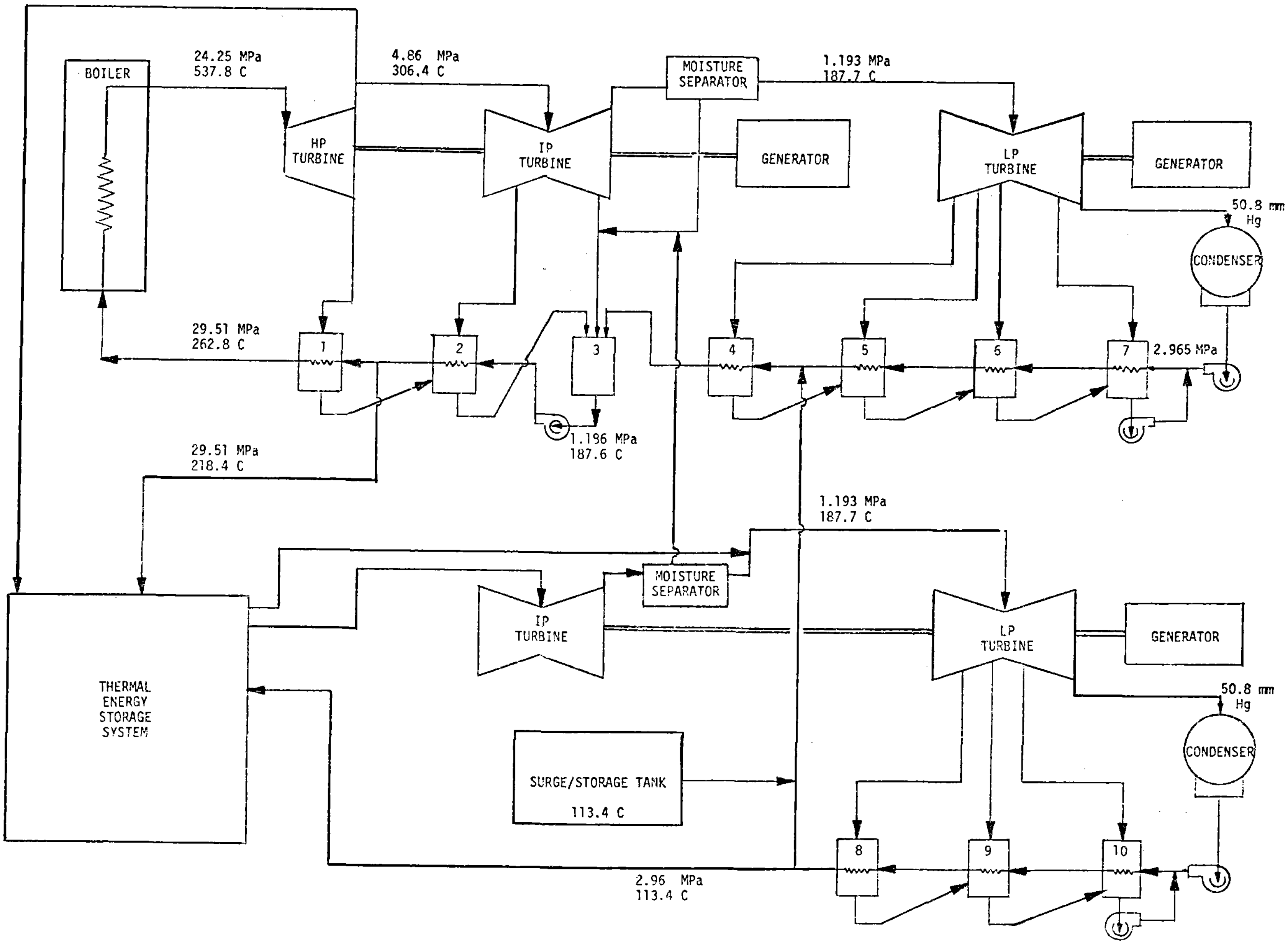


Figure 5-2. Plant #1 800 MW high sulfur coal modified for steam generating TES system.

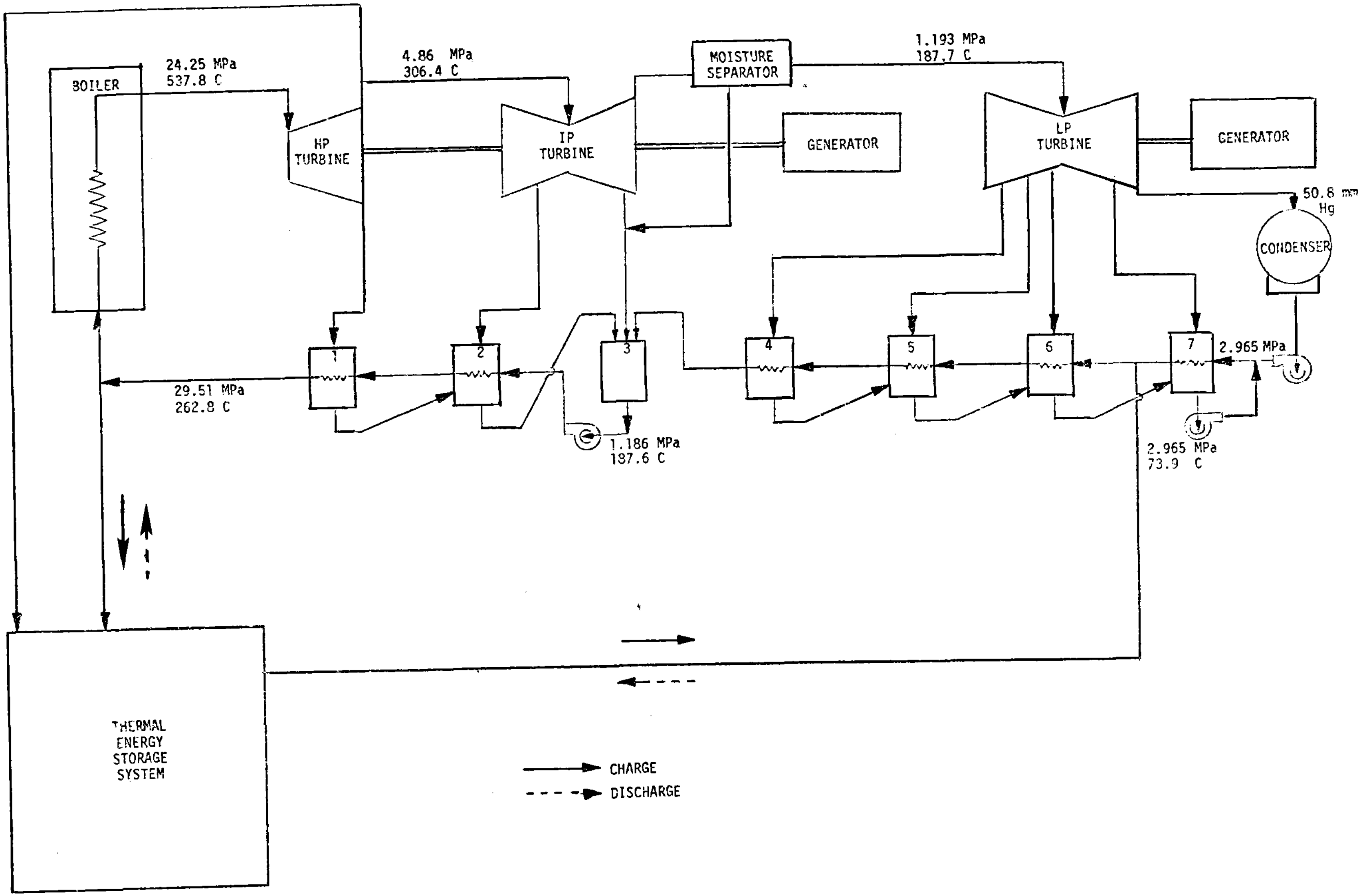


Figure 5-3. Plant #1 modified for feedwater heating TES system.

mediate fluid rather than the standard feedwater heaters. Because of the temperature drops inevitable with heat exchangers, these systems require a small amount of steam flow to heat the intermediate fluid above the final feedwater temperature. Figure 5-3 is also appropriate for this type system. An alternative approach is to use extraction steam to heat the intermediate fluid in a train of condensing heaters comparable to the feedwater heaters. A separate set of heat exchangers is required to heat the feedwater from the intermediate fluid during discharge.

Plant #2 — Nuclear

Adapting the 1140 MW nuclear plant to operate with TES systems involves much the same considerations as described for the coal plant. Of course, the major difference is the lack of high pressure superheated steam in the nuclear plant. In fact the nuclear prime steam supply is similar to the output steam from the HP turbine in plant #1.

STEAM-GENERATING TES SYSTEMS. The major modification made to the nuclear plant is the removal of the reheater preceding the LP turbine, since control of the reheater under varying loads may present difficulties. Figure 5-4 shows a simplified flow diagram of the modified nuclear plant coupled with a steam generating TES system. This diagram is essentially the same as that for plant #1 (Figure 5-2), with the omission of the HP turbine and its associated feedwater heater. This permits combining the main unit IP and LP turbines (both at 1800 rpm) on the same shaft. The peaking unit uses an identical arrangement, and is essentially the same as that shown in Figure 5-2 except that the feedwater return temperature is adjusted to match that of the nuclear plant.

FEEDWATER HEATING TES SYSTEMS. Figure 5-5 shows the flow diagram appropriate for the nuclear plant with HTW TES systems. It is also applicable for sensible heat storage systems using the feedwater to heat an intermediate fluid.

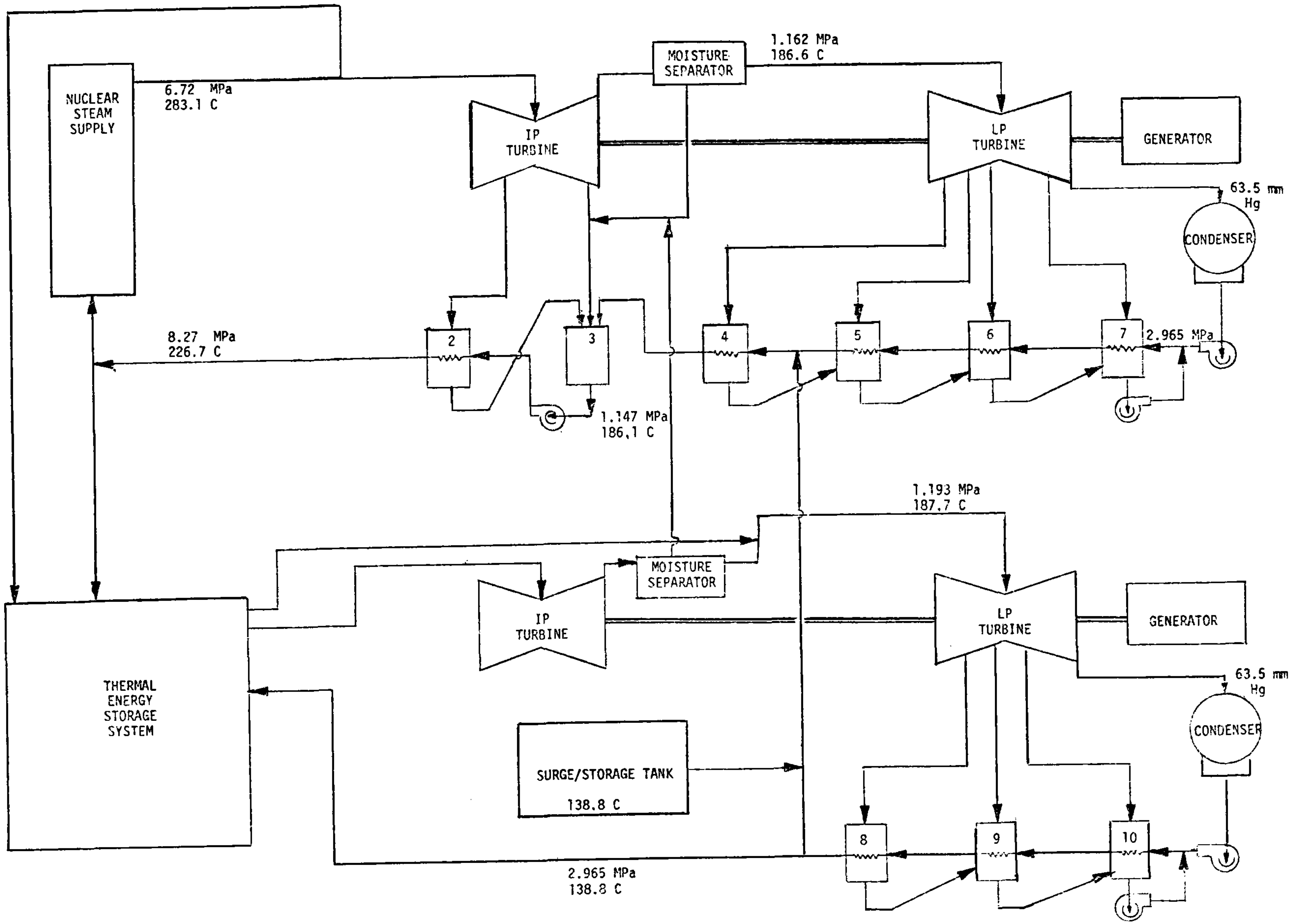


Figure 5-4. Plant #2 1140 MW nuclear plant modified for steam generating TES system.

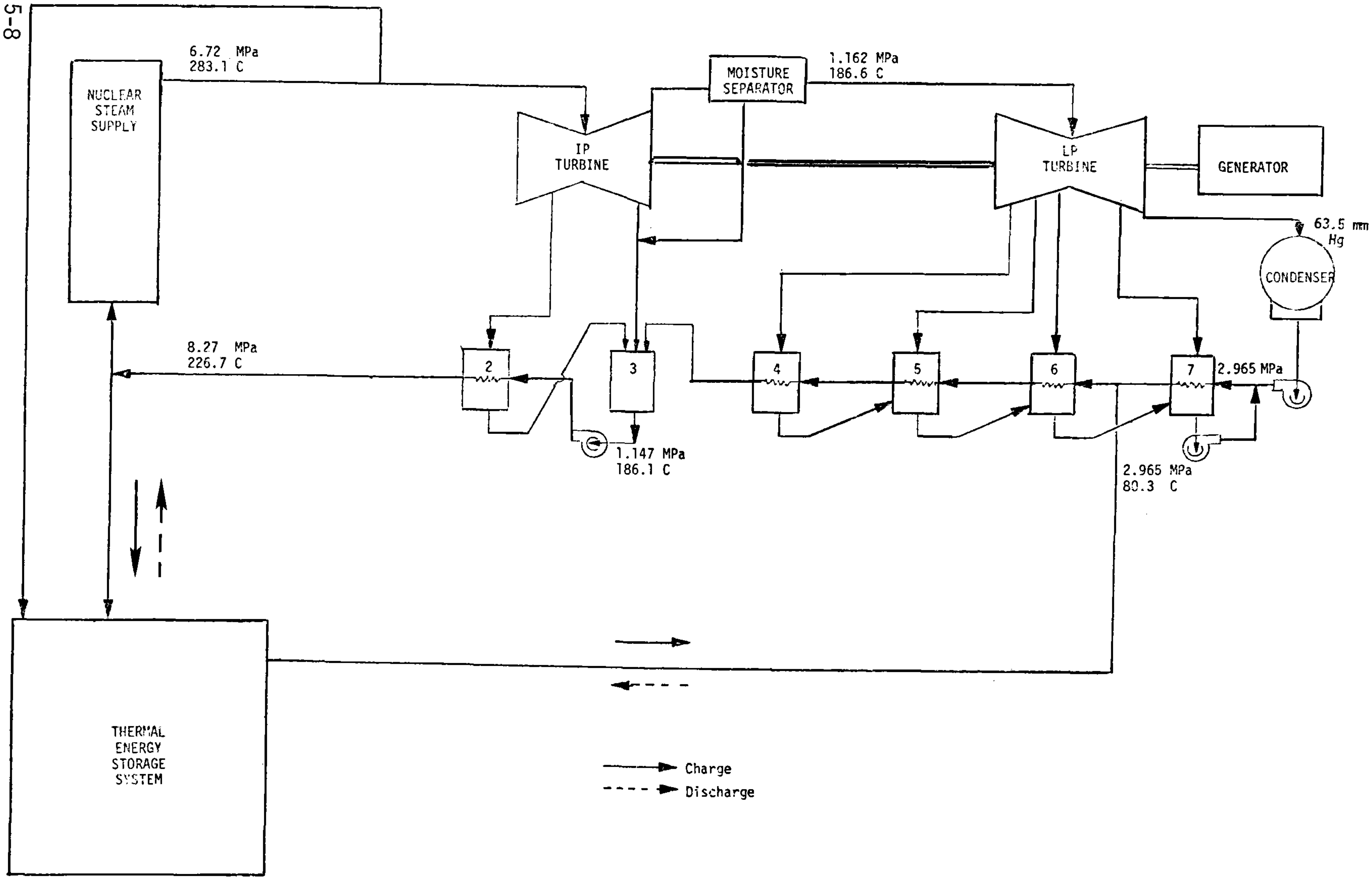


Figure 5-5. Plant #2 1140 MW nuclear plant modified for feedwater heating TES system.

Modeling Assumptions and Approximations

In order to provide the capability to rapidly evaluate the performance of the plants under various operating conditions, computer models of the four basic flow diagrams shown in Figures 5-2 through 5-5 have been developed. Each model consists of an executive program which calls individual subroutines for each of the components in the system. The component subroutines were developed by GE-Energy Technology Operation and utilize the computerized steam tables from the Large Steam Turbine Generator computer library.

Because the primary emphasis in this study is to identify the three most promising TES concepts, simple models are used. The goal is to include all phenomena that would affect the relative ranking of the various TES systems, but to omit complexities that would affect all systems equally. It is important to bear in mind that the models are not intended to duplicate existing equipment, but rather to be a reasonable representation of future equipment capability.

In implementing this philosophy, numerous assumptions and approximations are made. The most important ones relating to the turbine performance are:

- Linear expansion line, ie, enthalpy is a linear function of entropy through the expansion.
- Pressure distribution is independent of steam flow rate, therefore enthalpy at extraction ports is constant even when large quantities of steam are diverted to charge the TES system.
- Separate moisture removal at the extraction ports is not modeled.
- Turbine efficiency is constant independent of moisture content and steam flow rate. Table 5-1 lists the efficiencies assumed.
- For the main unit LP turbines the enthalpy of the output steam is increased by a leaving-loss correction to approximate the effect of steam flow rate on heat rate or cycle efficiency.

Table 5-1. Turbine efficiencies.

	Main Unit		Peaking Unit
	HP and IP Turbines	LP Turbine	IP and LP Turbines
800 MW Coal Plant	85	91.5*	85
1140 MW Nuclear Plant	80	83 *	80

* These are modified by the leaving-loss correction so that at normal rated output the effective efficiencies are 85 and 80 percent respectively.

Figure 5-6 shows the leaving-loss correction curves used for saturated steam, as a function of the exit velocity. The exit velocity is calculated from the mass flow rate of the vapor as

$$V_e = \frac{M_e v_e x_e}{3600\ A_e} \quad \text{m/s} \quad , \tag{5-1}$$

where

M_e = mass flow rate (kg/hr)

v_e = specific volume of saturated vapor (m^3/kg)

x_e = quality of steam

A_e = turbine exhaust area (m^2).

The leaving-loss correction from Figure 5-6 is then modified by an empirical relationship to account for the moisture content as

$$\Delta h_e = [0.35\ x_e + 0.65\ x_e^2]\ \Delta h_{e_{sat}} \tag{5-2}$$

The moisture separators are assumed to remove all of the moisture and put out saturated steam. For plant #1 the separator input steam contains only 4 percent moisture so that separator could probably be eliminated with negligible effect.

- The condenser pressures are assumed constant, independent of steam flow. This implies a variable coolant flow rate as the

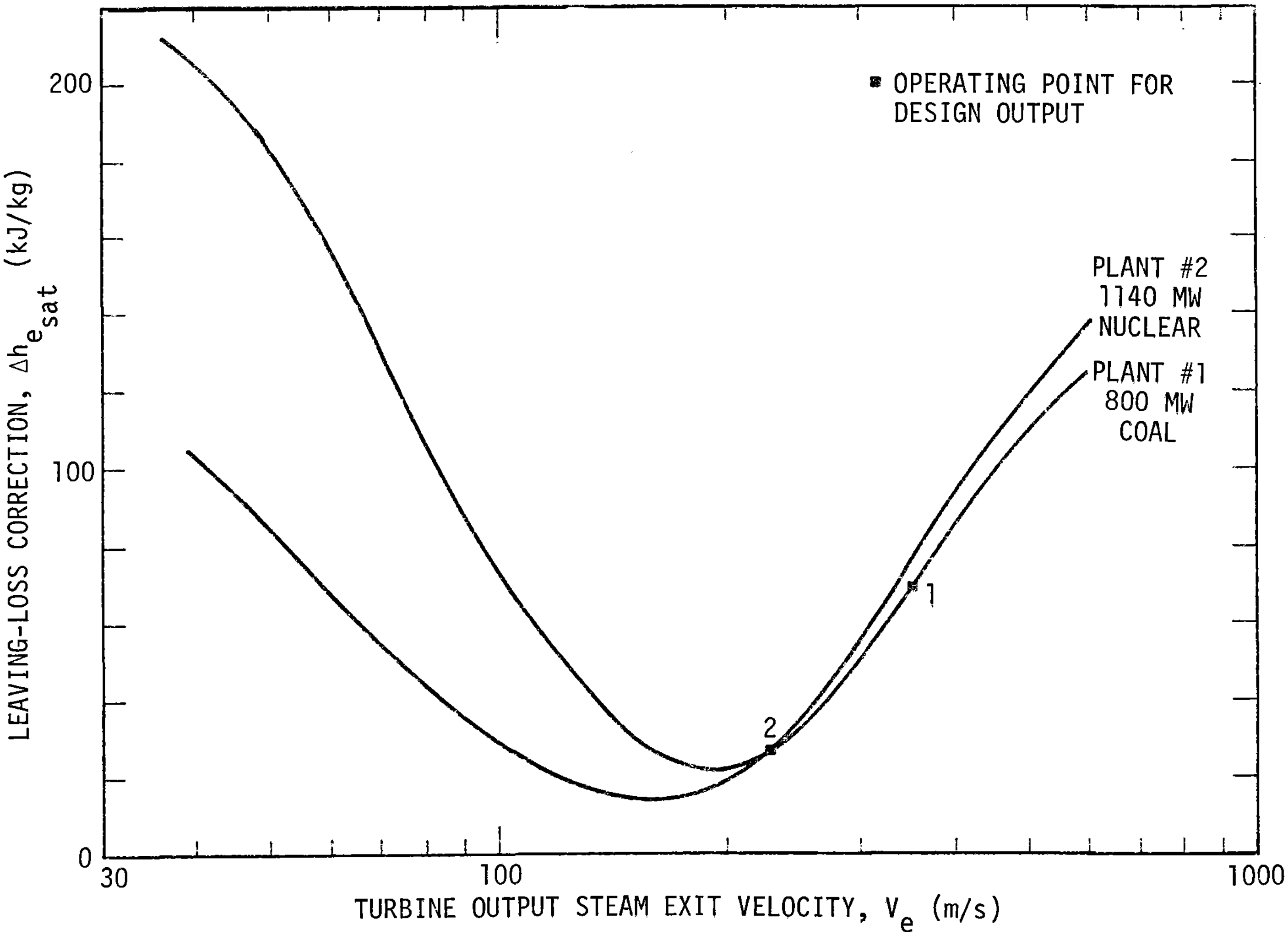


Figure 5-6. Leaving-loss correction for saturated steam.

heat rejection requirements vary. However, auxiliary power requirements for the cooling system are neglected.

- Pressure drops in the system are assumed to occur at discrete locations – at moisture separators, deaerators, and at the steam supply system.
- The feedwater pumps are assumed to be 65 percent efficient and all other pumps 60 percent. The generator efficiencies are taken as 98.7 percent.

Performance Estimates for Plant #1

For the coal plant shown in Figure 5-2, the boiler produces 3.09 x 10^6 kg/hr (6.81 million lb/hr) of supercritical steam at 24.25 MPa (3512 psia). During normal operation of the main unit (TES system inactive) the gross plant output is 849 MW, with a net output of 800 MW. The condenser heat rejection rate is about 1.03 GW_{th} (3.52 x 10^9 Btu/hr).

Because the various TES systems differ greatly in the combinations of steam and feedwater required for charging, the plant output during the TES charge cycle is different for each one. However, a typical example is useful at this stage. The sensible heat, steam-generating TES systems divert intermediate pressure (IP) steam from the input of the IP turbine, condense and cool it, and pump the condensate back to the inlet of the high pressure feedwater heater. The HP turbine and its associated feedwater heater are thus unaffected by the charging operation. The maximum charge rate is determined by the minimum allowable flow through the IP and LP turbines. For this analysis it is assumed that the minimum flow to the condenser is about 20 percent of the normal design flow. Figure 5-7 shows the net output and condenser heat rejection as a function of the steam flow rate into the TES system. Note that the minimum output is about 385 MW, or 48 percent of design output. The HP turbine accounts for about 300 MW, independent of charging rate, with the remainder coming from the IP and LP turbines.

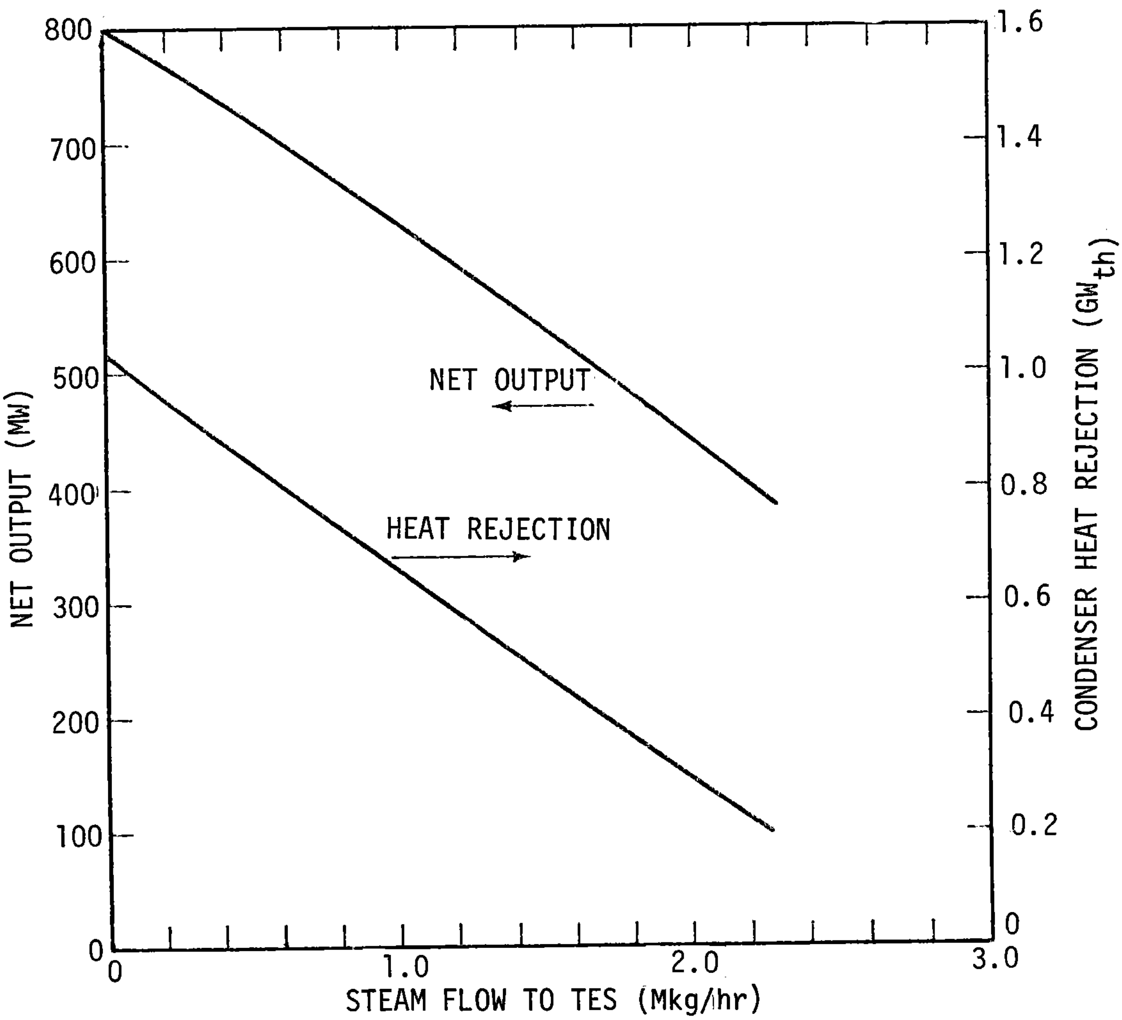


Figure 5-7. Performance of plant #1 during TES charge cycle.

Performance Estimates for Plant #2

The nuclear steam supply of the modified nuclear plant shown in Figure 5-4 produces 6.80 x 10^6 kg/hr (15 million lb/hr) of saturated steam at 6.72 MPa (975 psia). During normal operation of the main unit this produces 1166 MW gross output and 1133 MW net output. The condenser heat rejection rate is about 2.23 GW_{th} (7.61 x 10^9 Btu/hr).

To charge the sensible-heat steam-generating TES systems, live steam is diverted from the nuclear steam supply (NSS) outlet, condensed and cooled, then pumped to the NSS inlet. Figure 5-8 shows the net output and condenser heat rejection as a function of the charge steam

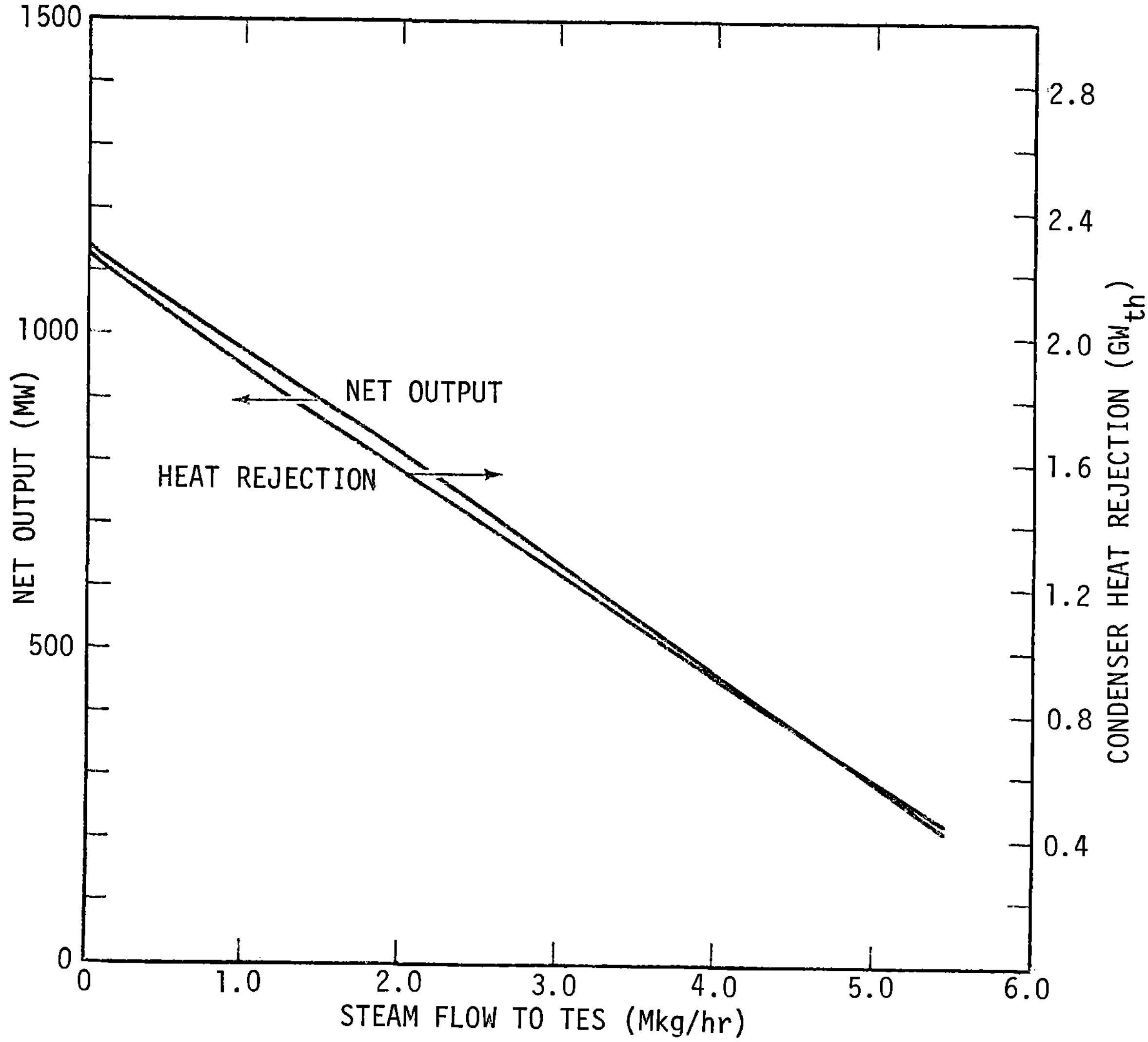


Figure 5-8. Performance of plant #2 during TES charge cycle.

flow rate. Note that the minimum output is about 216 MW (corresponding to a condenser flow about 20 percent of design flow). The nuclear plant thus gives a much larger downward power swing than obtained with the coal plant because the steam flow through all the turbines is reduced.

HIGH TEMPERATURE WATER TES MODELING

The proposed HTW system concepts all store water under adequate pressure to prevent vaporization. They differ only in the design of the containment vessel and the method of operating it. The design of the containment vessel essentially influences only the thermal losses during storage and the auxiliary power requirements. Since all methods of containment can be designed to lose less than one percent of the energy stored, thermal losses are neglected in the modeling. The auxiliary power requirements may differ somewhat depending on whether the vessel is located underground or on the surface. The density of steam is so small (about 1 lb/ft^3 or 16 kg/m^3) that this difference can be safely ignored for systems that transport steam in and out of underground storage vessels. For systems that transport water the auxiliary power may be significant. However, it is neglected here on the assumption that any power used in removing water from storage can be recovered from the water injected into the storage, with the exception of pumping losses.

This leaves the method of operating the accumulator as the major difference among the candidate TES systems. For steam generating systems all three accumulator modes (ie, variable pressure, expansion, and displacement) are appropriate. For feedwater storage systems, no steam is wanted, and the temperature and pressure of the HTW discharged should remain constant unless some steam extraction is used for trimming between storage and the boiler inlet. The displacement mode would seem most appropriate if the means of containment is suitable for this mode. The expansion mode would require a large supplementary storage for cold feedwater.

There are numerous design parameters that affect the performance and cost of a combined power plant with a TES system attached. Although it is likely that an "optimum" (eg, minimum cost) set of parameters exists for each combination of power plant and TES system, no attempt is made to determine these optimum designs. Rather, one plant-TES combination (the 800 MW coal plant with a Variable Pressure accumulator storage system) is selected for sensitivity analyses with the major design parameters. A "good" set of parameter values is chosen as the base case to be evaluated for all other system configurations.

There are also numerous performance indices that can be used to describe the various systems. For convenience in later work (and hopefully, also for clarity) the "turnaround efficiency" and "specific output" are chosen as the primary measures of performance. Turnaround efficiency is simply the ratio of the peaking electrical energy generated during the discharge cycle to the reduction of electrical energy during the charge cycle. For these analyses, where constant power generation is assumed during each cycle, this becomes simply

$$\eta = \frac{(P_d - P_n)t_d}{(P_n - P_c)t_c} \quad , \tag{5-3}$$

where

P_d = power generation during discharge cycle (MW)
P_c = power generation during charge cycle (MW)
P_n = power generation in normal operation (TES system inactive)
t_d = discharge time (hr)
t_c = charge time (hr).

Specific output is the ratio of the total electrical energy generated during the discharge cycle to the total volume of storage required to produce it, or

$$e_0 = \frac{(P_d - P_n)t_d}{V_s} \cdot 10^3 \ \text{kWh/m}^3 \quad , \tag{5-4}$$

where V_s is the storage volume in m^3.

<u>Variable Pressure Accumulator — Plant #1</u>

Figure 2-3 shows a schematic diagram of a variable pressure accumulator. When fully charged the cushion of saturated steam is a few percent of the total volume. During discharge steam is drawn from the top, reducing the pressure in the vessel and causing some of the HTW to flash to steam to restore equilibrium conditions. The temperature and pressure in the vessel thus decrease steadily throughout the discharge cycle. The throttle in the output line is necessary to control the rate of steam generation and to provide steam to the turbine at a constant pressure.

Recharging the accumulator is essentially the reverse process. In order to return to the same conditions existing before the discharge, the mass and total enthalpy added must equal the mass and total enthalpy removed. When charging with superheated steam from the coal plant it is necessary to mix in a small amount of feedwater to obtain the balance. Charging with saturated steam from the nuclear plant requires removing a small amount of the stored water. The throttle in the input line is simply to control the rate of charge.

Since the variable pressure accumulator is a non-equilibrium thermodynamic process, it is modeled by assuming equilibrium processes are valid for small changes in the storage pressure and temperature. Thus the accumulator performance during discharge is evaluated by the iterative computational procedure diagrammed in Figure 5-9. Figure 5-10 shows a typical discharge cycle for an initial storage pressure of 4.65 MPa (675 psia).

During recharge the input steam is assumed to have a constant specific enthalpy, so the model is much simpler. The differences in mass and total enthalpy between the charged and discharged states are calculated, thereby determining the specific enthalpy required in the input steam. The enthalpy of the charging steam from plant #1 exceeds the requirement, so the amount of feedwater to be mixed with the charging steam is calculated. The saturated steam from plant #2 does not meet the required specific enthalpy, so some HTW must be removed from the accumulator. The most efficient procedure would be to remove the HTW before recharging. However, the amount is so small that the overall TES system performance is not significantly affected by the technique chosen. For convenience the HTW is removed continuously during the charging and returned to the inlet of the NSS.

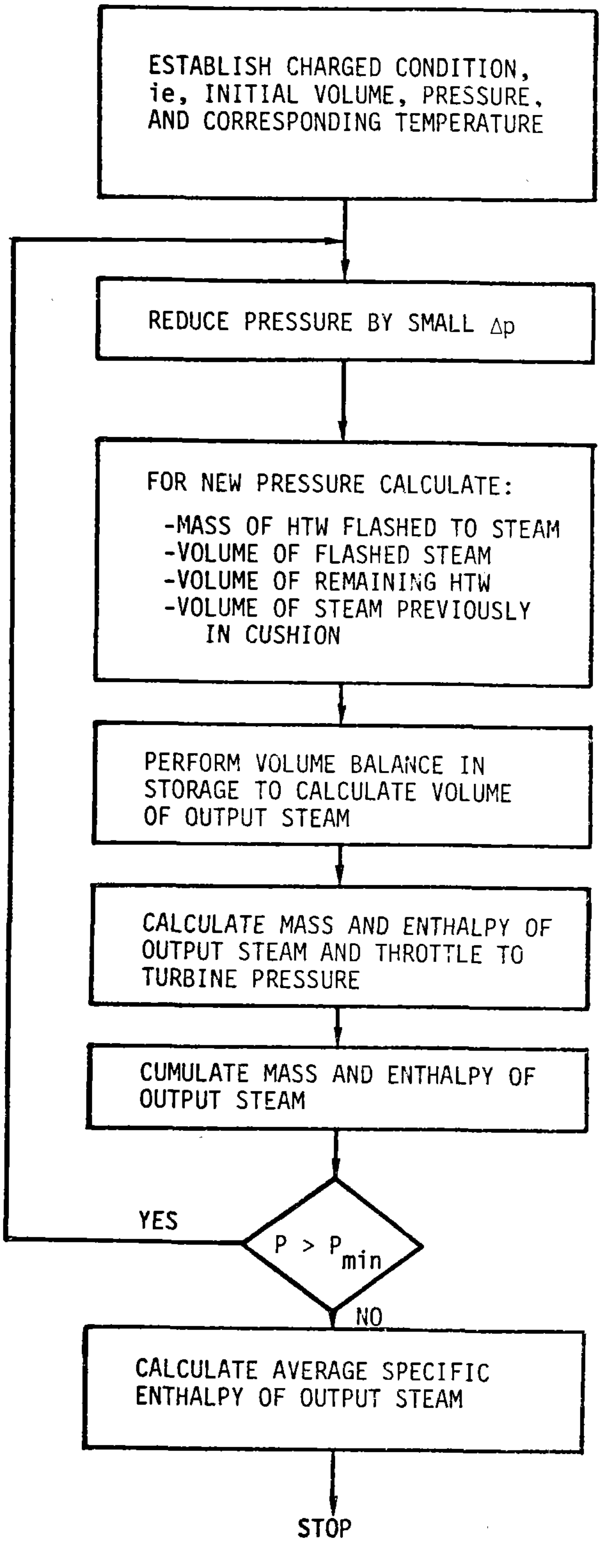


Figure 5-9. Computational procedure for discharging variable pressure accumulator.

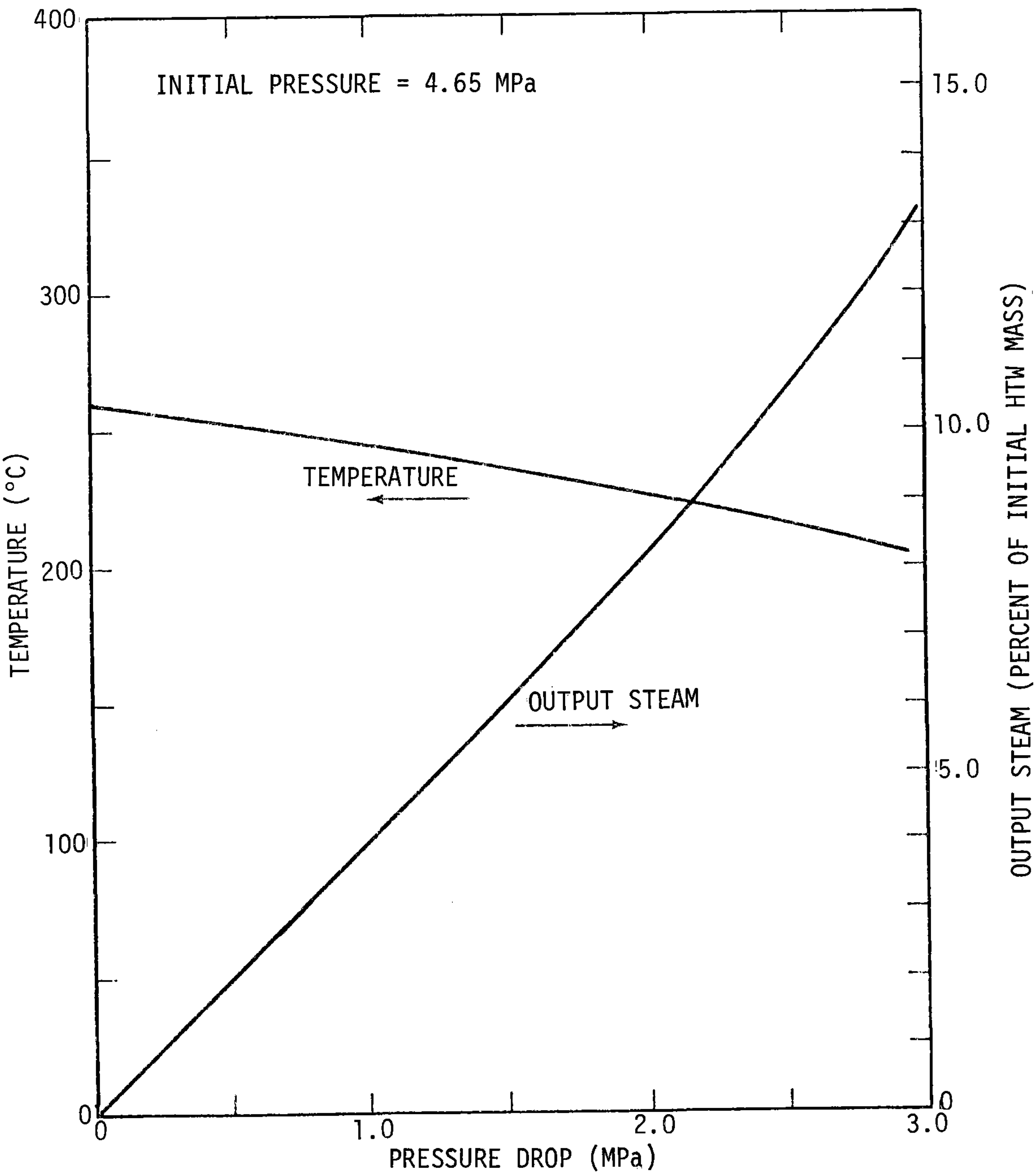


Figure 5-10. Typical discharge cycle for variable pressure accumulator.

SENSITIVITY ANALYSIS. In order to select a reasonable combination of design parameters, sensitivity analyses are performed for the 800 MW coal plant with a variable pressure accumulator TES system. A tentative base case set of values is chosen for the critical parameters and the system performance evaluated. The parameter values are then varied, individually and in combination, to determine the change in performance. These are used together with a preliminary cost analysis, and consideration for operational constraints, to define the base case set of parameters for use with other systems. For some of the key parameters it is unrealistic to select a single value, so a limited range is retained. Table 5-2 lists the critical parameters, the values selected for the tentative base case and the range of values used in the sensitivity analyses.

Table 5-2. Design parameter values for sensitivity analyses.

Parameter	Tentative Base Case Value	Range of Values for Analyses
Charge Steam Pressure (MPa)	4.86 (IP steam)	4.86 and 1.19
Storage Pressure (MPa)	4.65	4.65 to 1.03
Output Throttle Pressure (MPa)	2.24	2.41 to 0.52
Ratio of Discharge Time to Charge Time	0.75	1.00 to 0.37

The choice of charge steam condition is limited to three discrete values corresponding to the steam at the turbine inlets: 24.3 MPa, 538°C at the HP turbine; 4.86 MPa, 306°C at the IP turbine inlet; and 1.19 MPa, 188°C at the LP turbine inlet (crossover). HP steam is costly to store as HTW; at full pressure it could cost 3 to 6 times as much for containment as IP steam. Throttling the steam to intermediate pressures and removal of superheat loses available energy which could produce electric output if it were passed through the HP turbine. Use of LP steam necessarily implies low storage pressures, lower pressures in the steam generated, and consequently large low pressure peaking turbines. The IP steam condition is selected as the best base-case compromise between high-pressure high-cost storage vessels, and low-cost storage with high-cost turbines, condensers, and heat rejection systems. The LP steam is retained as a case to be evaluated in verifying this selection. The HP steam case can easily be rejected, since the benefits of HP and high temperature are obtained from the HP turbine. Only the available maximum power swing is limited by rejecting HP steam for charging.

In general the storage pressure should be as close to the charge steam pressure as possible, since throttling to lower storage pressures represents an unrecoverable loss. Hence the base case storage pressure is chosen to be 0.21 MPa (30 psi) below the charge steam pressure. The range listed in Table 5-2 includes storage 0.16 MPa (23 psi) below the LP steam pressure. These storage pressures are chosen as round numbers in the English System (675 and 150 psia) which represent reasonable pressure drops from the charging steam.

In order to limit thermal stresses in the storage vessel due to temperature cycling, the output throttle pressure is selected to permit about a 40°C temperature drop during discharge. For the 4.65 MPa storage pressure the throttle pressure is varied between 2.41 and 1.72 MPa (350 and 250 psia). Throttle pressures below these correspond to lower storage pressures, and are chosen to be approximately one-half the storage pressure. The accumulator is allowed to discharge until the internal pressure drops to the throttle pressure. In practice it could be discharged further, but the peaking turbines would then be receiving reduced flow. This flexibility may be an operating advantage.

SENSITIVITY — CHARGE TIME. In selecting the ratio of discharge time to charge time (the discharge/charge ratio) several factors must be considered. The peaking turbines are assumed to be operating at their design output, hence varying the output implies varying the size of the peaking turbines. Because of this assumption the peaking unit steam rate (kg steam/kW_e) and efficiency are independent of the output. Thus the discharge time and rate affect the required stored volume, but not the specific output or turnaround efficiency.

The main unit is assumed to be a fixed size operating at reduced load during the charge cycle. The leaving-loss correction effectively modifies the efficiency as a function of steam flow through the turbines. A glance at Figure 5-6 shows that there is a minimum steam flow rate, and hence an optimum rate at which to charge the TES system without loss of efficiency. As exit velocity and mass flow rate are roughly proportional, a 30 percent decrease in mass flow (and in power output) has little effect on efficiency but a much greater decrease in mass flow would carry an efficiency penalty (see Figure 4-5). For a given discharge period and peaking swing, eg 6 hours and 50 percent swing, the optimum charging period may be longer than reasonably attainable for the utility daily load pattern ratio of off-peak hours to peak hours. To explore this effect, daily charge periods of 6 to 16 hours are considered for 6 hours charging, ie ratios of 1.0 to 0.37. A ratio of 0.75 is chosen for the base case.

Table 5-3 shows the accumulator performance for the tentative base case. This is the same case as shown in Figure 5-10, except that the discharge is stopped when the pressure drops 2.41 MPa to 2.24 MPa (325 psia). Figure 5-11 shows the net electrical output from plant #1 while

Table 5-3. Variable pressure accumulator performance for base case.

	Output Steam	Charge Steam	Charge Feedwater
Mass (fraction of initial HTW mass)	(R_D) 0.1032	(R_C) 0.0955*	0.0077*
Pressure (MPa)	2.24	4.86	29.51
Specific Enthalpy (kJ/kg)	2805.6 (average)	2956.1	947.6
Temperature (°C)	218.6 (average)	306.4	218.4

* To balance mass and enthalpy in the accumulator, feedwater from the inlet of the high pressure feedwater heater is mixed with IP steam in the indicated ratio.

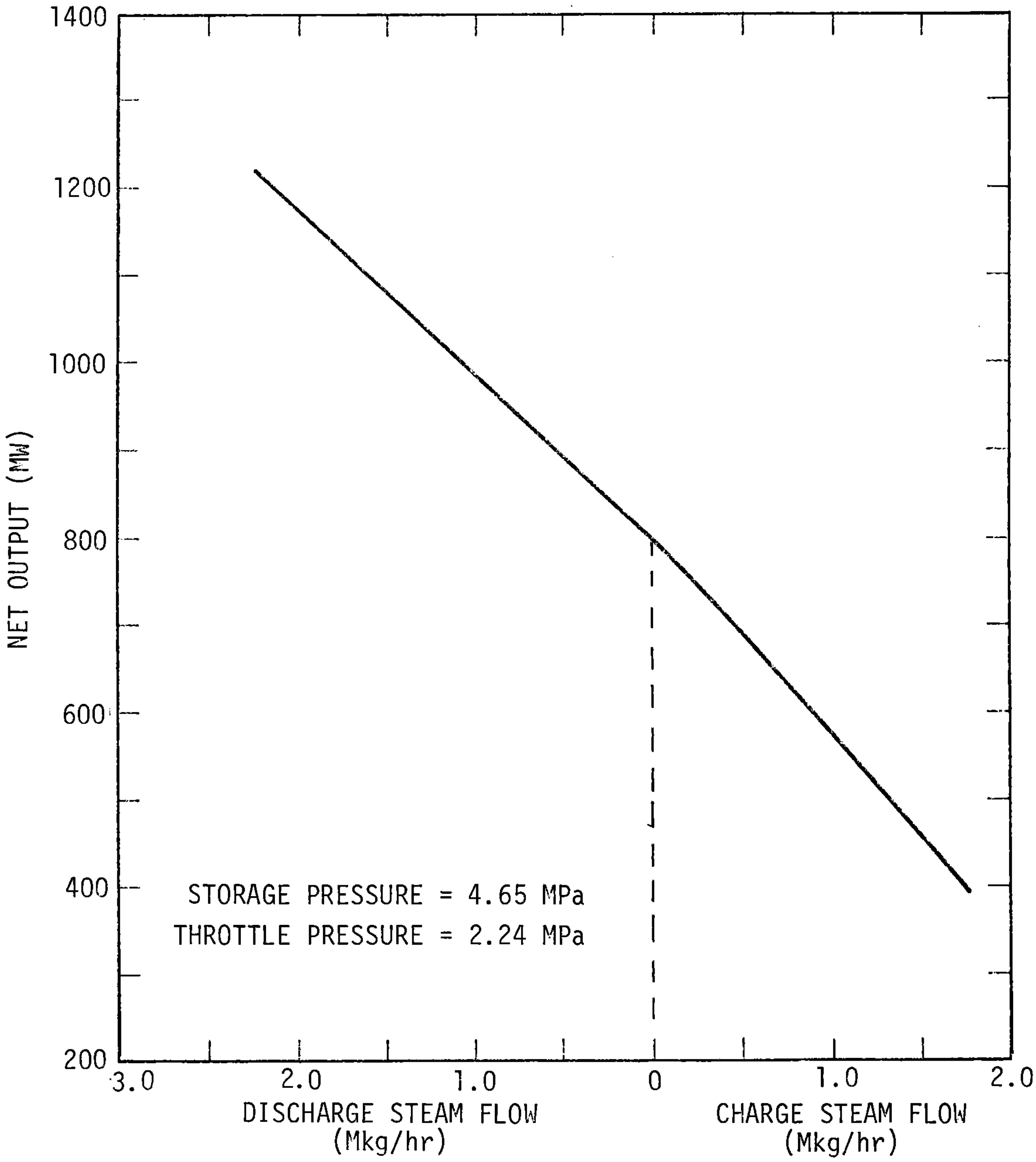


Figure 5-11. Output of modified 800 MW coal plant during TES operation.

charging and discharging the variable pressure accumulator. Note that the discharge portion of the curve is a straight line with a slope of 0.186 kWh/kg independent of the swing in MW and the discharge time. The specific volume of HTW saturated at 4.65 MPa is 0.00128 m^3/kg (steam tables). Using these numbers and the fraction of the stored mass (and volume) that is converted to steam, 0.1032 (Table 5-3) gives $0.186 \frac{kWh}{kg} \cdot \frac{kg}{0.00128\ m^3} \cdot 0.1032 = 15\ kWh/m^3$ as the specific output from storage. The peaking unit condenser heat rejection is 2.44 kW_{th}/kW_e (Figure 5-10).

The charge portion of Figure 5-11 is non-linear so the turnaround efficiency will depend on the peaking output and discharge time as well as the charge time. The turnaround efficiency is calculated from the data in Figure 5-11 and Table 5-3. The desired peaking output is chosen and the corresponding discharge steam flow found in Figure 5-11. The required charge steam flow is then computed as

$$\dot{W}_C = \dot{W}_D \left(\frac{t_d}{t_c}\right) \left(\frac{R_C}{R_D}\right) \quad , \qquad (5\text{-}5)$$

where

$\dot{W}_D$ = discharge steam flow (kg/hr)

R_C = ratio of charge steam to initial HTW mass from Table 5-3

R_D = ratio of output steam to initial HTW mass from Table 5-3.

The output during charging can then be obtained from Figure 5-11 and used in Equation 5-3 to get the turnaround efficiency. For example, a peaking output of 1040 MW (30 percent above the design output or a 30 percent peaking swing) requires $\dot{W}_D = 1.30 \times 10^6$ kg/hr. Choosing $t_d/t_c = 1$ and using $R_C = 0.0955$ and $R_D = 0.1032$ from Table 5-3 gives $\dot{W}_C = 1.20 \times 10^6$ kg/hr. This corresponds to an energy-cycle output of 532 MW and a turnaround efficiency of 89.6 percent.

Figure 5-12 shows the turnaround efficiency as a function of the discharge/charge time ratio for several values of peaking swing. Figure 5-12 makes it clear that long charging times are desirable, particularly for large peaking swings. This is true simply because the main turbines can operate closer to their "optimum" output when long charging times are available. However, operational considerations impose constraints that prevent extremely long charge times. A 6-hour discharge time and an 8-hour charge time (corresponding to a ratio of 0.75) are chosen as representative of typical daily-load curves and are used for all remaining calculations, bearing in mind that longer charging times would improve the efficiency.

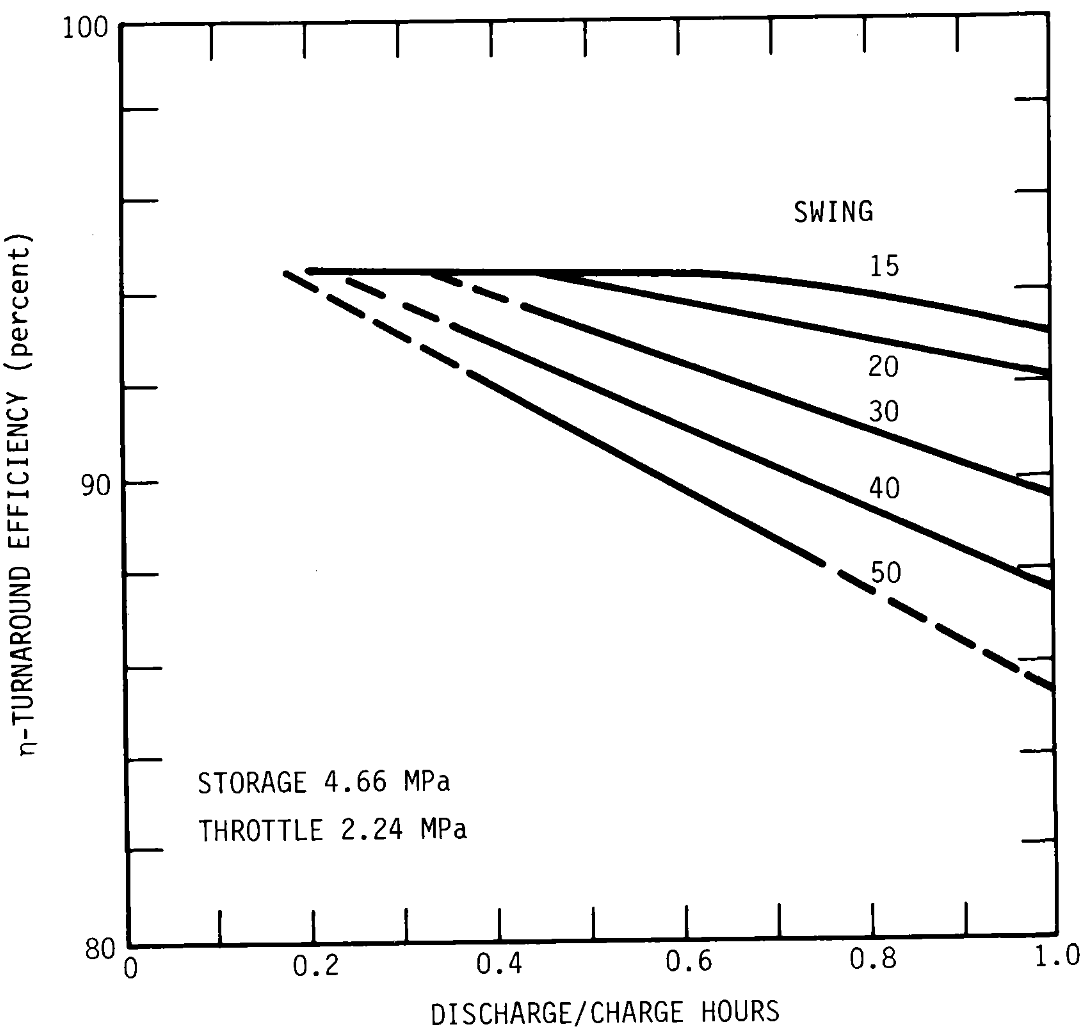


Figure 5-12. Effect of discharge/charge time ratio on turnaround efficiency.

SENSITIVITY — THROTTLE PRESSURE. Using a higher output-throttle pressure increases the electrical output for a unit mass of steam, but decreases the total mass of steam generated. Conversely, low throttle pressures provide less output per unit mass but more total mass. Table 5-4 lists the accumulator performance and specific output for throttle pressures of 2.41 and 1.72 MPa (350 and 250 psia), with other parameters set at the base case. Figure 5-13 shows the turnaround efficiency as a function of throttle pressure for several values of peaking swing. From these results it is apparent that the higher throttle pressure results in less throttling loss and a higher turnaround efficiency. The heat rejection requirements are also reduced, permitting a less expensive condenser. However, the specific output is reduced so a larger storage vessel is necessary. Preliminary cost analyses indicate that the minimum cost system occurs for throttle pressures lower than 1.72 MPa (250 psia).

Table 5-4. Variable pressure accumulator performance for varying throttle pressures.

	Throttle Pressure (MPa)					
	2.41			1.72		
	Output Steam	Charge Steam	Charge Feedwater	Output Steam	Charge Steam	Charge Feedwater
Mass (percent of initial HTW mass)	9.45	8.74	0.71	13.14	12.15	0.99
Pressure (MPa)	2.41	4.86	29.51	1.72	4.86	29.51
Enthalpy (kJ/kg)	2805.8 (average)	2956.1	947.6	2804.6 (average)	2956.1	947.6
Temperature (°C)	222.2 (average)	306.4	218.4	207.1 (average)	306.4	218.4
Specific Output (kWh/m^3)		13.88			18.17	
Peaking Unit Heat Rejection (kW_{th}/kW_e)		2.40			2.61	

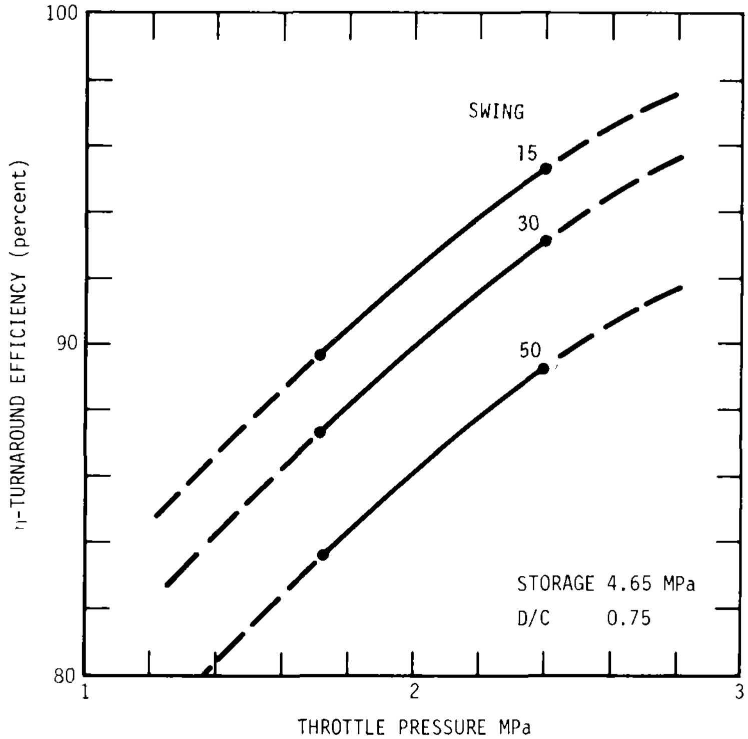


Figure 5-13. Effect of throttle pressure on turnaround efficiency.

SENSITIVITY – STORAGE PRESSURE. Reducing the storage pressure reduces the unit cost of the storage vessel, but increases the throttling losses if the source of charge steam remains the same. The first two columns of Table 5-5 list the important accumulator performance parameters for storage pressure of 2.41 and 1.03 MPa (350 and 150 psia) when charged with IP steam. For comparison purposes the throttle pressure is chosen as one-half the storage pressure. Figure 5-14 shows the turnaround efficiency as a function of the storage pressure for several values of the peaking swing. Besides the two pressures listed in Table 5-5, the value for a storage pressure of 4.65 MPa and a throttle pressure half as big (2.33 MPa) can be derived from Figure 5-13. The slight change from 2.24 MPa to 2.33 MPa gives specific output as 14.44 kWh/m^3 and condenser heat rejection as 2.42 kW_{th}/kW_e.

Table 5-5. Variable pressure accumulator performance for varying storage pressure.

	Storage Pressure (MPa)		
	2.41*	1.03*	1.03†
Output Steam			
Mass (percent of initial HTW mass)	7.62	5.81	5.81
Pressure (MPa)	1.21	0.52	0.52
Enthalpy (kJ/kg)	2797.9	2767.1	2767.1
Temperature (°C)	192.0	158.8	158.8
Charge Steam Mass (percent of initial HTW mass)	7.02	5.26	5.75
Charge Feedwater Mass (percent of initial HTW mass)	0.60	0.55	0.06
Specific Output (kWh/m^3)	10.24	6.57	6.57
Peaking Unit Heat Rejection (kW_{th}/kW_e)	2.85	3.60	3.60

* Charge steam and feedwater conditions are the same as those shown in Tables 5-3 and 5-4.

† Charge steam pressure and enthalpy are 1.19 MPa and 2787.4 kJ/kg respectively. Charge feedwater pressure and enthalpy are 1.19 MPa and 798.3 kJ/kg.

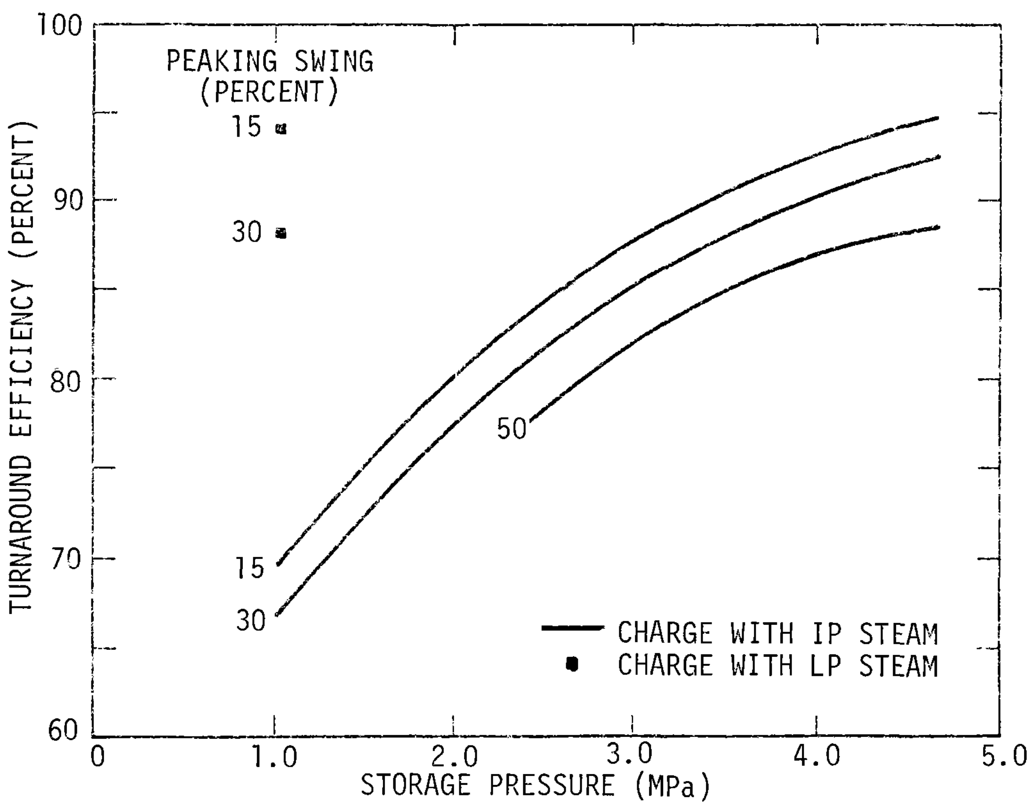


Figure 5-14. Effect of storage pressure on turnaround efficiency.

It is clear from Figure 5-14 that the throttling losses incurred by charging with IP steam are significant for the lower storage pressures. One method of avoiding such losses is to expand the charging steam through the IP turbine rather than a throttle, ie, to charge with steam from the crossover. The third column of Table 5-5 lists the accumulator performance parameters for a 1.03 MPa storage pressure when crossover steam is used for charging. Note that the output related parameters are identical to those for the same accumulator when charged with IP steam. The resulting turnaround efficiency is shown in Figure 5-14 for peaking swings of 15 and 30 percent. Note that for small peaking swings the efficiency is essentially the same as that achieved with storage pressure of 4.65 MPa (675 psia). However, it falls much faster with increased swing, because all of the swing is now accomplished in the LP turbine, causing it to operate further from the optimum output.

In summary of the sensitivity analyses, the very low storage pressures are not an attractive option unless charged with crossover steam. This limits the storage pressure to either 4.65 or 1.03 MPa (675 or 150 psia), or values reasonably close to these. The lower storage and throttle pressure dictate very large peaking turbines and condenser. The base case throttle pressure is chosen to be one-half of the storage pressure, but excursions below that are retained to permit evaluating the effect on system costs. The base case cycle is assumed to be 6 hours TES discharging and 8 hours charging. Two values of peaking swing (15 and 50 percent) are retained. The lower one permits a comparison of all systems on equivalent terms and the higher one shows the effect of large swings on those systems that are capable of them.

Variable Pressure Accumulator — Plant #2

A briefer analysis with fewer excursions from a base case is described. The highest steam pressure available in the nuclear plant (Figure 5-4) is 6.72 MPa (975 psia). The storage pressure is chosen as 6.21 MPa (900 psia). During charging of the accumulator a small amount of the stored HTW is removed and pumped to the nuclear steam supply inlet. Table 5-6 lists the accumulator performance parameters for output throttle pressures of 3.10 and 2.59 MPa (450 and 375 psia). Figure 5-15 shows the plant electrical output as a function of steam flow for the charge and discharge cycles. The resulting turnaround efficiencies for 15 and 50 percent peaking swings are included in Table 5-6. It should be noted that the maximum charge rate permits peaking swings in excess of 75 percent.

Table 5-6. Variable pressure accumulator performance with plant #2 — LWR.

	Output Throttle Pressure (MPa)	
	3.10	2.59
Output Steam		
Mass (percent of initial HTW mass)	11.30	13.57
Pressure (MPa)	3.10	2.59
Enthalpy (kJ/kg)	2800.4	2801.6
Temperature (°C)	235.7	225.7
Charge Steam Mass (percent of initial HTW mass)	11.46	13.77
Mass of HTW Removed (percent of initial HTW mass)	0.16	0.20
Specific Output (kWh/m^3)	15.39	17.88
Peaking Unit Heat Rejection (kW_{th}/kW_e)	2.40	2.52
Turnaround Efficiency		
15 percent swing	93.0	90.0
50 percent swing	90.0	87.0

Expansion Accumulator

Figure 2-4 is a schematic representation of an expansion accumulator with the output HTW used in flash evaporators. When fully charged there is a small steam cushion on top of a large volume of HTW, as in the variable pressure accumulator. During discharge, HTW is withdrawn from the bottom of the storage vessel, lowering the internal pressure. The steam cushion expands and some of the remaining HTW flashes to steam to restore equilibrium. The temperature and pressure in the vessel decrease steadily throughout the discharge cycle but not as much as in a variable pressure accumulator. In this mode of operation

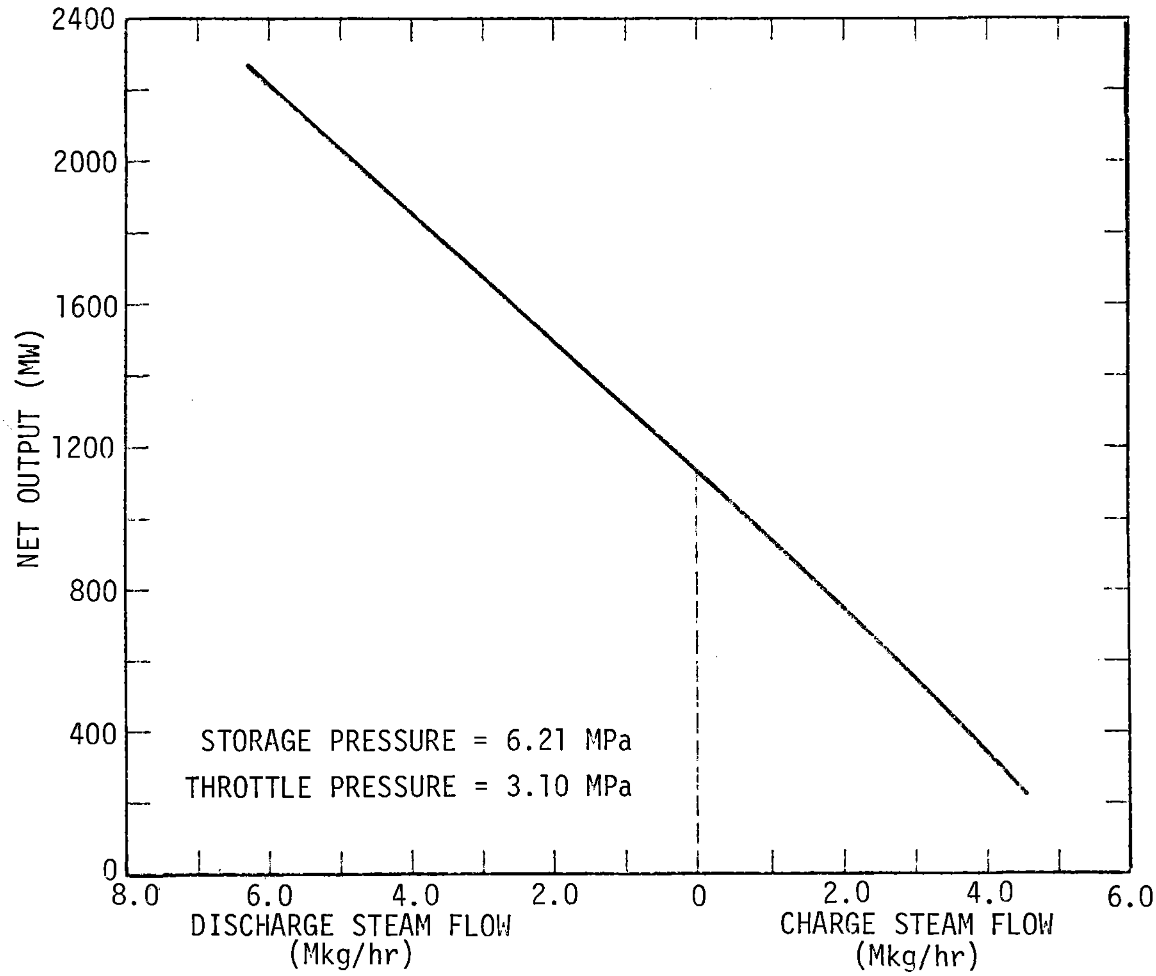


Figure 5-15. Output of plant #2 - 1140 MW nuclear plant - during TES operation.

nearly all of the stored HTW can be withdrawn for external steam generation.

The HTW removed from the accumulator is throttled to a lower pressure in a flash evaporator. The output steam is then used in a peaking turbine. The evaporator drain water can be pumped into the main turbine feedwater loop, stored, or throttled to a still lower pressure in another flash evaporator. Any number of evaporators may be used, but this requires multiple peaking turbines or a multiple inlet turbine.

To recharge the accumulator a mixture of steam and feedwater is admitted to the storage vessel, gradually raising the pressure and temperature until the initial charged condition is reached. Because of the latent heat of steam the mass flow of feedwater greatly exceeds that of steam in the charge mixture.

In many respects the thermodynamic processes in the expansion accumulator are similar to those in the variable pressure accumulator. Thus the modeling approach is similar. The performance during discharge is evaluated using the iterative procedure shown in Figure 5-16. Figure 5-17 shows a typical discharge cycle for an initial storage pressure of 4.65 MPa (675 psia). Note that the final pressure, with all the HTW removed from storage is about 70 percent of the initial storage pressure. A large fraction of the HTW can be removed with very little pressure and temperature drop. For recharging, the mix of feedwater and steam required is calculated by a mass and enthalpy balance between the charged and discharged conditions. It is assumed that the mix remains uniform during the entire charging process, although this implies that initially the feedwater will flash to steam and be recondensed later in the cycle as the internal pressure rises.

Early in the study consideration was given to using a combination of steam generation and feedwater supply with the expansion accumulator (Section 3, Selection 1). The drain from the final flash evaporator is pumped into the feedwater loop at a point where the temperatures match. This scheme requires a sizeable surge/storage tank to accommodate the cold feedwater replaced by the drain water from the evaporators. The

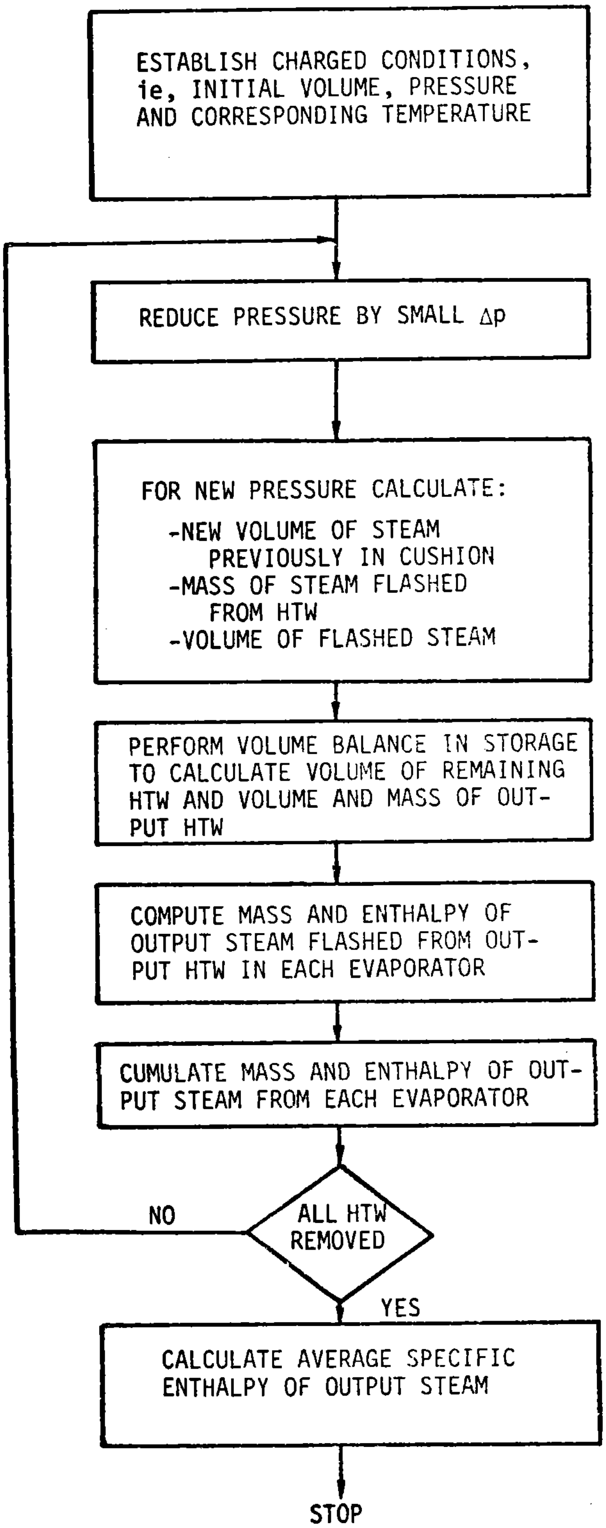


Figure 5-16. Computational procedure for discharging expansion accumulator.

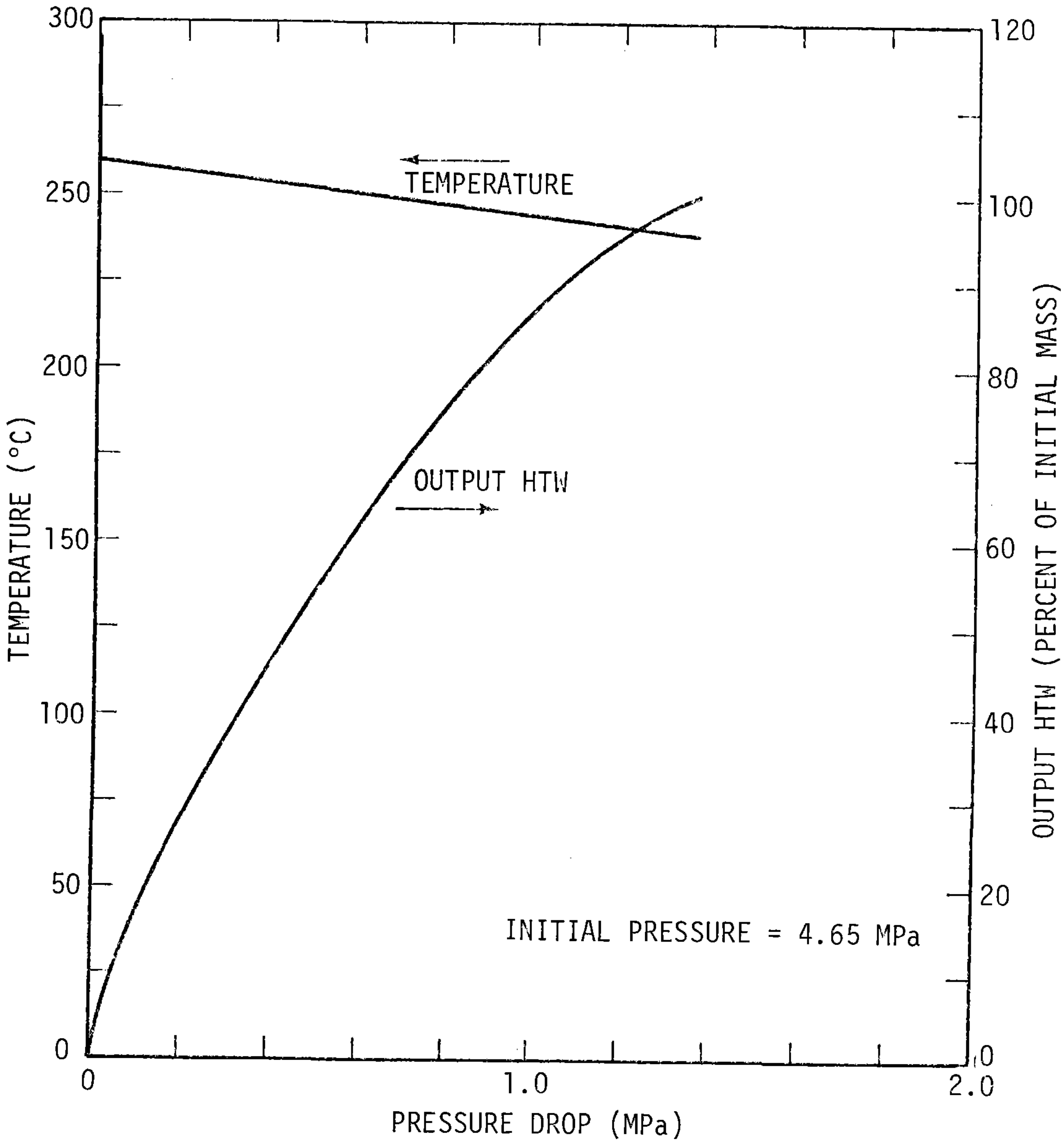


Figure 5-17. Typical discharge cycle for expansion accumulator.

peaking swing is also severely limited because the discharge rate of the accumulator is restricted by the boiler feedwater flow. In fact the maximum swing is not much greater than for a pure feedwater storage system. For this reason the concept is dropped from further consideration and all analyses assume that the evaporator drain water is stored in a supplementary storage vessel at an intermediate pressure.

PLANT #1. Table 5-7 lists the performance parameters for an expansion accumulator with a storage pressure of 4.65 MPa (675 psia) charged with IP steam and feedwater from the inlet of the high pressure heater. Data are shown for systems using 1, 2, and 3 flash evaporators. In all cases the drain is stored at a pressure slightly above that of the final evaporator. The third evaporator is chosen to operate at 0.16 MPa (23 psia) so that the drain water can be stored at low pressure. It requires a large low-pressure turbine and condenser which likely offset any cost savings due to the low pressure drain storage.

Table 5-7. Expansion accumulator performance with plant #1.

Number of Evaporators	1	2	3
Output Steam Mass (percent of initial HTW mass)			
2.24 MPa (325 psia)	8.45	8.45	8.45
1.21 MPa (175 psia)	-	6.12	6.12
0.16 MPa (23 psia)	-	-	12.20
Drain Water Mass (percent of initial HTW mass)	89.44	83.31	71.11
Drain Storage Volume (percent of accumulator volume)	83	74	59
Charge Steam Mass (percent of initial HTW mass)	7.67	13.43	24.48
Charge Feedwater Mass (percent of initial HTW mass)	90.22	84.46	73.41
Specific Output (kWh/m^3)	11.33	18.92	28.26
Peaking Unit Heat Rejection (kW_{th}/kW_e)	2.44	2.60	3.50
Turnaround Efficiency			
15 percent swing	88.0	84.3	68.8
50 percent swing	81.8	78.3	*

* Maximum peaking swing is 40 percent with efficiency of 64.6 percent.

PLANT #2. Table 5-8 lists performance parameters for an expansion accumulator with a storage pressure of 6.21 MPa (900 psia) charged with live steam and feedwater from the boiler inlet. The drains are again stored at a pressure slightly above that of the final evaporator.

Table 5-8. Expansion accumulator performance with plant #2.

Number of Evaporators	1	2
Output Steam Mass (percent of initial HTW mass)		
3.10 MPa (450 psia)	8.84	8.84
1.21 MPa (175 psia)	-	9.70
Drain Water Mass (percent of initial HTW mass)	88.35	78.65
Drain Storage Volume (percent of initial HTW volume)	83	68
Charge Steam Mass (percent of initial HTW mass)	8.67	18.57
Charge Feedwater Mass (percent of initial HTW mass)	88.52	78.62
Specific Output (kWh/m^3)	10.87	21.31
Peaking Unit Heat Rejection (kW_{th}/kW_e)	2.40	2.74
Turnaround Efficiency		
15 percent	86.4	79.6
50 percent	83.4	76.6

Displacement Accumulator

Figure 2-5 shows a schematic representation of a displacement accumulator with the output HTW used in flash evaporators. When fully charged the storage vessel is full of HTW at slightly above saturation pressure. During discharge HTW is withdrawn from the top of the vessel and throttled to one or more flash evaporators. The drain from the final evaporator is pumped to the bottom of the vessel, creating a sharp temperature gradient (thermocline) between the HTW and the drain water. If care is taken to avoid mixing, the thermocline can be maintained reasonably sharp. Because some steam has been produced and the drain water has a lower specific volume than the HTW removed, water at the drain temperature is required from a supplementary storage tank to keep the accumulator full. Note that the temperature and pressure of the output HTW are constant throughout the discharge until the thermocline reaches the top of the tank.

To recharge the accumulator, cold water is circulated from the bottom of the tank, mixed with charging steam and returned to the top of the tank, pushing the thermocline down. Because of the steam added and the increased specific volume, excess cold water must be removed and returned to the supplementary storage. In general the mass of water returned to storage during charging is not equal to that removed during discharge.

Modeling the accumulator is relatively straightforward since only equilibrium thermodynamic processes are involved so no detailed description is given. All that is required is to maintain a mass, volume, and enthalpy balance. The thermocline is assumed to be perfect; thermal losses and pressure drops are neglected.

The thermal stresses introduced by the motion of the sharp thermocline can be a serious problem. This imposes a limit on the temperature difference across the thermocline, which restricts the allowable pressure drop in the evaporators and the quantity of steam generated. A combination of steam generation and feedwater supply is not appropriate for the displacement accumulator. From the preceding description it is clear that the accumulator would be providing feedwater during the charge (off-peak) cycle and requiring excess feedwater during the discharge or peaking cycle.

PLANT #1. Table 5-9 lists the important performance parameters for a displacement accumulator with a storage pressure of 4.65 MPa (675 psia) charged with IP steam. Data for systems with 1 and 2 flash evaporators are included. A comparison of Table 5-9 with Table 5-7 indicates that the displacement accumulator gives slightly higher turnaround efficiency and specific output than the expansion accumulator. It also requires a smaller supplementary storage tank.

PLANT #2. Because the results for a displacement accumulator with the coal plant are so similar to those obtained for an expansion accumulator, no evaluation was performed for the nuclear plant.

Table 5-9. Displacement accumulator performance with plant #1.

Number of Evaporators	1	2
Output Steam Mass (percent of initial HTW mass)		
2.24 MPa (325 psia)	10.57	10.57
1.21 MPa (175 psia)	-	6.12
Supplementary Storage Volume (percent of initial HTW volume)	17	26
Charge Steam Mass (percent of initial HTW mass)	9.56	15.21
Temperature Difference Across Thermocline (°C)	40.4	70.1
Specific Output (kWh/m^3)	13.90	21.30
Peaking Unit Heat Rejection (kW_{th}/kW_e)	2.44	2.60
Turnaround Efficiency		
15 percent Swing	88.8	85.2
50 percent Swing	83.0	79.4

Feedwater Storage Systems Modeling

HTW feedwater storage systems are just what the name implies. During the charge cycle excess feedwater is drawn from a cold storage reservoir, heated in standard feedwater heaters by extraction steam, and stored in a pressure vessel just above the saturation pressure. When extra electrical output is required, the stored HTW is pumped to the boiler inlet, replacing a part of the normal feedwater. This reduces the extraction steam flow, allowing more steam to flow through the entire turbine and producing extra power. No large steam turbine is currently capable of operating with all (or most) of the extraction steam shut off. The maximum peaking swing is estimated by various authors and proponents at 6 to 35 percent. Some assume quite low boiler inlet temperatures (Selection #5), others assume very high boiler inlet temperatures (Selection #8) in part accounting for the variance. Conventional near-term available plants are most likely to be limited to under 20 percent.

A displacement accumulator or a two-tank system are suitable for feedwater storage. Since boiler quality feedwater should not be exposed even to inert gases, the "cold" tank of a two-tank system should be near 100°C with a steam cushion. Except for the thermal stresses developed in the displacement accumulator there is essentially no other difference between the two, so a two-tank system is modeled here. In order to handle the extra steam flow during peaking operation, the exhaust area of the main turbines in both the coal plant and the nuclear plant are increased by 25 percent, giving a slightly increased output at the design flow rates.

Table 5-10 lists the performance for both the coal and nuclear plants as shown in Figures 5-3 and 5-5. The maximum peaking swing for both plants is about 17 percent. Both plants achieve very good turn-around efficiency and high specific output.

Table 5-10. Performance of feedwater storage systems.

	Plant #1 — 800 MW Coal Plant	Plant #2 — 1140 MW Nuclear Plant
Increase in Output at Design Flow (percent)	1.1	0.9
Maximum Peaking Swing		
Percent Above Output at Design Flow	17	17
Percent Above Nominal Output	18.3	18.0
Temperature Difference Cold to Hot (°C)	188.8	146.3
Specific Output (kWh/m^3)	40	30
Heat Rejection (kW_{th}/kW_e)		
Maximum Charge Rate	0.90	1.85
Design Flow	1.26	1.94
Maximum Peaking	1.57	2.04
Turnaround Efficiency at 15 Percent Swing Above Design Flow Output	88.0	90.8

Summary

Table 5-11 presents a summary of the HTW TES systems for easy comparison. It is interesting to note that there are no significant differences in the TES system performance between the coal plant and the nuclear plant.

The parameters common to data in the summary should be recalled. All are for 6 hours discharge, 8 hours charge (or D/C = 0.75). The

Table 5-11. Summary of HTW systems.

Concept	Plant #1 — 800 MW HSC				Plant #2 — 1140 MW LWR			
	Steam Pressure (MPa)	Specific Output (kWh/m^3)	Turnaround Efficiency (percent) Swing 15	50	Steam Pressure (MPa)	Specific Output (kWh/m^3)	Turnaround Efficiency (percent) Swing 15	50
Variable Pressure Accumulator	2.41	13.9	95.4	89.2				
	2.24	15.0	94.2	88.0	3.10	15.4	93.0	90.0
	1.72	18.2	89.7	83.6	2.59	17.9	90.0	87.0
	0.52	6.6	70(94)	<65				
Expansion Accumulator								
1 Evaporator	2.24	11.3	88.0	81.8	3.10	10.9	86.4	83.4
2 Evaporators	2.24,1.21	18.9	84.3	78.3	3.10,1.21	21.3	79.6	76.6
3 Evaporators	2.24,1.21 0.16	28.3	68.8	<63				
Displacement Accumulator								
1 Evaporator	2.24	13.9	88.8	83.0				
2 Evaporators	2.24,1.21	21.3	85.2	79.4				
Feedwater Storage	-	40	88.0	-	-	30	90.8	-

charge steam pressure for plants #1 and #2 are 4.86 MPa (705 psia) and 6.72 MPa (975 psia). There is one exception. When crossover steam at 1.16 MPa (168 psia) is used as charge steam, the turnaround efficiency is the higher value shown in parentheses (last line of variable pressure accumulator data).

ONE-BAR TES SYSTEMS MODELING

The "one-bar" or atmospheric pressure thermal energy storage systems are characterized by the use of low vapor pressure (LVP) fluids as a heat storage medium, as a heat transfer fluid to a solid phase for heat storage, or in both roles. The primary requirements on the fluid are its low vapor pressure at the temperatures of interest, which permits containment in conventional atmospheric pressure steel tanks, large heat capacity, sufficiently low viscosity, and stability under repeated heating/cooling cycles.

In Sections 2 and 3, a number of sensible heat storage concepts employing low vapor pressure fluids were described which differed in the configuration and mode of operation of the storage system itself. The point was made in the preliminary screening discussion that there is a large degree of independence between the heat exchangers which interface with the utility power plant and the tankage which contains the heat storage medium. Thus, with the same interface and mode of use of stored thermal energy, the storage system can be configured as multiples of variously sized liquid-filled tanks or of packed-bed thermocline tanks operated such that the void volume is kept filled with fluid or is drained once the unit has been charged to its upper temperature. In modeling these systems, it is found that the nature of the interface with the power plant (ie, the design of the heat exchangers) and the physical properties of the heat transfer fluid dominate the power-related aspect of the TES system and that these factors are significantly decoupled from the configuration and mode of operation of the heat storage units which dominate the energy-related aspect of the system.

The two ways of utilizing the stored energy in these sensible heat systems are the same as those investigated for the high temperature water (HTW) systems: feedwater heating, allowing the main turbine to operate with reduced extraction thereby generating additional power during peak demand periods; and steam generation, employing the stored heat to generate steam for admission to a separate peaking turbine when demand rises. The one-bar, sensible heat systems differ from the HTW systems in that provision must be made to keep the heated medium physically separate from the working fluid by the use of appropriately designed heat exchangers.

Steam Generation Systems Modeling

Thermal energy stored as sensible heat in a fluid or fluid plus solid medium during the off-peak or charge phase of a load cycle can be used to generate steam for admission to a separate peaking turbine-generator to provide increased power during the on-peak or discharge phase. The virtually complete decoupling of the main and peaking turbines results in flexibility of equipment design and operation for the charge and discharge phases. An essential part of the analysis of these concepts is to investigate their performance and cost as a function of certain primary design parameters.

GENERAL CONCEPT DESCRIPTION. In the most general sense, thermal storage steam generator systems consist of a train of three heat exchangers (desuperheater, condenser, subcooler) which serve as a storage heater, and transfer enthalpy from the charging steam supply to the storage medium; tankage, piping, and pumps to circulate the heat transfer fluid between heat exchangers and storage; and a train of three heat exchangers (preheater, boiler, superheater) which serve as a steam generator and from which steam is fed to the peaking turbine. Some obvious variants are possible. If the charging steam is superheated, the desuperheater heat exchanger may be replaced by an attemperator or spray desuperheater; if saturated, the steam may be admitted directly to the condenser. Similarly, the steam generator output may be taken directly from the boiler if saturated steam is desired or may be superheated if that is economically preferable.

PRIMARY DESIGN VARIABLES. The qualitative temperature relationships among the charge steam, the storage medium, and the generated steam are displayed in Figure 5-18. The highest temperature profile represents the charge steam; in general, the major part of its total enthalpy decrease occurs as the latent heat of condensation is transferred to the storage medium at saturation temperature.

The intermediate sloping line represents the heat transfer fluid to the storage system, which may also be the storage medium. As long as the temperature dependence of the heat capacity of the storage medium is small, its temperature profile can be represented by a line of essentially constant slope, indicating that all the energy transferred to it is in the form of sensible heat, ie, no phase change occurs. A useful choice of the two parameters required to specify the position of this line is the temperature difference between it and the hot end of the condenser, and its slope. The temperature difference specifies the fluid temperature *approach* or "pinch point," and is a result of the *effectiveness* of the heat exchangers. The slope specifies the temperature swing of the storage medium and depends on the *mass flow ratio* between the heat transfer fluid and the charge steam;

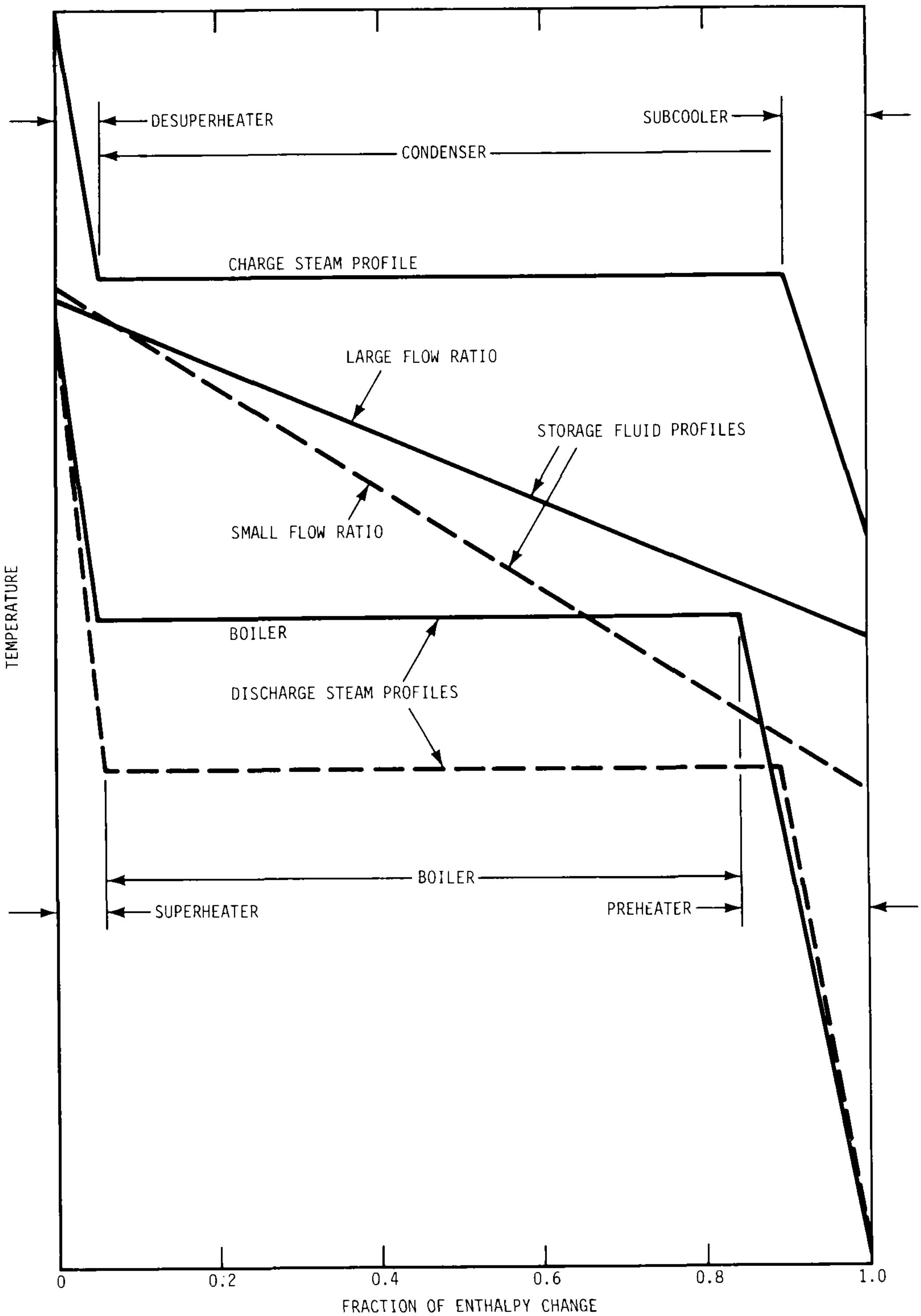


Figure 5-18. Sensible heat storage: representative temperature profiles.

a large ratio corresponds to a smaller slope and a smaller fluid temperature swing than is the case with a small ratio.

The impact of the position of the fluid temperature profile line on the character of the steam that can be generated is evident from the solid and broken lines in the figure. Note that, for a given boiler inlet temperature approach, the large mass flow ratio (solid lines) permits a higher generated steam temperature (and pressure) than does the smaller mass flow ratio (broken lines), since the pressure is the saturation pressure at the boiling temperature. Although superheating may produce almost the same output steam temperature in the two cases, the available energy of the steam is greater in the higher pressure case.

The conclusions to be drawn from this qualitative discussion are that once the configuration of this kind of system is known (charge steam properties, choice of storage medium, etc), the key parameters which define the thermodynamic performance of the system are the values of the temperature approach at all heat exchanger pinch points and the ratio of the quantities of heat storage fluid and charge steam involved. Once these parameters are specified, the properties and flow rate of the generated steam can be determined.

COMPUTATIONAL PROCEDURES. The unique element in the thermodynamic modeling of the steam generator systems is the quantification of the temperature profiles shown in Figure 5-18. The results sought are the characteristics of the output steam: its properties and normalized flow rate, ie, kg of discharge steam per hour per kg of charge steam per hour for a given charge/discharge time ratio. These enable specification of the performance and design requirements for the storage heater and steam-generator heat exchangers and, combined with the downswing performance of the main turbine during the charge phase and the upswing performance of the peaking turbine in the discharge phase, permit specification of the turnaround efficiency, system size parameters and costs.

Heat Exchangers. Figure 5-19 displays a generalized system diagram with the main components and state points (nodes) identified. TES charging steam is admitted at node 1, where a three-way option exists: if the steam is superheated, it may flow through either a desuperheater (node 2), or an attemperator (node 2′) where it is mixed with return feedwater (node 6); if the steam is saturated, it is passed directly to node 2″. The steam/water states and flow rate at nodes 1 and 7 are the primary independent variables. The state at nodes 2 or 2″ is saturated at the node 1 pressure. If an attemperator is specified, the intensive properties at 2′ are the same as those at 2, but the flow is increased by the factor $1 + (h_1 - h_2)/(h_2 - h_6)$ as a result of feedwater added from node 6, where the h's represent specific enthalpies and the subscripts identify the node.

At the condenser outlet, node 3, the state is saturated water at the pressure and temperature of node 2. This condensate is subcooled at node 4 to a state such that the temperature at the feedwater return pump output, node 5, matches that specified for the main cycle feedwater return, node 7. As is the case throughout this section, feedwater pumps are assumed to have a constant efficiency of 60 percent. Intensive properties are identical at nodes 5, 6, and 7; the flow rates, however, depend on whether water is diverted to the attemperator, node 6, as discussed above.

The temperature profile of the counterflowing heat transfer fluid (nodes 10 to 13) is determined as a function of the two major parameters of the system: temperature approach, α, and fluid to charge steam mass flow ratio, M_c. For the selected approach, the fluid temperature at the hot end of the condenser, T_{12}, is $T_2 - \alpha$, where T_2 is the steam condensation temperature at node 2. This represents the specified pinch point as described in connection with Figure 5-18. The fluid temperatures at nodes 10, 11, and 13 are now determined by equating the enthalpy change in each segment of the steam profile with the enthalpy change in the corresponding segment of the fluid profile. For example, for a fluid where heat capacity, c_p, is independent of

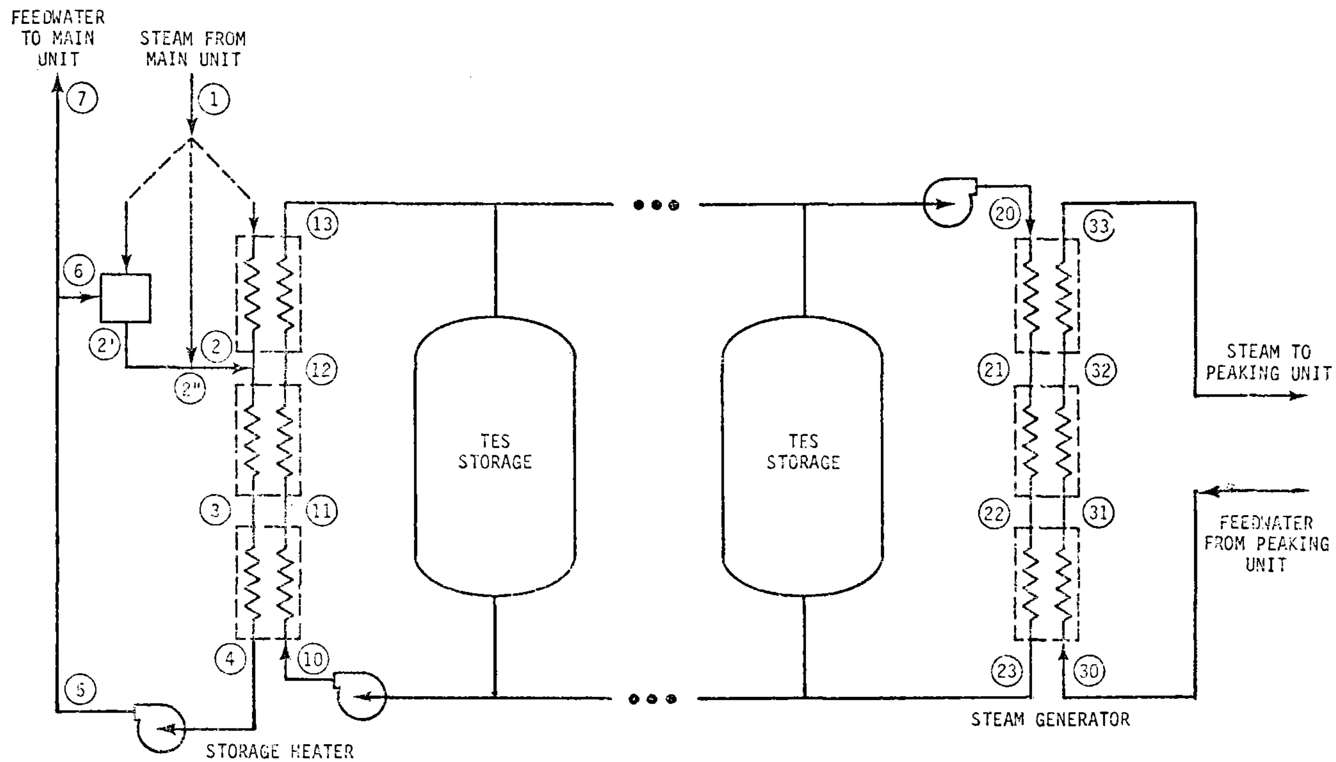


Figure 5-19. Thermal energy storage steam generator system.

temperature (eg, molten salt mixtures such as HITEC): $h_1 - h_2 = c_p(T_{13} - T_{12})M_c$ where h_1 and h_2 are steam-specific enthalpies at nodes 1 and 2 and T_{13} and T_{12} are fluid temperatures at nodes 13 and 12; whence $T_{13} = T_{12} + (h_1 - h_2)/c_pM_c$.

Similarly, for fluids whose heat capacity may be represented as: $c_p = c_1 + c_2T$ where c_1 and c_2 are constants (eg, heat transfer oils such as Caloria HT-43), the equivalent relationship is:

$$T_{13} = \left[\left(\frac{c_1}{c_2} + T_{12}\right)^2 + \frac{2(h_1 - h_2)}{c_2M_c}\right]^{1/2} - \frac{c_1}{c_2} \quad . \tag{5-6}$$

The identical method is applied to determine the remaining fluid temperatures, T_{10} and T_{11}. This completes the specification of the steam/water and fluid states in the storage heater as appropriate for the charge phase of the cycle. The performance specifications of the three heat exchangers are now determined. In each case, the capacity rates of the two streams are defined as the product of the mass flow rate and the heat capacity. In general, the main heat capacity of the steam/water stream in an exchanger is best calculated as the enthalpy change divided by the temperature change, while that of the fluid stream is best evaluated at its mean temperature, T_m. Since for charging, the steam/water stream is the hot stream and the fluid is the cold stream, we have

$$C_h = \dot{W}_c \ \Delta h/\Delta T \tag{5-7}$$

$$C_c = \dot{W}_{fc} \ c_p(T_m) \tag{5-7a}$$

where C_h and C_c are the hot and cold capacity rates, $\dot{W}_c$ is the charge steam mass flow rate, and $\dot{W}_{fc}$ or $M_c \cdot \dot{W}_c$ is the fluid mass flow rate. Note that in the condenser, because of the isothermal phase change, the hot stream capacity rate becomes infinite.

The capacity rate ratio, R, is defined as C_{min}/C_{max}, where those quantities are the minimum and maximum of C_h and C_c, respectively. If a phase change occurs, R is zero.

The effectiveness (ε) — number of thermal units (N_{tu}) method is used to characterize the heat exchangers. Basic theory (Reference 236) defines the effectiveness as:

$$\varepsilon = \frac{C_h}{C_{min}} \frac{(T_{H1} - T_{H2})}{(T_{H1} - T_{C1})} = \frac{C_c}{C_{min}} \frac{(T_{C2} - T_{C1})}{(T_{H1} - T_{C1})} \qquad (5\text{-}8)$$

where the temperature subscripts designate hot stream or cold stream (H, C respectively) and inlet or outlet (1, 2 respectively). The number of thermal units relates the UA-product to C_{min}:

$$N_{tu} = UA/C_{min} \qquad (5\text{-}8a)$$

where U = overall heat transfer coefficient: $kJ/(m^2 \cdot h \cdot K)$
A = effective heat transfer area.

For counterflow heat exchangers, it can be shown that

$$N_{tu} = \frac{1}{1 - R} \ln \frac{1 - \varepsilon R}{1 - \varepsilon} \quad , \qquad (5\text{-}9)$$

which reduces to $(-\ln(1 - \varepsilon))$ for phase change exchangers where $R = 0$. Thus, the capacity rates and temperatures which characterize a heat exchanger determine the UA-product, and an independent calculation of U determines the effective area required, hence the size and, with other specifications such as design type and operating pressure, the cost of the exchanger.

Overall heat transfer coefficients, U, were estimated by standard methods (Reference 215, Section 10) from inside and outside film coefficients, constant fouling resistances, and steel tube-wall conduction assuming nominal 2.5 cm (1 in.) outside diameter tubes of 0.4 cm (0.15 in.) wall thickness. Film coefficients were calculated using Colburn (j-factor) correlations for forced convection under conditions of fully turbulent flow. In general, standard tabulated values were used for film coefficients of water or steam as tube-side material, and film coefficients were calculated for the various heat transfer fluids or shell-side material flowing normal to staggered tube banks.

Once the quantification of the storage heater performance is complete, the calculation proceeds to determine the comparable values for the steam generator. Given the state of the feedwater returned from the peaking turbine (node 30), the problem is to find the maximum boiling temperature that can be developed (node 31) by the available supply of hot fluid stored at $T_{13} = T_{20}$. In principle, the boiler temperature, T_{31}, is at the intersection of the peaker feedwater heating curve (nodes 30 to 31) and a line below the fluid temperature profile by the amount α to account for the specified temperature approach at the boiler inlet (see Figure 5-18). If the heat capacities of both substances were constant, both lines would be straight and their intersection could be determined directly. Since this is not the case, an iterative computation is necessary. An initial boiler temperature is assumed and the temperature of the heat storage fluid at the corresponding node 22 is calculated by enthalpy balance over the preheater. If the difference between the assumed T_{31} and the calculated T_{22} differs from α by more than an arbitrary allowable error (0.028°C or 0.050°F was used), the value of T_{31} is altered and the calculation is repeated. The algorithm used employs the first two trials to extrapolate an estimate, usually adequate, based on the assumption of constant heat capacities. When necessary, further estimates of T_{31} are obtained as the mean of the current T_{31} and $T_{22} - \alpha$ and the process is repeated.

Once T_{31}, the boiler temperature and its saturation pressure are known, the remainder of the discharge steam profile is calculated directly. If saturated output steam is desired, the discharge steam flow rate is determined by the requirement for an overall enthalpy balance, with the fluid being cooled to its original temperature, T_{10}, in preparation for another cycle. If superheated output steam is desired, its temperature is assumed to be α below the hot fluid temperature, T_{13}, and again the overall enthalpy balance determines the now somewhat smaller discharge steam flow rate.

Mass conservation of the heat storage fluid, given the relative duration of the charge and discharge phases, establishes the fluid flow rate on discharge, $\dot{W}_{fd}$. This flow rate, the discharge steam flow rate, and the temperatures at the nodes of the steam generator permit computation of the performance characteristics (R, ε, N_{tu}, and UA) in an entirely analogous manner to that described for the charge heat exchangers.

In summary, the calculation described defines the quantity and characteristics of the output steam that can be generated per unit of specified charge steam for a given temperature approach and fluid-to-charge-steam mass ratio. The heat transfer area of all the exchangers involved is also determined.

The complete concept analysis of the steam generator type of TES system requires two additional kinds of calculations: one defines the power swings of the composite (main plus peaking unit) steam plant as a function of the charge and discharge steam flows, and the other provides the sizing and costs of the heat storage medium. The first calculation is accomplished by the set of computer codes which model the various steam cycles and plant configurations; these are described earlier in this section. The related up- and downswings of system power and the relative duration of the charge and discharge phases determine the turnaround efficiency of the storage cycle.

Energy Storage. For the base case, the energy storage calculation assumes the use of rock and gravel packed-bed thermocline tanks with a bed volume fraction of 0.75, operated in the filled mode so that the fluid volume fraction is 0.25. Cost sensitivity excursions about the base case are made by varying the bed volume fraction from zero, ie, an all-fluid storage medium with no packed-bed, to unity, ie, an "all-bed" or drained-tank storage medium.

The weight fraction of rock in the storage medium, x_r, is given by

$$x_r = \frac{y_r \rho_r}{y_r \rho_r + (1 - y_r) \rho_f} \tag{5-10}$$

where y_r = volume fraction of rock in storage medium

ρ_r, ρ_f = density of rock, fluid taken at mean cycle temperature.

The specific enthalpy change of the dual media when cycled between the discharged (low) temperature and its charged (high) temperature is then

$$\Delta h_m = x_r \ \Delta h_r + (1 - x_r) \ \Delta h_f \tag{5-11}$$

where Δh_r, Δh_f are the specific enthalpy changes of the rock and fluid, respectively, between the same limiting temperatures, calculated by integrating the heat capacity expression with respect to temperature.

Given the total energy (enthalpy) to be stored, ΔH, as determined by the duration of charging and the heat flow, the total weight of storage medium required is then $\Delta H/\Delta h_m$, from which the weights and volumes of the rock and fluid components are directly calculated. The sum of the component volumes, based on each component's density at the high storage temperature, determines the total tankage volume.

Cost Estimating. Relationships were developed for the three main components of sensible heat storage systems: the heat exchangers, the tanks, and the heat storage media. Two costing approaches were used for the heat exchangers: the method given by Guthrie (Reference 216) and a simplified expression derived from feedwater heater cost data contained in the NUREG-0241/2/3/4 reports (References 92,93,211,212). Guthrie's method estimates a base cost as a function of heat transfer area and modifies this by factors reflecting design type, tube pressure, shell pressure, shell/tube materials, cost escalation, and installation labor and material factors to obtain direct costs. The simplified expression takes account only of heat transfer area and design pressure, and takes the form

$$\begin{aligned} C &= 300 \ (A/929)^{0.67} \ (P/6.9)^{0.6} \quad \text{(Metric)} \\ &= 300 \ (A/10^4)^{0.67} \ (P/10^3)^{0.6} \quad \text{(English)} \end{aligned} \tag{5-12}$$

where C = heat exchanger direct cost (thousands of 1976 dollars)

A = heat transfer surface area m^2 (sq ft)

P = design pressure MPa (psia).

Comparison of the two cost formulations for the types and sizes of heat exchangers required indicates that they are in good agreement for design pressures below about 5 MPa (700 psia), but that the pressure dependence of the simple formula is too extreme above this value. Consequently, the simple formula is used at the lower pressures and the Guthrie approach at the higher. In the analysis, individual heat exchangers were limited in size to a maximum surface area of 2800 m^2 (30,000 ft^2) per unit. This is achievable in a counterflow, tube and shell unit of 1.8 m (6 ft) o.d. and 14.6 m (48 ft) length using 0.025 m (1 in.) tubes with a triangular pitch of 1.25 times the tube diameter. The cost of multiple units, when needed, is taken as the same multiple of the unit cost.

The cost of storage tanks is based on the estimating relationships given by Guthrie (Reference 216) for large, field erected, welded storage tanks with conical roofs to API specifications. Assuming a nominal size tank as 40 m (131.2 ft) in diameter and 10 m (32.8 ft) high with a capacity of 12,190 m^3 (430,000 ft^3), an estimate of the cost of insulation was made and incorporated as a constant factor for tanks of all sizes. The direct cost of the nominal size tank was found to be $295,700 in 1976 dollars; when required, multiple tanks are costed as multiples of the unit cost.

PARAMETERS OF SYSTEM PERFORMANCE. In the analysis of TES systems appropriate for plant #1, intermediate pressure steam is used to charge the storage system for reasons having to do with the design and operation of the furnace, the boiler, and the HP-turbine, as discussed earlier in Section 5. This IP-steam is at 306°C (584°F) and 4.86 MPa (705 psia). Saturation temperature at charge steam pressure is 262°C (504°F), so there is about 44°C (80°F) of superheat. Condensate is to be subcooled to 217°C (423°F) before being pumped back to boiler entry pressure.

Because of the relatively low maximum temperature encountered in the cycle, a hydrocarbon heat transfer oil such as Exxon's Caloria HT-43 is the fluid of choice on the basis of its high heat capacity to cost ratio, given that its operating temperature range is not exceeded. Thus, the baseline system configuration to be investigated for plant #1 employs Caloria HT-43 as the heat transfer fluid, and rock-bed thermocline tanks with a 75 percent bed volume, operated filled with oil as the heat storage medium to reduce the system cost below that of an all-oil system.

Once the charge steam and system configuration are defined, the remaining parameters are temperature approach, α, and fluid to steam flow ratio, M_c. These quantities are varied systematically, α between 2.8°C (5°F) and 11.1°C (20°F) in steps of 2.8°C, and M_c between about 8 and 20 (subject to the limiting values which correspond to slopes of the oil temperature profile which violate the specified temperature approach at one or another pinch point). In addition to the α and M_c variation, the computation is performed for two system configurations; one employing a desuperheater heat exchanger, and one with a spray desuperheater or attemperator. For each case, the output data comprise the state variables of the steam or water and the heat transfer oil at each system node, and the effectiveness, number of thermal units, and UA-product of each heat exchanger.

Figures 5-20, 5-21, and 5-22 display the results of this analysis by showing the discharge steam pressure, the (normalized) discharge steam flow, and the turnaround efficiency, respectively, as a function of the oil to charge steam ratio, M_c, for various values of the temperature approach, α. Note that the discharge steam pressure (as a measure of thermodynamic availability) increases with M_c: as the fluid temperature profile becomes more nearly horizontal, the boiling temperature and saturation pressure increase. It decreases with α: as the fluid profile is depressed, the boiling temperature decreases. The discharge to charge steam ratio of Figure 5-21 displays a completely inverse dependence as it must to satisfy the constant enthalpy change condition: recall that all these results derive from the same flow of charge steam of a specified state, and differ only in the heat exchanger (α) and the oil flow rate (M_c). As is to be expected, the

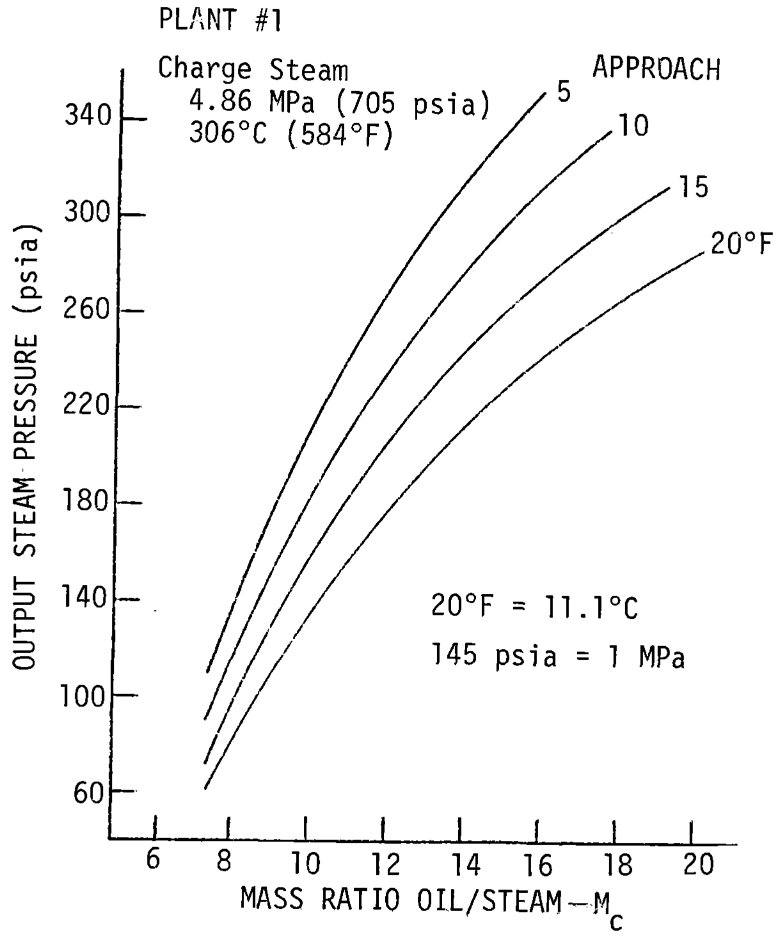


Figure 5-20. Output steam pressure, p, versus fluid/charge steam ratio, M_c.

turnaround efficiency of the storage process, Figure 5-22, parallels the thermodynamic availability of the steam; a large oil-to-charge-steam ratio and a small temperature approach makes for a more efficient, albeit a more expensive, system.

SELECTED SYSTEM CHARACTERISTICS. Comparing the parameterized systems under constant conditions of 50 percent swing, 8-hour charge, and 6-hour discharge periods, and taking account of the cost of both the main and peaking plant components, a minimum cost system configuration can be selected to represent the sensible heat storage, steam generation class of TES systems. The parameters of the selected system are summarized in Table 5-12 and of its heat exchangers in Table 5-13; the

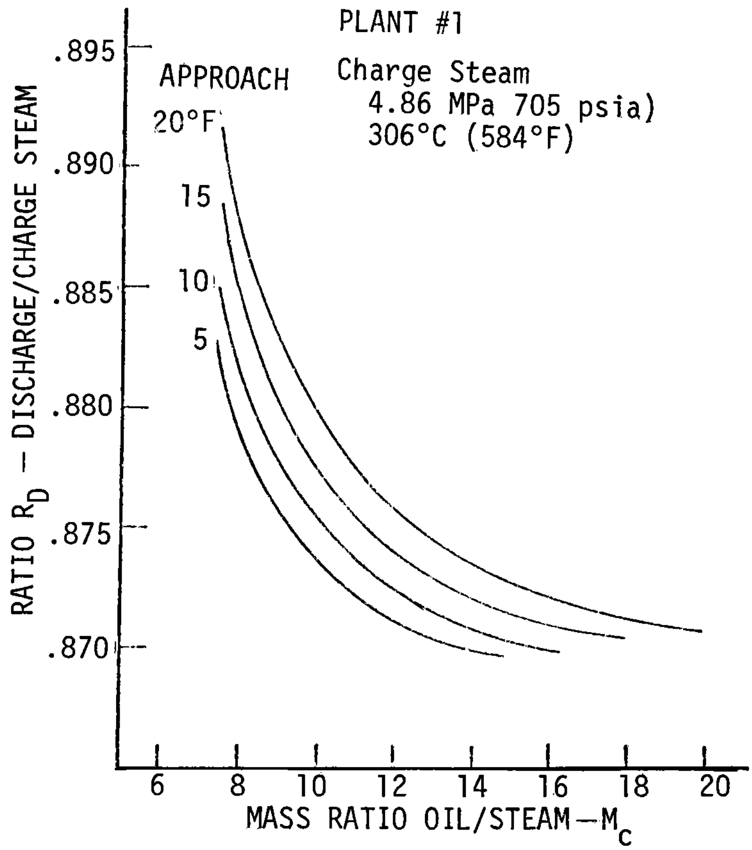


Figure 5-21. Discharge/charge steam ratio, R_d, versus fluid/charge steam ratio, M_c.

Table 5-12. Selected system characteristics.*

Configuration:	Charge steam attemperator
Heat transfer fluid:	Oil (Caloria HT-43)
Temperature approach:	5.6°C (10.0°F)
Oil to charge steam ratio:	15
Storage unit:	Filled thermocline tanks, granite gravel packed bed, bed volume fraction 0.75
Charge steam rate:	$2.0 \cdot 10^6$ kg/hr ($4.4 \cdot 10^6$ lb/hr)
Discharge steam rate:	$2.2 \cdot 10^6$ kg/hr ($4.9 \cdot 10^6$ lb/hr)
pressure:	2.02 MPa (292 psia)
temperature:	251°C (484°F)
Turnaround efficiency:	83 percent
Storage tanks:	16 units
diameter:	42.5 m (139 ft)
height:	12.7 m (42 ft)
volume:	$18.1 \cdot 10^3$ m^3 ($4.8 \cdot 10^6$ gal)
Gravel (total):	$560 \cdot 10^6$ kg ($616 \cdot 10^3$ tons)
Oil (total):	$57.5 \cdot 10^3$ m^3 ($15.2 \cdot 10^6$ gal)

* For 8-hr charge, 6-hr discharge, 50% swing, Plant #1.

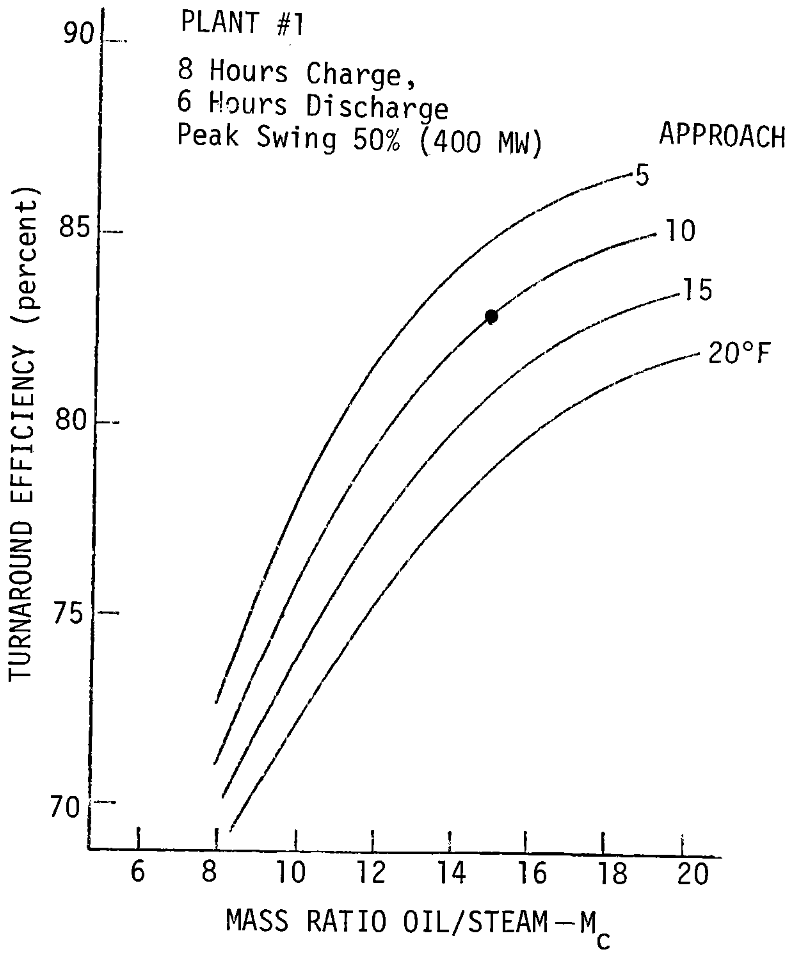


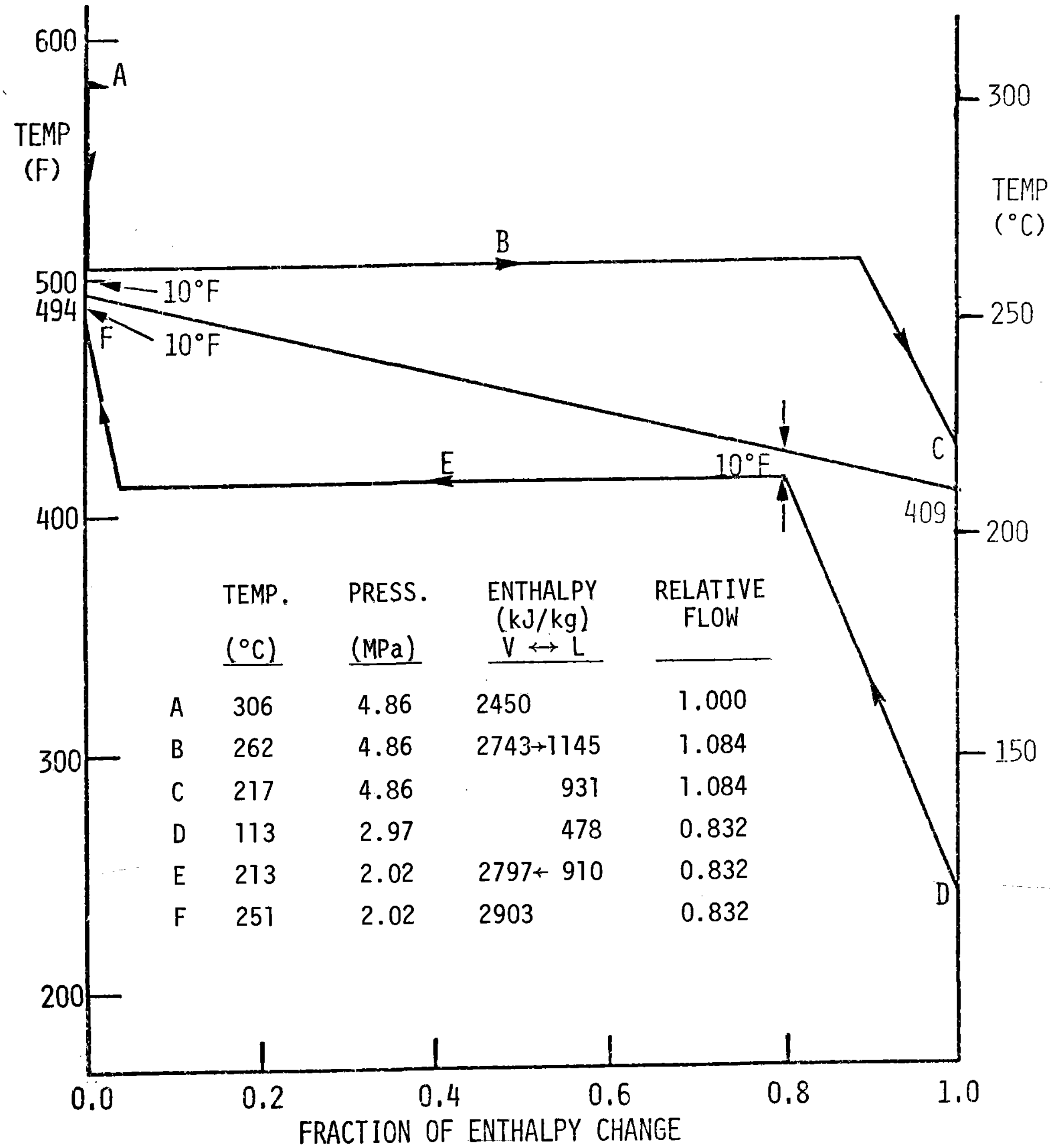
Figure 5-22. Turnaround efficiency, η, versus fluid/charge steam ratio, M_c.

Table 5-13. Selected system heat exchanger characteristics.*

Unit	Effectiveness	Required Number	Area per Unit (10^{+3} m^2)	(10^{+3} ft^2)
Condenser	0.882	35	2.78	29.9
Subcooler	0.851	5	2.74	29.5
Preheater	0.947	5	2.89	31.1
Boiler	0.867	30	2.52	27.1
Superheater	0.873	5	1.99	21.4

*For 8-hr charge, 6-hr discharge, 50% swing, Plant #1.

temperature profiles of the system are shown in Figure 5-23 along with the steam/water state points. Direct cost of all heat exchangers is 30.6 million dollars; of the storage tanks and media, 28.2 million dollars.



	TEMP. (°C)	PRESS. (MPa)	ENTHALPY (kJ/kg) V ↔ L	RELATIVE FLOW
A	306	4.86	2450	1.000
B	262	4.86	2743→1145	1.084
C	217	4.86	931	1.084
D	113	2.97	478	0.832
E	213	2.02	2797← 910	0.832
F	251	2.02	2903	0.832

Figure 5-23. Temperature profile for selected system example.

Feedwater Heating Systems

Of the various configurations identified as concepts, the one chosen for modeling is thermodynamically simplest and involves the smallest number of special components (heat exchangers), but would have a number of practical drawbacks if actually implemented. As modeled, however, it should be the least expensive version of this type, and so should compete most favorably among alternative systems. The application described here is evaluated for plant #2, the 1140 MW LWR.

SYSTEM DESCRIPTION. The system employs Caloria HT-43 oil as a heat transfer medium and rock and gravel packed-bed thermocline tanks kept filled with oil as the heat storage medium. As shown in Figure 5-24, cold oil is drawn from the bottom of the tanks at temperatures below 93°C (200°F) during the charge (off-peak) phase of the cycle, and is passed through a separate circuit in the feedwater heaters or separate train of heaters of similar design in parallel with the normal feedwater return flow, where it is heated by the increased flow of extraction steam caused by its presence. The oil circuit enters the feedwater heater chain above the lowest pressure heater (which is physically located in the condenser), where the feedwater is at about 80°C (177°F), passes through five heaters in series, and leaves the highest pressure one at 227°C (440°F), the same temperature as the feedwater being returned to the nuclear steam supply system. To increase the oil temperature above this point, it is passed through a "trim heater" fed from the main steam line at 283°C (541°F) where its temperature is raised to 238°C (460°F) to provide for the 11°C (20°F) temperature approach assumed for the discharge heat exchanger.

From the trim heater, the hot oil is directed to the top of a discharged thermocline tank where it transfers its heat to the rock bed as it flows downward, leaving as cold oil to repeat the circuit.

During the discharge (on-peak) phase of the cycle, the turbine's output power is increased by diverting a fraction of the return feedwater flow from its normal path through the extraction heaters to the

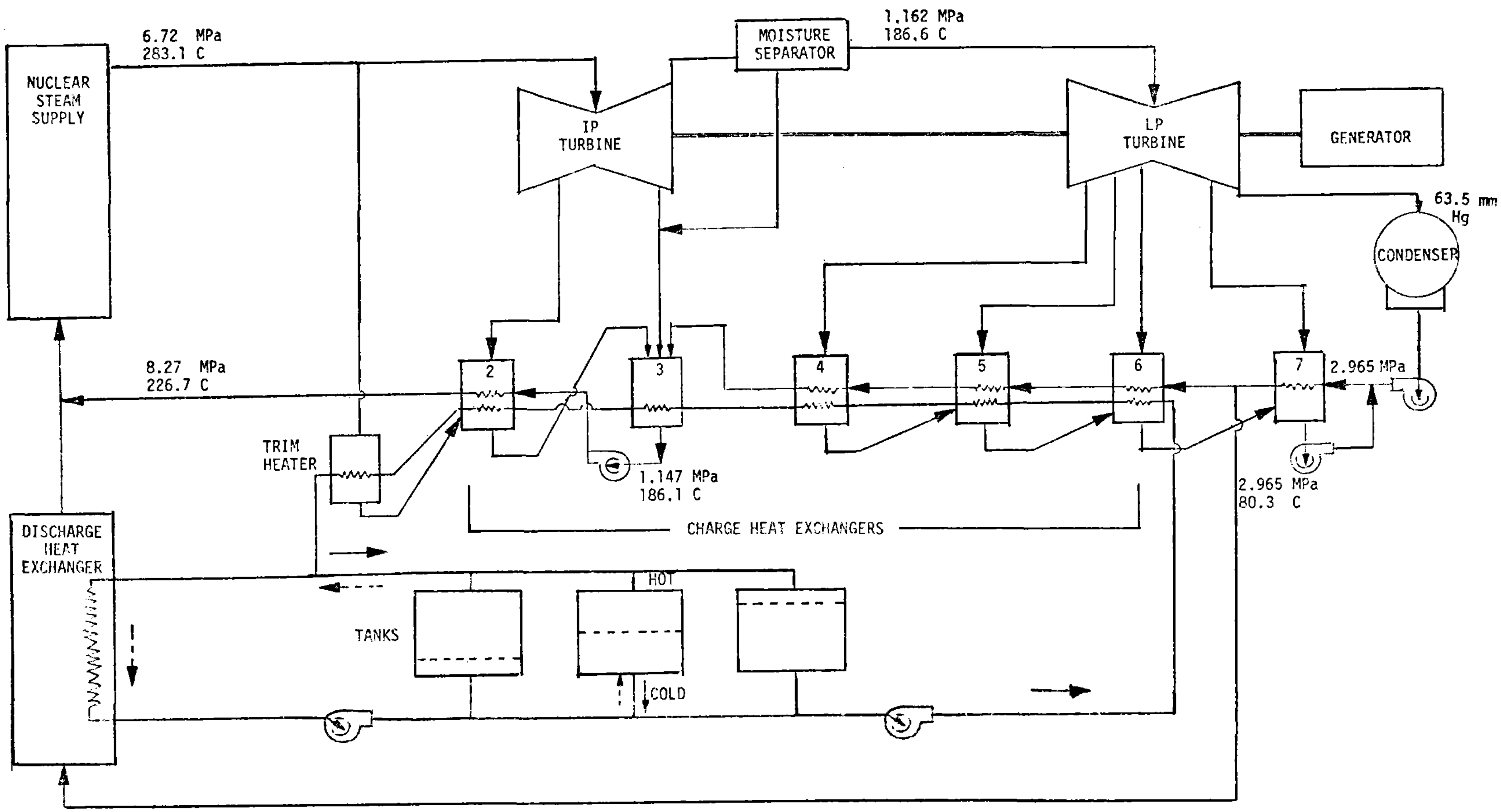


Figure 5-24. Feedwater heating TESS for plant #2.

TES system discharge heat exchangers, where it is heated to boiler entry temperature in countercurrent flow against the hot oil drawn from the top of charged thermocline tanks. A separate feedwater pump in the diverted flow line raises the pressure to its boiler entry value of 8.3 MPa (1200 psi).

SYSTEM PERFORMANCE. The calculations referred to here are based on maximizing the power swing of the main turbine. As the fraction of feedwater to be reheated by the hot oil increases, that which passes through the regular extraction heaters decreases until a point is reached at which the feedwater flow is incapable of accepting the enthalpy of the flow from the moisture separator (following the IP turbine exhaust) without violating the fixed boiler-feedwater-input conditions. This effect is increased by the fact that the reduced feedwater flow leads to reduced extraction of flows from the IP turbine which in turn result in a larger quantity of wet steam at the IP turbine exhaust and a consequent increase in the amount of separated moisture.

Assuming the discharge heat exchanger to be characterized by the same 11°C (20°F) temperature approach at both ends, this limit is reached at a feedwater flow of approximately 5455 Mg/h (12 million lb/hr) heating by the oil and 1364 Mg/h (3 million lb/hr) heating in the extraction heaters. At this point, the limiting discharge power swing is +17.6 percent and the requisite oil flow rate is 10,100 Mg/h (22.27 million lb/hr). To heat enough oil for the assumed 6-hour discharge period during the assumed 8-hour charge period requires a downswing in power of -15.4 percent, corresponding to a turnaround efficiency of 85.3 percent.

SYSTEM CHARACTERISTICS. The heat exchanger characteristics required for this feedwater heating system can be derived from the hot and cold stream temperatures and flow rates indicated by the thermodynamic model of the system. From these data, the heat exchanger effectiveness, number of thermal units rating, overall heat exchange area, and cost are determined; the results are presented in Table 5-14.

Table 5-14. Heat exchanger characteristics for plant #2 feedwater heating system.

Heat Exchanger	Discharge Phase Feedwater Heater		Charge Phase Extraction Heaters		Charge Phase "Trim" Heater	
Hot Stream Fluid	Oil		Condensing Steam		Condensing Steam	
Temp., inlet, °C (°F)	238	(460)	107 - 234	(224 - 453)	283	(542)
outlet, °C (°F)	92	(198)	86 - 193	(187 - 379)	238	(460)
Flow, 10^6 (kg/hr (lb/hr))	10.14	(22.3)	0.48	(1.06)[a]	0.13	(0.28)
Cold Stream Fluid	Water		Oil		Oil	
Temp., inlet, °C (°F)	81	(178)	92	(198)	227	(440)
outlet, °C (°F)	227	(440)	227	(440)	238	(460)
Flow, 10^6 (kg/hr (lb/hr))	5.45	(12.0)	7.59	(16.7)	7.59	(16.7)
Capacity Rate Ratio[b]	0.987		0.0		0.0	
Effectiveness	0.929		0.948		0.197	
Number of Thermal Units	12.06		2.95		0.22	
Overall Heat Transfer Coefficient U, $W/m^2 \cdot C$	483		522		522	
($Btu/h \cdot ft^2 \cdot F$)	(85)		(92)		(92)	
Heat Transfer Area, 10^3 (m^2 (ft^2))	164	(1760)	28	(300)	2.3	(25)
Direct Cost, M$ (1976)	19.490		3.306		0.311	

[a]Total flow to five extraction-steam oil heaters.

[b]The minimum capacity rate stream in each case is the cold stream.

The discharge oil-to-feedwater heat exchanger is larger and more expensive than the charge unit for three reasons: it is a sensible heat exchanger between two condensed fluid phases operating under almost balanced conditions (capacity rate ratio $\approx$ 1) rather than a latent heat exchanger taking advantage of a phase change; it is sized for the higher flow rate of the six-hour discharge portion of the cycle rather than the eight-hour charge portion; and the overall heat transfer coefficient for the water-oil interface is less than that for the condensing steam-oil case. Another factor accounting for the much greater cost of the discharge heat exchangers is the 8.3 MPa (1200 psi) pressure rating as compared with the graduated pressures of the charge heaters ranging downward from 6.7 MPa for the small trim heater to 3.0 MPa at the last extraction heater and less than 140 kPa at the first heater. Cost estimates of these heat exchangers are based on data and procedures presented by Guthrie (Reference 216), employing multiples of a basic module having 2800 m^2 (30,000 ft^2) of heat exchange surface.

The energy storage components of this system are rather modest by comparison with the heat exchangers. Assuming packed rock bed thermocline tanks with a 25 percent void volume filled with oil, the requirement is for $123 \cdot 10^6$ kg ($136 \cdot 10^3$ tons) of gravel for 2.05 M$; $14.4 \cdot 10^3$ kg ($3.8 \cdot 10^6$ gal) of oil for 2.89 M$; and $63.5 \cdot 10^3$ m^3 ($16.7 \cdot 10^6$ gal) of tankage for 1.53 M$; for a direct cost of 6.47 M$.

ECONOMIC MODELING

The objective of economic modeling (really cost modeling) is to put together the costs of the components of the TESS concept configurations being compared, along with the costs of required modifications of the peaking turbine and other Turbine Island accounts, in a uniform procedure for comparative evaluation. The parameter for economic comparison is the incremental cost in dollars per kilowatt ($/kW) incurred in adding the TESS to the modified reference plant. That is, the sum of all increments of capital cost is divided by the increment in peaking capacity that is provided.

As discussed in Section 4, in connection with Tables 4-2 and 4-5, results will be presented at the TOTAL Cost level with all adders, compatible with the $/kW costs given in EPRI's TAG (Reference 172) for other generating capacity. At the component level it is convenient to start with direct costs (installed costs), and convert to TOTAL Costs for the system level using the factor given in Table 4-2.

The use of $/kW gives a convenient comparison independent of size of plant and the magnitude of the peaking swing added to the reference plant. It will of course vary somewhat with economies of scale. For comparability of costs with other forms of storage it is instructive to separate the system cost into a portion that is power related and a portion that is energy related. The former includes the cost of the peaking turbine (or incremental costs for modifying the main turbine), and heat exchangers, evaporators, pipes, pumps, etc which are energy-flow and mass-flow dependent. The latter includes the costs proportional to the energy stored such as the storage media, and tanks or containment. Using the nomenclature:

C_T = Total costs charged to TESS concept, \$/kW

C_{ES} = Energy related costs of storage, \$/kWh

C_{PS} = Power related costs of storage, \$/kW

C_{PP} = Costs of incremental power capacity, \$/kW

H = Equivalent number of hours of storage discharge at full rate

C_L = A capital cost equivalent of the turnaround efficiency, \$/kWh ($C_L \cdot H$ in \$/kW will be called L)

the system cost is:

$$C_T \text{ (\$/kW)} = (C_{PS} + C_{PP}) + (C_{ES} + C_L) \cdot H \quad . \tag{5-13}$$

The first term is the power-related cost, C_P. The costs of the incremental Turbine Island capacity are separated from storage costs because they are in general independent of the internal details of the TES system and common to a number of concepts that have the same interface parameters between TESS and the baseline plants.

The second term is the energy-related cost, C_E. If the energy stored can supply the incremental turbine capacity for H hours, multiplying by H converts this term to \$/kW also.

The customary comparison of storage systems is in terms of capital costs and turnaround efficiency separately. For a fairer and easier comparison, a way of expressing the energy loss from the turnaround efficiency as a capital cost term is derived in this section. Since the energy lost per kW of capacity is proportional to the hours of discharge, it is properly part of the energy related cost.

Cost Comparisons with Baseline Plants

Turnaround efficiency, as the ratio of the electricity generated from stored thermal energy divided by the electricity not produced because of the diversion of thermal energy to storage, clearly requires thermodynamic comparisons between the daily electric output attainable with the baseline plant operated in its normal mode and the same plant operated with a daily charging and discharging of storage. The term baseline plant, rather than reference plant, is used here to indicate the reference plants modified as described in Section 4 to model and meet the needs of storage.

Significant technical modifications were made, such as elimination of reheat, which modify thermodynamic performance, but not, it is believed, enough to modify the ranking of concepts to be compared.

Similarly, the cost elements of the baseline plant will not be identical to those of the reference plants as defined in Tables 4-1 through 4-5. It was pointed out that the elimination of reheater tubes should significantly reduce the cost of the coal-fired steam supply, and hence that of the Fixed Plant. At the same time the mass flow of steam through the turbine set must increase to produce the same electric output, so the cost of many parts of the Turbine Island will increase. To a first approximation these cost differences cancel and can be neglected for this comparison.

The cost of the IP and LP turbine are particularly affected by the increase in mass flow, since the costs of the condenser, the heat rejection system (cooling towers), the feedwater heaters, and the turbine itself are all roughly proportional to the energy- or mass-flow through them. The electric plant and some miscellaneous parts of the Turbine Island (eg Instrumentation and Control) are not flow sensitive, so partially mitigate the cost increase in $/kW for the Turbine Island. For flow increases of 10 to 20 percent for the nuclear and coal-fired reference plants, as estimated in Section 4, the cost increase in the Turbine Island for the baseline plant will be only 6 to 12 percent. This will be neglected, and the cost elements in Section 4 will be used. This assumption is not likely to alter rankings and is certainly favorable to storage, since the peaking turbine costs will be related to the baseline system Turbine Island costs through the ratio of peaking turbine flows to baseline system flows.

BASELINE PLANT — \$/MWH. The cost of electricity in a baseline plant in its normal (non-storage) mode is:

$$COE = \frac{C_{Fixed} + C_{Variable} \cdot CF}{8760 \cdot CF \cdot P_{MW}} 10^{+6} \ (\$/MWh) \tag{5-14}$$

or

$$= \left(\frac{C_{Fixed}}{CF} + C_{Variable} \right) \frac{10^3}{8.76 \cdot P_{MW}} \tag{5-14a}$$

where C_{Fixed} are the annual fixed costs in M\$

$C_{Variable}$ are the annual variable costs in M\$ (for 1.00 capacity factor)

CF is the capacity factor or fraction of the rated annual energy output that is produced.

It was indicated in Section 4 that the availability or maximum capacity factor achievable was assumed as 0.723 for reference plants 1 and 2. If a baseline or reference plant is used for load following it is convenient to separate the availability from the capacity factor:

$$F = CF/0.723$$

Where F is the fraction of the *available* energy produced. As an example, referring to Table 4-5, plant #1:

$$COE_{Baseline} = \frac{110.3 + 115.7F}{8760 \cdot 0.723 \cdot F \cdot 800} \cdot 10^6 = \frac{21.77}{F} + 22.83 \ (\$/MWh) \tag{5-15}$$

or a minimum of 44.60 \$/MW when F = 1.0.

BASELINE/TESS PLANT — \$/MWH. In a plant incorporating storage, a specific cost of electricity can be similarly derived, assuming a cycle of operation of storage charge and discharge that corresponds to the capacity factor CF, or the factor F defined above. The TESS plant will be assumed to use the Fixed Plant components at the maximum availability level, ie operating at full rated output whenever available.

The Turbine Island components are augmented by the addition of a peaking turbine, similar to the tandem IP and LP turbines of the main turbines of the baseline plant, or its equivalent in increased capacity of the main turbines. As discussed earlier in this section, the normalized parameters describing the magnitude of the peaking addition is called the swing, power swing, upward swing: for example a swing of 0.5 or 50 percent would be 400 MW in the 800 MW baseline plant. It should be noted that in this plant the 400 MW is not 50 percent of the main IP and LP turbine but of the total turbine complement including the HP turbines. In the baseline plant, the HP turbine output is over 300 MW because of the increased mass flow of steam. Thus, the 400 MW peaking turbine is almost as big as the 500 MW main IP-LP turbines. The parameter p will be used to express the swing as a fraction.

The TESS plant will include all the capital equipment of the baseline plant, plus the additional costs of the TESS system and the peaking turbine. The cost of electricity in the storage mode is:

$$COE_{TESS} = \frac{C_{Fixed} + C_{Variable} \cdot CF + C_1 \cdot C_S}{8760 \cdot CF \cdot P_{MW} (1 - C_2)} 10^6 (\$/MWh) \qquad (5\text{-}16)$$

where $C_S = (C_{pp} + C_{PS}) + C_{ES} \cdot H$, the sum of storage components in \$/kW.

C_1 is the constant needed to convert the storage cost components to annual costs in M\$. For the 800 MW baseline plant, this is the product of $pP_{MW}/1000$, the peaking capacity in GW; the fixed charge rate, 0.18; the factor to include fixed O&M, 1.032; and if C_S is in direct costs, the factor 2.16 to convert to the TOTAL cost level. If the storage costs are already converted to TOTAL costs, as will now be assumed in this analysis, the last factor will be omitted, and $C_1 = 0.149$.

The denominator of Equation 5-16 represents the annual energy out of the baseline/TESS plant. The fixed plant is assumed to operate at its maximum availability so that CF is 0.723. The electric energy out of the peaking turbine is

$$E = p P_{MW} \cdot (H/24) \cdot 8760 \cdot 0.723 \qquad (5\text{-}17)$$

and electric energy lost during storage charging is E/η. Therefore C_2, the fractional reduction in plant output due to the turnaround efficiency is $(\frac{1}{\eta} - 1) \cdot p \cdot (H/24)$. Retaining C_2 for the moment but inserting other values for plant #1 from Equation 5-15 gives:

$$COE_{TESS} = \frac{110.3 + 115.7 + 0.149\ C_S \cdot p}{8760 \cdot 0.723 \cdot 800\ (1 - C_2)} \cdot 10^6$$

$$= (21.77 + 22.83 + 0.0293\ C_S \cdot p)/(1 - C_2)\ (\$/\text{MWh}) \qquad (5\text{-}18)$$

If both baseline and TESS plants are to operate at the same capacity factor, load following daily with the same ratio of peak output to average output, then p and F are related:

$$\frac{\text{Peak Power}}{\text{Average Power}} = \left(\frac{1}{F}\right) \text{ for baseline} = \frac{(1 + p)}{(1 - C_2)} \text{ for TESS} \qquad (5\text{-}19)$$

For $COE_{BASELINE} = COE_{BASELINE/TESS}$ (\$/MWh)

$$\frac{21.77}{F} + 22.83 = \frac{21.77 + 22.83 + 0.0293\ C_S \cdot p}{(1 - C_2)} \qquad (5\text{-}20)$$

$$21.77 \left(\frac{1 - C_2}{F}\right) + 22.83 - 22.83\,C_2 - 21.77 - 22.83 = 0.0293\,C_S \cdot p \qquad (5\text{-}21)$$

$$21.77\,p - 22.83\left(\frac{1}{\eta} - 1\right) pH/24 = 0.0293\ C_S\,p \quad (\$/\text{MWh}) \qquad (5\text{-}22)$$

$$C_S = 743 - 32.0 \cdot H \cdot \left(\frac{1}{\eta} - 1\right) = 743 - C_L \cdot H\ (\$/\text{kW TOTAL cost}) \qquad (5\text{-}23)$$

It will be recognized that the first term is the specific capital cost (\$/kW) of the reference and baseline 800 MW plant (Table 4-2). The breakeven cost of the storage components must not exceed this value less a loss term, L, previously called $C_L \cdot H$. Thus, for breakeven:

$$C_T = C_S + C_L \cdot H = (C_{pp} + C_{PS}) + \left(C_{ES} + 32.0\left(\frac{1}{\eta} - 1\right)\right) \cdot H = 743 \qquad (5\text{-}24)$$

By similarly using the data on Reference Plant #2, the 1140 MW nuclear plant, the breakeven cost snd value of C_L are:

$$C_T = (C_{pp} + C_{PS}) + \left(C_{ES} + 28.5\left(\frac{1}{\eta} - 1\right)\right) \cdot H = 785 \qquad (5\text{-}25)$$

CAVEATS. The above analysis has accomplished two objectives: It has derived a loss term, L, with dimensions of \$/kW that will assist in comparing TESS that differ both in capital costs and in turnaround efficiency. It has also set a breakeven cost or target value to be met by TESS plants if they are to be preferred over load-following base load plants.

Not much importance should be placed on the second objective at this time in the study. The goal of this task is the comparison of TESS concepts with each other. Non-economic factors may make TESS or other storage systems preferable to load-following with large base load plants. The assumptions made include in the TESS turnaround efficiency the reduction in turbine efficiency at low load; the corresponding effect in a load-following plant is not included since the variable costs are assumed linear with capacity factor.

Peaking Turbine Cost Accounts

Costs for the peaking turbine and all associated power related equipment must be derived that are consistent with the cost data for the reference plants and for the other TESS costs derived in this section. The peaking turbine capacity is the largest power related component of TESS cost. In Table 4-4, the combination of accounts for the Turbine Island on Plant #2, 1140 MW nuclear, gave a direct cost of 167 $/kW. For Plant #1, 800 MW-HSC, by separating out a portion of those accounts associated with the high pressure turbine, a similar cost of 164 $/kW was derived for the remaining IP-LP turbines and other accounts of the Turbine Island. At the Total Cost level these are both 354 $/kW. This coincidence probably results from compensating effects of economies of scale favoring the nuclear plant, and better steam quality favoring the coal-fired plant.

It was noted that modifications made to the steam cycle for the baseline/TESS plants, such as eliminating reheat, would probably increase these turbine costs by 6 to 12 percent, but that this would be neglected, as common to all TESS concepts considered.

The Turbine Island configuration is considered the prototype for the peaking turbine; the same 354 $/kW will be used if the peaking turbine is operated under steam conditions comparable to the main turbine. However, the different TESS concepts considered and the different parameters explored within each concept will alter the steam conditions for the turbine, hence alter its design to produce the desired output. This effect must be considered in the comparison of concepts.

For HTW storage concepts in which part of the water is flashed to steam during storage discharge, saturated steam at about constant pressure and temperature is delivered to the peaking turbine. A throttle between storage and the turbine assures the constant pressure for constant turbine output. This constant, throttled pressure must be lower than the HTW storage pressure. The lower the pressure the larger the fraction of the HTW that can be flashed to steam, and the higher the storage density in kWh/meter^3. But the lower the saturated steam pressure, the greater the steam mass flow rate required per kilowatt of electric output from the turbine generator. The cost in $/kW of a number of the cost elements of the Turbine Island are almost directly proportional to the mass flow. As the turbine inlet pressure decreases, the specific cost of the peaking Turbine Island will increase.

There is a similar decrease in the turbine inlet pressure from charge steam used for storage in a sensible heat storage system, eg oil/rock, and the discharge steam deliverable from the storage output heat exchangers. In this case, however, TESS design may provide some superheat in the reduced pressure steam delivered to the turbine.

Only a rough estimate of the variation of peaking Turbine Island specific cost can be derived, as detailed turbine plant redesign and costing for each input steam condition is not feasible for this screening.

FLOW DEPENDENT ACCOUNTS. Between the extremes of considering the peaking Turbine Island costs as constant in $/kW and as proportional to the enthalpy flow per kilowatt at the condenser (hence roughly also in the cooling towers and feedwater train) is the more rational course of allocating the cost accounts to these extremes or some intermediate level.

The cost of the generator and the whole electric plant account #24 is clearly constant per kW (ie independent of steam quality). Table 5-15 indicates the relevant cost accounts for plants #1 (800 MW HSC) and #2 (1140 MW Nuclear), the direct costs in M$, the specific costs in $/kW (direct cost), and designates by A, B, C the share of each that is allocated to the HP turbine account (800 MW plant only), the peaking turbine plant, and the peaking electric plant.

Table 5-15. Cost account allocation.

Account	Plant #1			(M$)	Plant #2	
	A	B	C		B	C
21 Struct. & Improv.						
213 Turbine Bay	1.8	5.0	1.0		9.0	2.8
218 Misc. Struct.		1.4	0.2		5.8	0.3
23 Turbine Plant						
231 Turbine Gen.	7.9	11.9	11.8		41.4	20.0
233 Condenser		8.9			15.0	
234 Feedwater Htr.		10.8			15.0	
23X Misc. Aux.		10.4	3.5		14.0	5.9
24 Electric Plant	7.3		21.6			39.4
26 Heat Rej. Eq.		12.0			21.6	
Sum - Direct Costs	17.0	60.4	38.1		121.8	68.4
		98.5			190.2	
TOTAL Costs $/kW						
23A HP Turbine	184					
23B Peak. Turbine		217			227	
24C Peak. El. Plant			137			127
Sum		354			354	
Fraction		0.613	0.387		0.641	0.359

The result of the tabular analysis in Table 5-15 is that the electric plant component of cost, which is roughly constant in $/kW, is between 35 and 40 percent of the total, and the flow dependent account 23B is 60 to 65 percent of the total. Thus, if for some input steam condition at the peaking turbine inlet the enthalpy flow to the condenser per kilowatt hour is double that for the main turbine steam conditions, the incremental cost of peaking power capacity should be estimated at 1.63 x 354 or 577 \$/kW.

FLOW DEPENDENCE ON STEAM CONDITIONS. For the 800 MW baseline plant, cold reheat steam, which feeds the main IP turbine and is used to charge storage, is at conditions 4.8 MPa (700 psi), 307°C (583°F), 2946 kJ/kg (1269 Btu/lb). Similarly, the 1140 MW nuclear plant steam condition, used for the main turbine and to charge storage is 6.8 MPa (980 psi), 283°C (541°F), 2765 kJ/kg (1191 Btu/lb). The latter is saturated, the former has about 45°C of superheat.

Both for HTW storage and reconversion to steam by evaporators, and for sensible heat storage in oil, rock, molten salts, etc, there is a drop in pressure from the charge steam to the discharge steam. For a reasonable specific output (in kWh electric per cubic meter) the drop is at least two to one. When, to increase specific output, two- or three-stage evaporators are used, the pressure drop may exceed ten to one. Each factor of two drop in pressure decreases the work output available per kg of steam by a roughly constant amount. Thus, steam expanding from 10 MPa to 10 kPa (a factor of 1000 is about 2^{10}) will produce about 10 percent of its work output for each factor-of-two pressure drop. A mass flow of steam that enters the turbine at half the pressure will produce about 10 percent less power output.

Computer calculations of steam flow through the peaking turbine can give a better estimate of the power output per kg/hr of steam or its inverse the kg of steam per kWh output. This is a function of the steam input conditions expressed as pressure and specific enthalpy (kJ/kg), or its equivalent using temperature, degrees of superheat, or steam quality as a parameter. Figure 5-25 is derived from such runs for the plant #1 peaking turbine. The output scale is given both as the enthalpy flow through the condenser and heat rejection system per kWh of peaking output and as the equivalent estimated TOTAL cost of the incremental power capacity, C_{pp}, in \$/kW. The dashed line minimum indicates a constant \$/kW, and the maximum indicates the extreme if the turbine cost were exactly proportional to the enthalpy flow. Both

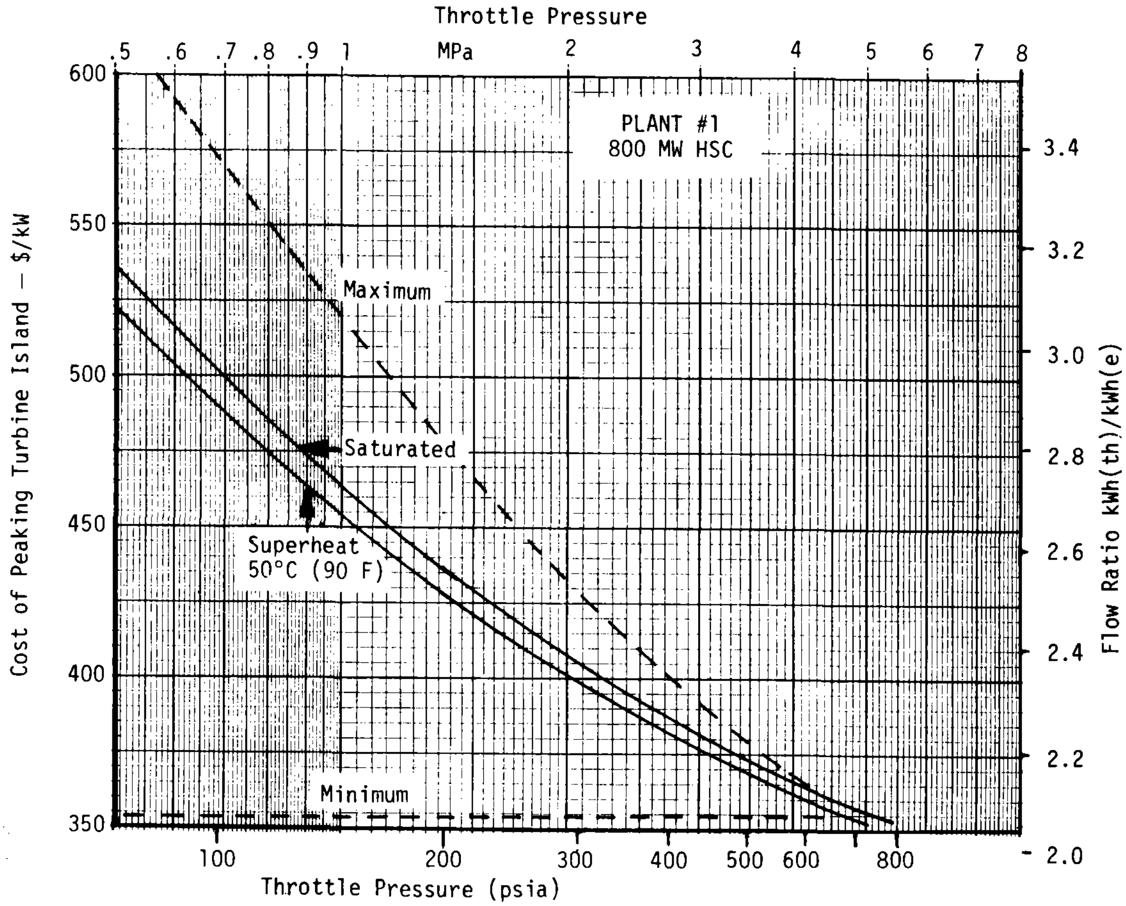


Figure 5-25. Specific cost of peaking Turbine Island for plant #1 as a function of throttle pressure.

saturated and one example of superheated steam input are given to show the effect of superheat on cost.

Figure 5-26 is the equivalent display for the plant #2 peaking Turbine Island.

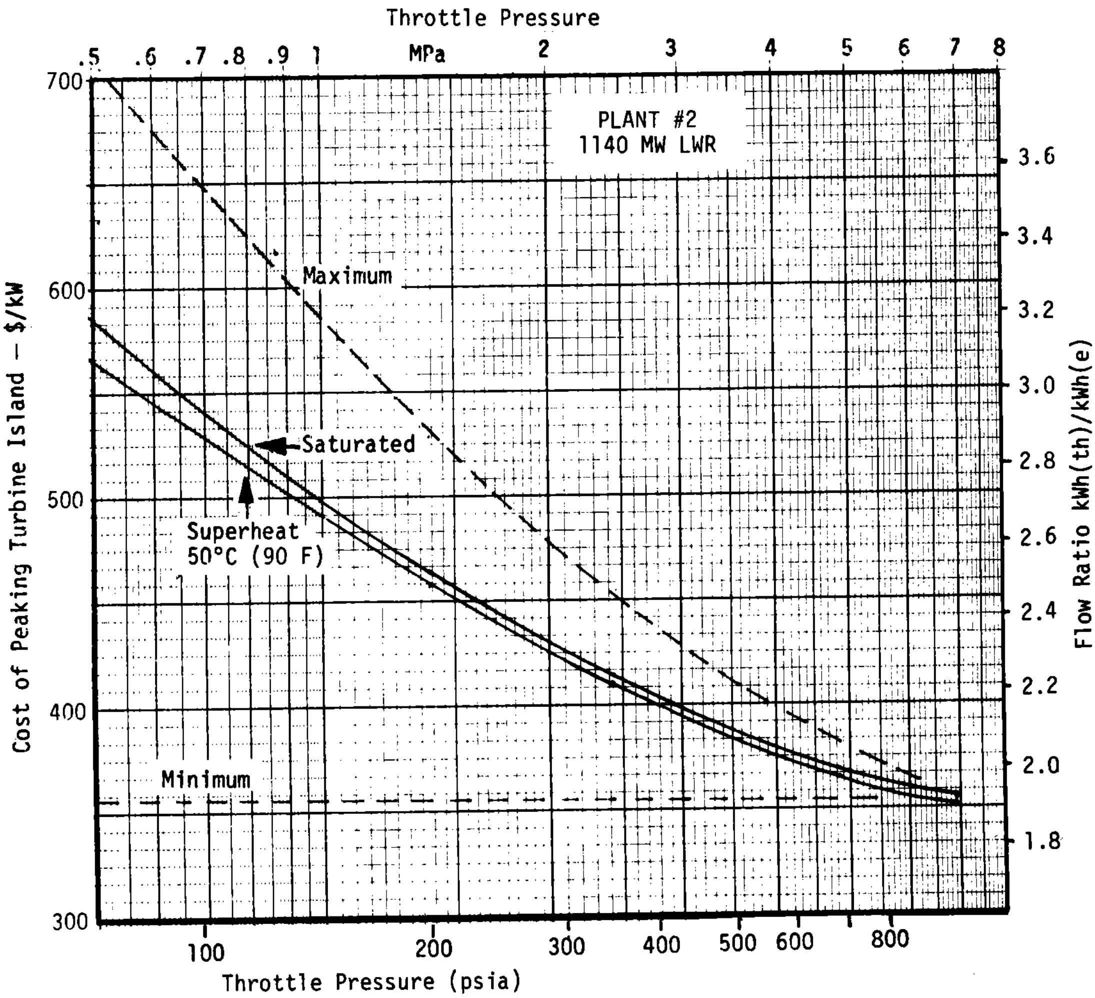


Figure 5-26. Specific cost of peaking Turbine Island for plant #2 as a function of throttle pressure.

SECTION 6
COMPARATIVE EVALUATION

In this section the preliminary selections described in Section 3 are compared, principally on performance and cost criteria. There is a large cost difference between alternative forms of containment and storage media, and therefore in the specific cost of TESS in dollars per kilowatt. All else being equal, the lowest specific capital cost (\$/kW) and the lowest specific energy cost (\$/MWh) are to be preferred. The next section will address the less quantitative criteria to evaluate where they have sufficient impact to alter the rank order on strictly economic criteria.

COMMON ASSUMPTIONS

For comparability, the candidate concepts selected, #1 to #12, must be evaluated on a common basis of assumptions. Many of these have been explicitly or implicitly stated earlier. In brief review, the major assumptions are:

- The methodology for comparing alternative forms of generation capacity is that described in the EPRI Technical Assessment Guide (TAG), (Reference 172, 1977). This includes use of TOTAL costs (see Section 4), a fixed charge rate for levelizing annual capital costs, and a levelizing factor to derive uniform annual fuel costs and O&M costs for assumed escalation scenarios.

- The total cost of reference plants, and plants #1 and #2 as modified to suitably interface TES systems, are based on the cost data in TAG. So are fuel costs, and O&M costs. Detailed cost accounts for the subsystems and components of plants are derived from a series called *Commercial Electric Power Cost Studies*, prepared for ERDA and the Nuclear Regulatory Commission by United Engineers and Constructors, Inc. Specifically

NUREG-0241 (Reference 93, *Capital Cost: Pressurized Water Reactor Plant*) is the basis for plant #2, 1140 MW LWR. NUREG-0243 (Reference 212, *Capital Cost: Low and High Sulfur Coal Plants — 800 MW_e*) is the basis for plant #1 — 800 MW HSC.

- Mid-1976 dollars are used for all costs, as in the above sources. The direct costs (installed costs) for the separate accounts in the NUREG series are converted to total costs compatible with the TAG by a multiplier (Total costs (TAG)/ Sum of all direct costs (NUREG)) which is virtually the same for plants #1 and #2: 2.16 and 2.12 respectively. Other capital costs, such as TESS components, will be converted from direct costs to TOTAL costs with the same factor. It is assumed that the rationale for loading all such components with the same adders subsumed in TAG TOTAL costs is as good as, and much simpler than, developing a set of adders for each component, yet having them comparable for concept evaluation.

  Total on peak used 1.4375

- The assumed escalation scenarios in TAG are for 6 percent annual inflation on capital costs and capital related O&M and installation costs, from 1976 to beyond 2020. A 10 percent discount rate and an 18 percent fixed charge rate are assumed as compatible with this general inflation. Each fuel has a specific escalation rate higher than 6 percent, so has a net or real escalation. Variable O&M is assumed to have the same escalation rate as the fuel, to keep them proportional. For analysis a plant starting operation in 1990 with a life of 30 years is assumed, ie even though 1976 dollars are used, the net escalation effects from 1976 to 1990 are included for cost elements not escalating at the general inflation rates, and the continuing escalation from 1990 to 2020 is included in the levelizing factors on variable costs.

- In the interfacing of a power plant with a TESS, two basic configurations suffice: that for steam generation and power production with a separate peaking turbine, as shown in Figure

5-2 for plant #1 and Figure 5-4 for plant #2; and that for feedwater heat storage, with increased main turbine capacity, as shown in Figures 5-3 and 5-5. Other possibilities, such as the use of a larger main turbine with steam generation concepts or a separate peaking turbine with feedwater heat storage concepts, would make at most minor differences in Turbine Island costs, insufficient to change relative rankings of TESS concepts.

- The range of peaking swings from 15 to 50 percent of nominal rated output has been explored. Most comparisons are made at 50 percent swing as potentially of more interest to utilities.
- As a means of comparing TESS selections, the specific capital costs ($/kW) of the TESS is the preferred measure. For comparability with other storage systems studies these costs are divided into energy-related and power-related portions.

HTW SELECTIONS

Selections 1 through 7 postulate high temperature water (HTW) storage. They differ principally in the means of containment of HTW under pressure, and in the mode of use of the pressure vessel as an accumulator. These options are largely separable; each form of pressure vessel can be considered with each mode of accumulator use for use with either plant #1 or plant #2. As the cost of the pressure vessel is a large fraction of the total TESS cost, the comparative cost of the vessel alternatives in Selections 1-7 will be developed before specific costs of TES systems are developed.

Specific Costs of Pressure Containment

The alternative forms of pressure containment for HTW in Selections 1 through 7 are:

	Selection
• Prestressed Cast Iron Vessels (PCIV)	1
• Prestressed Concrete Pressure Vessels (PCPV)	2
• Steel Pressure Vessels (Steel)	3
• Underground Cavity Containment (UG Cavity)	4,5,6
• Confined Aquifer Storage (Aquifer)	7

The cost of containment of HTW in these vessels is a function of both the design pressure and the volume. It is also a function of temperature. Pressure and temperature effects are closely correlated for saturated HTW so will be treated together. The cost versus volume relationship is not necessarily linear for a single pressure vessel, but when the volume required is many times the largest unit size believed to be practicable a linear relationship can be assumed. The measure of the cost of containment is dollars per cubic meter ($\$/m^3$).

PCIV. Prof. Paul V. Gilli in *Thermal Energy Storage Using Prestressed Cast Iron Vessels (PCIV)* (Reference 45), a 1977 study performed for ERDA/STOR, makes estimates on PCIV costs for a range of volumes and pressures. His baseline case, 8000 m^3 and 6 MPa, cost 1248 $\$/m^3$. The cost items listed (Reference 45, Table XIV, p 96) approximate the direct cost level. Transportation costs are specifically excluded, some items are included for erection and foundation, a small amount is included for engineering and testing.

Data given for other volumes and pressures can be approximated by: $\$/m^3 = 1248\ (0.953 + 376/V)\ (0.264 + 0.1226\ P)$, where V is in m^3 and P is in MPa. In order to display the comparative costs graphically, this relationship is shown on Figure 6-1 for the 8000 m^3 size. It will be noted from the above that only a few percent savings could be expected from larger size so 8000 m^3 will be taken as the module.

For 400 MW_e and 6 hours peaking, ie 2400 MWh stored, a volume of 120,000 m^3 could be required if the specific output, e_o, of a TES system were 20 kWh/m^3. Thus 15 modules of PCIV could be required of 8000 m^3 size.

PCPV. No proponent has specifically studied the use of prestressed concrete pressure vessels for containing HTW in the 3-10 MPa range. Cost data from several sources on PCPV versus pressure were located and compared as shown on Figure 6-1.

Ian Glendenning of the British Central Electricity Generating Board displays, in a study on compressed air storage systems using a

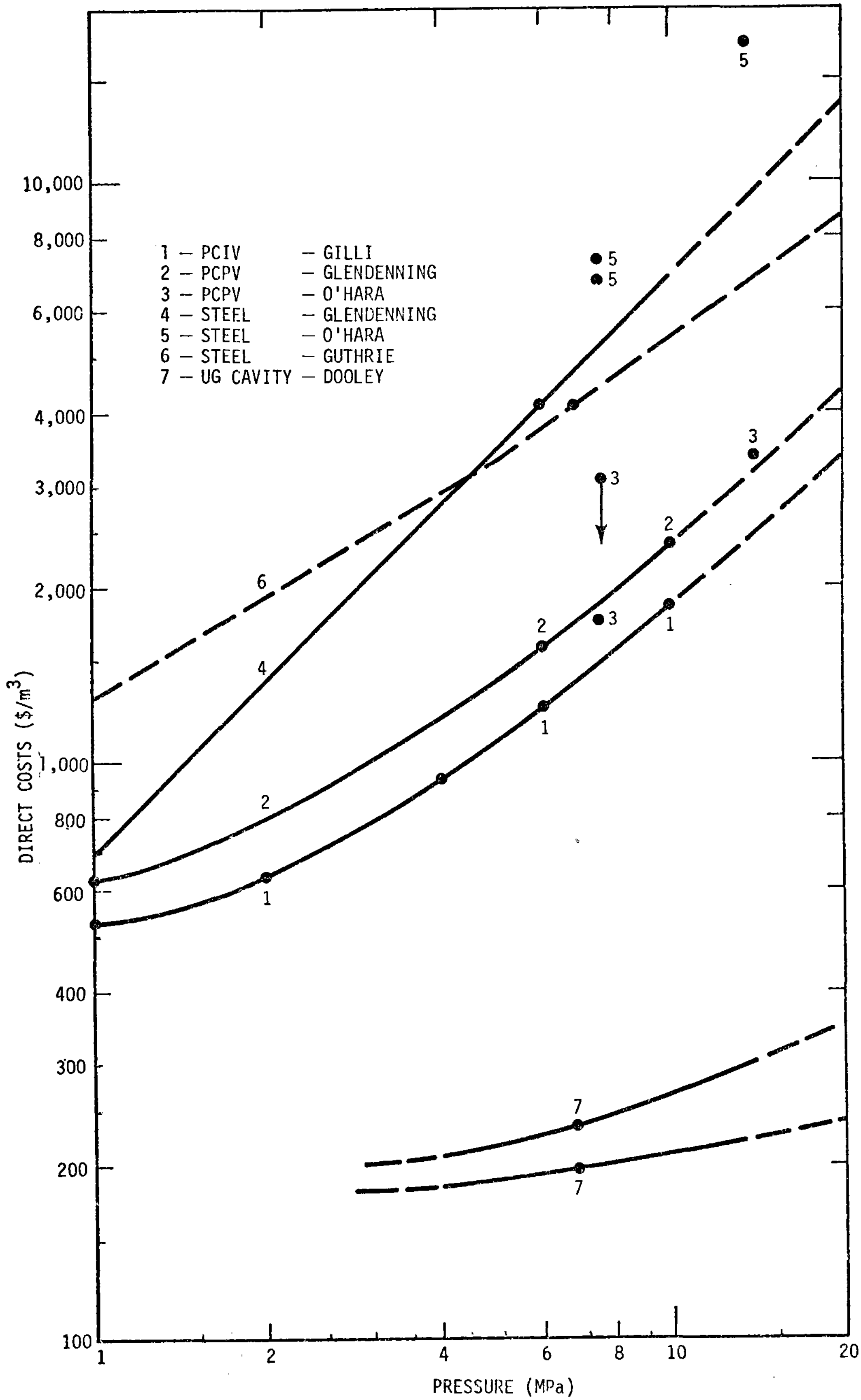


Figure 6-1. Comparison of the specific costs of pressure vessels for HTW containment.

rock-bed in PCPV for thermal storage (References 152, 153), a graph of $\$/m^3$ versus pressure for the PCPV alone. This is approximated by: $\$/m^3 = 1600\ (0.264 + 0.1224\ P)$ for the only size shown, 28,800 m^3. Again multiple modules would be required for the duty described above.

It was found that the Ralph M. Parsons Inc. were performing a study for the Department of Energy (Fossil Fuels) on the cost of PCPV containment of several coal gasifier process modules. The assistance of Messrs. James O'Hara and Richard Howell of that project was solicited. They separated the cost of the PCPV containment and liner from the process machinery internal to and external to the pressure vessel in their process studies.

The three modules conceptually designed had pressure, temperature and volume requirements as follows:

A. Absorber - 1620 m^3 - 7.5 MPa - 66°C

B. Dissolver/Separator - 4400 m^3 - 13.8 MPa - 455°C

C. Gasifier - 1860 m^3 - 7.5 MPa - 1650°C

The cost figures derived by R.M. Parsons were base costs, in December 1977 dollars. To reduce these to direct costs in 1976 dollars a factor of 1.4 was used. The three cases are represented on Figure 6-1 as points labeled 3. Two at the same pressure 7.5 MPa are above and below the Glendenning values. The upper one representing C above has excessively high temperatures; a significant part of the cost was the cooling system: both refractory bricks inside the steel liner, a thick layer of high temperature concrete, and an elaborate cooling system of finned tubes within the high temperature concrete. The arrow indicates it should be moved downward for comparability. Similarly the lower point at that pressure representing A is at a low temperature and should probably be raised for comparability. Both A and C are smaller in volume than the 28,800 m^3 for the curve 2 so might well be higher in specific cost. The higher pressure point for case B similarly falls a little above curve 2.

STEEL. Pressurized vessels of welded steel conforming to ASME Boiler and Pressure Vessel Codes are necessarily limited in volume if wall thicknesses are not to be excessive. Both Glendenning and O'Hara of R.M. Parsons Inc. derived costs for both the PCPV and steel vessels of comparable volume and pressure/temperature rating. Glendenning's result is a straight line, indicating that it comprises multiple small modules optimum for the pressure rating. R.M. Parsons found it necessary to use two to nine steel vessels to match the capacity of PCPV cases A, B, and C. Curve 4 represents Glendenning's curve, points 5 represent O'Hara's results. The latter are considerably higher than the former.

For a single steel vessel of a given size, the variation in cost with pressure is given by Guthrie (Reference 216, *Process Plant Estimating and Control*) as $P^{0.6}$, shown as curve 6.

UNDERGROUND CAVITIES. Two proponents emphasized underground cavity containment of HTW: James Dooley of R&D Associates (References 28,181) and Allen Barnstaple of Ontario Hydro (References 2,3). Their estimates for the cost of excavating underground cavities and preparing them for use as storage were reasonably comparable. James Dooley (Reference 28, *Feasibility Study of Underground Storage Using High Pressure HTW*) listed cost items in a more convenient way to derive costs comparable to the other forms of containment so was used to derive curve 7.

There are significant costs both for the cavity itself and for the shaft(s) from the surface that are needed to access the cavity, remove the muck during construction, and to carry steam pipes and other services from cavity to surface during operation. Dooley chose to consider the cavity as an energy-related cost and the shaft as a power-related cost since its principal role during operation is to carry the steam flow to and from the cavity or cavities.

The cost elements as a function of cavity size are given, summed both to direct costs and to total costs. The direct costs will be used to be comparable with the other costs on Figure 6-1. No indication of sensitivity of completed cavity cost to pressure is given. It is assumed that the cavity depth is proportional to pressure so that the rock overburden pressure will be compatible with the storage pressure. Probably the costs of excavation, rock preparation, lining, and injecting high strength concrete between rock and liner will not be very sensitive to the pressure or depth. However, the shaft costs contain cost components that are proportional to depth and hence more related to energy storage pressure than they are power-related. An approximation of the pressure sensitive component of shaft costs was transferred to the cavity costs as a better estimate of the division between energy-related and power-related costs.

For the smallest cavity described, 29,000 m^3, the direct costs of the cavity, 5.03 M$, gives a specific cost of 172 $/$m^3$. For larger cavities this varies roughly as $V^{-0.22}$. Shaft direct costs are estimated as 15.27 M$ and 20.98 M$ for depths of 360 and 720 m (for storage pressures of 6.9 and 13.8 MPa). The depth and pressure proportional components of these are principally shaft excavation and muck disposal, shaft preparation and lining, and steam piping. These total roughly 5 M$ out of 15.27 and 10 M$ out of 20.98 M$. The remainders, 10.27 and 10.98 M$, are roughly independent of depth and pressure. These values are for a shaft designed for 500 MW power capability or 10.5/ 0.5 = 21 $/kW power-related cost.

A 500 MW_e power capability for 6 hours discharge (3000 MWhrs) requires at 18 kWh/m^3 about 6 cavities of 29,000 m^3. Distributing the pressure dependent part of the shaft cost over the cost of these cavities leads to an energy-related specific cost of (172 + 4P) $/$m^3$. Using similarly the R&D Associates data for a 200 MW shaft, and two cavities, such as might be suitable for 15 percent swing, gives a power-related component of 48 $/kW and an energy-related cost of (172 + 9 P). These energy related costs are shown as the lower and upper curves 7 on Figure 6-1.

It is evident from the exponent of cost versus volume of cavity and from the decrease in shaft costs per kilowatt with increased capacity that underground excavation costs are more susceptible to economies of scale than the other forms of containment for which multiples of reasonably small modules seemed to be required. Since the

UG Cavity costs are considerably less than the other forms, no attempt will be made to justify larger cavity sizes than the one described. The upper curve 7 will be used for small swings and the lower one for large swings.

AQUIFERS. Since aquifer storage requires no excavation of cavities, construction of liners or other volume-dependent expenditures, it comes close to having zero energy-related costs. It relies upon natural formations confined at top and bottom to isolate it from other aquifers. These may extend for thousands of meters with heights of 10 to 100 meters, so extremely large quantities of energy can be stored for long times, making seasonal storage feasible. The only costs that can be considered energy-related are the operating costs, including thermal losses in the aquifer and pumping energy costs, and maintenance costs such as heat exchanger cleaning, well treatment to reduce plugging, etc.

There are, however, power-related costs for aquifer storage. Charles Meyer (Reference 108, *The Role of the Heat Storage Well in Future U.S. Energy Systems*) uses \$150,000 to \$450,000 per installed doublet well including pumps for a 20 MW thermal capability of heat injection and withdrawal. Using \$400,000 gives 20 \$/kW direct costs. The heat exchanger (Figure 3-7) will cost an additional 20 \$/kW, totaling 40 \$/kW. The above assumes a storage temperature of 175-200°C and a return, or supplementary storage, temperature of 70°C.

Selected Case for Sensitivity Analysis

The first selection, the use of PCIV storage with steam generation for a peaking turbine, can be used as the exemplar for the methodology used in determining the cost elements of a TESS concept and combining them. The format will not differ significantly, although the numbers will, when other forms of containment are used, or various storage and throttle pressure levels are used with various accumulator modes of operation.

VARIABLE PRESSURE ACCUMULATOR. Section 5 treated as a base case, pages 5-20 to 5-25, plant #1 with charge steam pressure of 4.86 MPa

(HP turbine outlet), storage pressure of 4.65 MPa, peaking turbine throttle pressure of 2.24 MPa (325 psia), p = 0.50 (50 percent swing), 6 hours discharge and 8 hours charge time. The critical parameters for costing for these base case conditions are a specific output e_o of 15.0 kWh/m^3 and a turnaround efficiency of 0.880 (Figure 5-12).

For the storage pressure 4.65 MPa, the specific cost of PCIV capacity is 1041 \$/m^3, from Figure 6-1 or the text equation. The TOTAL cost (using this specific cost, 6 hours storage, a specific output of 15.0 kWh/m^3 and 2.16 conversion from direct to TOTAL cost) is
$(1041 \cdot 6 \cdot 2.16)/15.0 = 900$ \$/kW.

The turnaround efficiency determines the loss component L as
$32.0 \cdot (\frac{1}{0.88} - 1) \times 6 = 26$ \$/kW (Equation 5-23).

The specific cost of the peaking turbine is determined by the throttle pressure from Figure 5-25 as 400 \$/kW.

The sum of the energy-related costs is 926 \$/kW; since this is for six-hour discharge, it corresponds to 154 \$/kWhr. The sum of the power-related costs is 400 \$/kW. The specific cost of the whole TES system is 1326 \$/kW.

The data and results for this base case are shown in the first column of Table 6-1, as are the results for a number of other cases to be discussed below.

<u>Lower Throttle Pressure</u>. The next two cases keep the storage pressure constant and reduce the throttle pressure at the inlet to the peaking turbine. More of the stored HTW is flashed to steam since $(P_{STOR} - P_{THROTTLE})$ is larger, so the specific output increases. At the same time, the decrease in turnaround efficiency increases L, and the design of the peaking turbine for lower pressure inlet steam increases C_{pp}. Over the range explored, 1.72 and 1.03 MPa, the net effect is favorable, reducing the total specific cost to 1123 \$/kW. Still lower throttle pressure should be explored. However, there is clearly a limit; at a throttle pressure equal to condenser pressure, output is zero. The high specific volume of steam at pressures below 1 MPa requires very large pipes and expensive turbine technology.

Table 6-1. Summary of TESS costs: plant #1 — HTW systems.

Mode	Variable Pressure Accumulator					Expansion			Displacement			FWS
P_{STOR} — MPa	4.65	4.65	4.65	1.03	2.41	4.65	4.65	4.65	4.65	4.65	4.65	5.0
P_{THR} — MPa	2.24	1.72	1.03	0.52	1.20	2.24	2.24	2.24	2.24	2.24	2.24	-
—						-	1.21	1.21	-	1.21	1.21	-
—						-	-	0.16	-	-	0.16	-
η — TA Effic.	0.88	0.836	0.768	0.940	0.775	0.818	0.783	0.600	0.830	0.794	0.610	0.880
e_0 — kWh/m^3	15.0	18.2	22.5	6.6	10.24	11.33	18.9	28.3	13.9	21.3	30.6	40.0
p — Swing	0.50	0.50	0.50	0.15	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.15
$/kW												
PCIV — Tank	900	742	600	870	884	1194	714	477	971	634	441	340
— Supp.	-	-	-	-	-	641	261	69	107	93	55	49
— L	26	38	58	12	56	43	53	128	39	50	123	26
Energy Related	926	780	658	882	940	1878	1028	674	1117	776	619	415
Evap.	-	-	-	-	-	10	10	20	10	10	20	-
Turb.	400	420	465	535	446	400	422	536	400	422	468	359
FWH	-	-	-	-	-	-	-	-	-	-	-	136
Power Related	400	420	465	535	446	410	432	556	410	432	488	495
Total $/kW	1326	1200	1123	1417	1386	2288	1460	1230	1527	1208	1107	910
PCPV												
Tank		950						699			635	498
New Total		1407						1383			1246	1019
Steel												
Tank		2300						1479			1367	1054
New Total		2758						2231			2033	1624
UG Cavity												
Tank		134								114	80	82
New C_e		172								257	257	108
New C_p		477								489	545	667
New Total		649								746	802	775

Lower Storage Pressure. Several cases of reduced storage pressure were explored, namely 2.41 MPa and 1.03 MPa. Since the cost of PCIV containment goes down with reduced pressure (from Figure 6-1), the specific cost of the PCIV is 699 and 488 $\$/m^3$, compared to 1041 $\$/m^3$ for the base case. In each of these cases the pressure ratio of storage to throttle pressure was kept at 2:1. The specific output decreases as rapidly as the specific cost of the PCIV decreases so there is a negligible gain in C_{ES} from storage pressure reduction. Inspection of the five cases leads to an empirical relationship:

$$e_o \approx K \frac{(P_{STOR} - P_{THR})}{\sqrt{P_{STOR}}}$$

This would indicate that higher storage pressures and a low throttle pressure would lead to larger values of e_o, and should be explored. It does not assure that the loss term or turbine cost changes would not overbalance any improvement.

For a constant source of charge steam, turnaround efficiency decreases and L increases as throttle pressure decreases. For a low throttle pressure such as 0.52 MPa, a turnaround efficiency of 60-65 percent could be expected with 4.65 MPa charge steam (Figure 5-14). If the charge steam comes from the crossover (LP turbine inlet) at 1.03 MPa, the turnaround efficiency is much higher since the steam has been passed through the IP turbine generating work down to 1.03 MPa instead of being throttled to that pressure. This case, also from Figure 5-14, is shown as the fourth case in Table 6-1. The low specific output and the high Turbine Island cost outweigh the higher turnaround efficiency and lower storage pressure.

Summary. Of the Variable Pressure Accumulator cases explored, the third column gives the most favorable results with energy-related costs of 658 $/kW, power-related costs of 465 $/kW, and TOTAL specific cost of 1123 $/kW.

EXPANSION ACCUMULATOR. The procedure for Expansion Accumulators is similar, except that external evaporators are used for steam generation and, when there are multiple evaporators in cascade, steam at two or three throttle pressures is fed into separate turbines. Almost all of the HTW is removed from the expansion accumulator; that which is not flashed to steam must be stored in a separate tank at the drain pressure and temperature. For a single evaporator, as shown in Table 5-7, the supplementary tank or drain storage volume is 83 percent of the storage tank volume, and must stand a pressure of 2.24 MPa. Also the specific output is lower than for the variable pressure accumulator at the same throttle pressure. As a result, the costs of storage tank, supplementary tank, and loss term gives an energy-related cost of 1878 $/kW.

Evaporators are very small in volume compared to storage volumes, are very simple and low in cost. Including the valves and piping associated, the cost is estimated at 10 $/kW within a factor of 2. Since the third evaporator at very low pressure will be larger the specific cost is arbitrarily doubled. Neither of these values play a significant role in screening. The resultant total specific cost for a single evaporator is 2288 $/kW.

Multiple Evaporators. The specific output is markedly improved by multiple evaporators and a lower steam pressure at the final evaporator. Both the size and the pressure of the supplementary tank required for drain storage are reduced, leading to further reductions in cost. However, with multiple steam supplies generated, a turbine for each throttle pressure must be costed. The share of the output power produced by each turbine is in proportion to the increment in specific output: 11.33, (18.9 - 11.33), (28.3 - 18.9). Combining the specific turbine costs, C_{pp}, for two and three evaporators in these proportions gives the values shown in Figure 6-1.

Despite the higher L and C_{pp}, the three-evaporator case costs less than the two-evaporator case. However, the use of very low pressure steam at 0.16 MPa (23.5 psia) for a fairly large power capacity (over 130 MW_e) may pose very difficult turbine design problems.

DISPLACEMENT ACCUMULATOR. As with the expansion accumulator, evaporators are required, and some supplementary storage. However, in the thermocline mode, the bulk of the HTW is always in the main pressure vessel, either as hot or cold water. Only enough supplementary tankage is needed to account for the expansion of the water when heated. The specific output of the first evaporator is about 20 percent higher than the corresponding expansion accumulator case. The significant decrease in the cost of both storage and supplementary tanks reduces C_{ES} to 1177 \$/kW, with a corresponding reduction in the total \$/kW since L and C_{pp} are comparable.

Multiple Evaporators. The improved specific output of the first evaporator improves the combined specific outputs for the two- and three-evaporator cases reducing the energy-related costs below the expansion accumulator counterpart. Since the highest pressure turbine produces a larger share of the total power produced, the turbine cost, C_{pp}, is also less, and for three-evaporator case the total, 1107 \$/kW, is closely comparable to the best value found with the variable pressure accumulator, ie 1123 \$/kW.

FEEDWATER STORAGE. Feedwater storage, or manipulation of the relative mass flow in the feedwater heat train during the charge and discharge cycle, has inherently a high specific output ie 40 kWh/m^3. As also is shown in the last column of Table 6-1, 340 \$/kW for the PCIV tank is the lowest of all the cases.

A displacement accumulator, or a two-tank system can be used for feedwater storage; a two-tank system is assumed in Table 6-1, so the cost of the supplementary tank reflects the large volume for cold water that must be stored between discharge and charge. For the displacement mode, this cost item would be reduced by a factor of about four. The high turnaround efficiency gives a low value for L. As noted earlier, feedwater storage cannot be used at 50 percent swing; 15 percent swing is assumed in this case, which also makes the turnaround efficiency higher.

To estimate the cost of increased capacity in the main turbine, allowance must be made for the increased requirement of feedwater heaters for increased steam extraction during the charge cycle, and the fact that the added turbine capacity during discharge does not require feedwater heaters. A cost item for a major addition of feedwater heaters to the main turbine complement is in part balanced by a deletion of the feedwater heater cost from the added turbine cost relationship given by Figure 5-25.

For the 15 percent swing, it is found that the mass flow of steam at the IP turbine inlet and outlet increase by 6 and 20 percent respectively during discharge. The mass flow at inlet and outlet of the LP turbine increase 20 and 31 percent. In the normal mode of operation the rated output of the IP turbine is about 160 MW_e and of the LP turbine is about 320 MW. The average increase in steam flow through the IP turbine stages, about 13 percent, corresponds to added capacity of 20 MW_e.

A separate peaking turbine with 20 MW in the IP turbine and 40 MW in the LP stages would be in the same proportion as the main turbine. The added turbine capacity is estimated to be equivalent to a 60 MW_e turbine at IP inlet pressure (4.86 MPa) and the balance of (120 - 60) = 60 MW_e at LP turbine inlet pressure (1.2 MPa). Combining 354 \$/kW for the former and 455 \$/kW for the latter in equal shares gives 405 \$/kW. From Table 5-15, the feedwater heaters are (10.8/98.5) or 11 percent of the Turbine Island cost. Deleting 11 percent from 405 gives 359 \$/kW as shown in Table 6-1.

During charge the feedwater flow is increased by 70 percent. Added feedwater heaters increase the system cost by 10.8 M\$ $\cdot$ 0.70 $\cdot$ 2.16 = 16.3 M\$; allocating this to the 120,000 kW of peaking capacity gives 136 \$/kW as an added power-related cost unique to feedwater storage systems.

The total \$/kW for this case is 910 \$/kW: 415 energy-related and 495 power-related. It is lower than any of the other cases explored.

PLANT #2 CASES. A similar set of analyses were made for the 1140 MW nuclear plant, and results are shown in Table 6-2. The methods already described were used; there were no surprises.

Table 6-2. Summary of TESS costs: plant #2 – HTW systems

Mode	Variable Pressure Accumulator		Expansion		FWS
P_{STOR} – MPa	6.21	6.21	6.21	6.21	3.70
P_{THR} – MPa	3.10	2.59	3.10	3.10	-
–			-	1.21	-
η – TA Eff.	0.90	0.87	0.834	0.766	0.88
e_0 – kWh/m^3	15.4	17.9	10.9	21.3	30.0
p – Swing	0.50	0.50	0.50	0.50	0.15
$/kW					
PCIV – Tank	1078	927	1522	779	388
– Supp.	-	-	794	213	50
– L	19	26	34	52	23
Energy Related	1097	953	2350	1044	461
Evap.	-	-	10	10	-
Turb.	394	412	394	435	375
FWH	-	-	-	-	87
Power Related	394	412	404	445	462
TOTAL $/kW	1491	1365	2754	1489	923
UG Cavity					
Tank		141		118	100
New C_e		167		209	167
New C_p		478		511	633
New Total		645		720	800

A higher IP inlet steam pressure is available in plant #2 than in plant #1. This makes a higher storage pressure feasible (6.21 MPa). This increases both the specific $/m^3 cost of PCIV and the specific output e_0. For the throttle pressures considered, the net result is a C_{ES} that is higher than the corresponding cases for plant #1. The insight gained from the plant #1 cases would indicate that a lower throttle pressure, say 1.16 MPa – the LP turbine inlet pressure, would have a specific output of 25 kWh/m^3. This would reduce C_{ES} to 664 $/kW, probably increase L to 60 $/kW, and C_{pp} to 490 $/kW giving a total $/kW of 1214 $/kW; a drop comparable to that found with plant #1.

The feedwater storage case is the least costly, as with plant #1. The specific output is 30 rather than the 40 kWh/m^3 found for plant #1, due mostly to the smaller temperature differential from hot to cold feedwater.

Selection #1 – PCIV

The PCIV form of containment has been examined for a variety of storage configurations and input and output conditions. In all cases the specific costs of incremental capacity added by the TESS is higher in $/kW than the 743 $/kW and 785 $/kW of the reference plants #1 and #2 when used in a load-following mode.

The lowest equivalent capital cost 910 $/kW (including the term L as the capital cost equivalent of turnaround efficiency losses), is in the feedwater storage mode in plant #1 (Table 6-1). Plant #2 in the same mode was very close, 923 $/kW. These have the disadvantages of a limited swing, ie 15 percent. For larger swing capabilties, a variable accumulator mode with low throttle pressure, and a three-evaporator displacement mode proved lowest in cost.

The proponents of Selection #1 had suggested a combination of an expansion mode and a single evaporator with feedwater storage as a concept, selecting the drain temperature from the evaporator to be at boiler inlet temperature. This was not explored. It could produce some improvement for small swings.

Further judgments on Selection #1 must await comparison with the other selections.

Selection #2 – PCPV

Prestressed Concrete Pressure Vessels are closely analogous to PCIV containment. The data in Figure 6-1 and accompanying text suggest that PCPV may be higher in cost than PCIV by a factor of 1.28. The impact of this on a subset of the PCIV cases analyzed is shown in Table 6-1. The resulting higher value of the main tank cost is shown for the selected cases. This is followed by the new total cost in $/kW which includes the incremental costs on both the main and supplementary tanks.

For the assumed costs, there is no economic basis for selecting PCPV over PCIV. However, there is an uncertainty level in the costs of each form of containment. It is considered unlikely that the PCIV would cost less than indicated; it could easily cost 20-40 percent more than indicated, since there is little background of experience. The PCPV could also cost more, particularly with special requirements imposed, such as the thermal cycling of a thermocline in the displacement mode. There is some possibility of overlap in the costs of these two containment forms.

Selection #3 – Steel Vessels

Steel Vessels were found to be considerably more expensive than PCIV and PCPV as illustrated in Figure 6-1. A factor of 3.1 over PCIV is a fair estimate over the pressure range for the cost ratio $\$/m^3$. The resulting Total \$/kW for cases shown in Table 6-1 are clearly non-competitive with other forms of containment or with alternative storage systems.

The high cost of steel vessels is in part due to code requirements. A history of catastrophic accidents has caused agreement on a set of specifications on the quality and properties of steel used, the techniques of welding, methods of inspection and testing, and factors-of-safety in design that minimize to "acceptable" levels the risk of failure. The specific concept described as Selection #3 tried to reduce the cost of steel containment by using a lower cost, more available grade of steel and an efficient low cost method of welding. No cost and risk estimates were available from the proponents. It is estimated that by the time the design was modified and codes were modified to make this concept acceptable it would be at least as costly as the current technology steel vessels described by Figure 6-1.

Selection #4 – UG Cavity - Concrete Stress Support

The discussion earlier in this section developed the rationale for dividing the costs of HTW storage in an underground cavity into an energy-related and a power-related component. Using the lower curve 7 from Figure 6-1 for the 50 percent swing cases gives at 4.65 MPa a specific cost of 190 $\$/m^3$. A factor of 0.18 adequately relates this to the cost of PCIV, although the pressure dependence of the two are not quite parallel.

The comparison with PCIV in the second column of Table 6-1 shows an 82 percent reduction to 134 \$/kW for containment. The power-related costs include, in addition to the same turbine cost as for the PCIV, the power-related cost for the shaft: 21 \$/kW · 2.16 · 500/400 = 57 \$/kW. This modifies the directs costs of 21 \$/kW for a 500 MW shaft by ratio of 500 MW/400 MW and converts to TOTAL costs.

The proponent of Selection #4, J. Dooley of R&D Associates, favored the variable pressure accumulator mode. For a low specific cost of containment, the tradeoffs effects of storage and throttle pressure on specific output, turbine cost, and turnaround efficiency are opposite from those found for PCIV. Going to a lower pressure (third column of Table 6-1) will reduce C_{ES}, but the increase in C_{pp} and L will exceed this reduction. Going to a higher throttle pressure (column 1) again has compensating effects leading to a slight improvement in total costs.

<u>Selection #5 — UG Cavity - Air Supported</u>

This selection differs from Selection #4 in several respects. A steel liner in the UG Cavity is separated from the rock by a layer of compressed air for stress transfer of the HTW pressure to the rock. A displacement mode accumulator is used, with a moving thermocline separating hot and cold water during charge and discharge. A feedwater storage TESS configuration is proposed as the source and utilization of the stored energy.

The sizing of components for the feedwater storage application is done in the last column of Table 6-1. The sum of the main tank and supplementary tank for PCIV is multiplied by 0.21 rather than 0.18 because the upper curve 7 of Figure 6-1 applies for the limited 15 percent swing. The result 82 \$/kW is taken as the cost of the displacement accumulator in an air supported cavern. The power-related cost for the shaft is 48 \$/kW $\cdot \frac{200}{120} \cdot$ 2.16 = 172 \$/kW, corresponding to an adjustment from direct to TOTAL costs and the small swing of 120 MW. Combining the cost elements gives a total cost of 775 $/kW. For plant #2 the total cost is 800 $/kW.

Selection #6 – UG Cavity - Evaporators

This selection closely parallels the last one except that the method of utilization proposed is multiple evaporators to generate steam. The costs for both two- and three-evaporator cases are developed in Table 6-1 by multiplying the PCIV containment costs by 0.18 and adding the power-related shaft costs of 57 $/kW as was done for Selection #4, because these cases are also for a 50 percent swing. The displacement mode is assumed. The evaporators and supplementary tanks are assumed to be above ground. They each operate at or contain HTW at a different pressure than the main storage tank, so that a different pressure of compressed air would be required to support them if they were underground. The two-evaporator case is preferable to the three-evaporator case.

A displacement accumulator with thermocline poses potential problems of fatigue and failure from cyclic thermal stresses. A thin steel shell, compressed air supported, may be easier to design for these conditions than the concrete stress support of Selections #2 and #4. In Selections #5 and #6 there is the additional problem of pumping water to the surface and restraining its flow from surface to cavity in both the charge and discharge cycles. Prevention of low pressure causing steam flashing or overly high pressure endangering the thin shell must be guarded against.

Selection #7 – Aquifer Storage

Aquifer storage is an anomalous form, in that although energy is stored as HTW, energy is also stored in the sand and gravel of the aquifer, and the HTW used in the aquifer is not the boiler quality feedwater so a heat exchanger must be used. There are strong resemblances to rock-and-oil systems, with the groundwater, like oil, being considered as a heat transfer fluid.

Although the use of aquifers at elevated temperatures cannot be considered as near-term available since demonstrations and data are missing, there can be a reasonable degree of confidence that it is feasible to some temperature level, on the basis of long experience in well drilling and use, both for injection and withdrawal.

If it should prove that aquifer storage is feasible to temperatures between 200° and 250°C, without major added O&M costs to maintain the wells and heat exchangers over the system lifetime, the concept can be compared with the feedwater storage concept discussed as Selection #5. The specific output and turnaround efficiency should be comparable. The 100 $/kW cost of storage volume and the 172 $/kW cost of shaft are replaced by the well and heat exchanger costs. At 40 $/kW (thermal) direct costs, the power-related TOTAL cost for the aquifer storage would be $(40/0.24) \cdot 2.16 = 360$ $/kW (electric). Here the 0.24 is an estimate of the conversion efficiency. For plant #1 this would lead to energy-related costs of 75 $/kW, power-related costs of 855 $/kW, and a total $/kW of 930. This is higher than for Selection #5 and comparable to the PCIV.

If aquifer storage is limited to lower temperatures than 200°C, the available maximum swing decreases and the cost of conversion increases.

LVP SELECTIONS

Low Vapor Pressure (LVP) systems, also called sensible heat systems, atmospheric pressure, or one-bar systems, is the second major class of selections considered. The names above describe related characteristics of the systems; a liquid is used for heat transfer and storage that has a low vapor pressure (less than 0.1 MPa) at the temperatures of interest for storage, so that containment may be at atmospheric pressure (ie, one bar). This results in low cost containment compared to those discussed for HTW containment. The system data used as an example in the modeling description in Section 5 (page 5-52) gives a direct cost of $295,700 for a tank of 12190 m^3, or a specific cost of 24.3 $/$m^3$.

To use such low cost containment, storage media of higher cost than HTW must be used, and heat-exchanger trains must be used to keep the HTW and steam separate from the storage media yet transfer heat to and from storage. The costs of these items must be compared to the reduced containment cost in order to evaluate Selections #8 through #12.

Selected Case for Sensitivity Analysis

In order to describe adequately the modeling of heat exchangers and the conversions of charge steam to storage to generated steam for peaking, a set of selected system characteristics were described in Table 5-12. To the parameters there considered, the specific cost of the peaking Turbine Island, and a loss term L, to include turnaround efficiency in specific capital cost comparisons, can now be added.

The selected system characteristics were for plant #1, with charge steam at IP turbine conditions 4.86 MPa (705 psia), 306°C (584°F), 44°C superheat (90°F). The baseline case generated discharge steam at 2.01 MPa (292 psia), 251°C (484°F). Storage was in granite rock-beds with the voids filled with the heat transfer fluid, Exxon Caloria HT-43 or its equivalent. It was assumed that the volume of the storage media was 25 percent oil and 75 percent rock.

The discharge steam conditions are determined by the heat exchanger parameters assumed. For the baseline case, giving the discharge steam conditions stated above, the approach, α, is 5.6° C (10°F) and the mass flow ratio, M_c, of oil to charge steam is 15.0. These parameters, along with the properties of the oil, such as specific heat, density, and viscosity as a function of temperature, dominate the design of the heat exchanger. The properties of HTW and steam also contribute to the heat transfer coefficient, determining the area of heat exchange systems required for each part of the heat exchanger train as shown in Table 5-13. The requirements there given are for a 50 percent swing.

The direct cost of these heat exchangers is 30.6 M$. Converting to specific TOTAL costs for 50 percent swing gives 165 $/kW. This is one of the power-related components of storage cost, shown in Table 6-3 in a format similar to Tables 6-1 and 6-2.

Table 6-3. Summary of TESS costs: plant #1 — LVP systems.

Fluid	Oil	Oil	Oil	Oil	Salt	Salt	Salt
Fraction	0.25	0.25	1.00	0	1.00	0.25	0
Rock							
Fraction	0.75	0.75	0	1.00	0	0.75	1.00
α °C	5.6	8.4	5.6	5.6	5.6	5.6	5.6
M_C	15.0	12.5	10.0	10.0	20.0	20.0	20.0
P_{THR} MPa	2.28	1.47	1.24	1.24	1.81	1.81	1.81
η	0.831	0.781	0.759	0.759	0.755	0.774	0.794
$/kW							
C_{TM}	154	134	281	68	1138	370	75
L	39	55	61	61	62	56	50
Energy Related	193	188	342	129	1200	426	125
C_{HX}	165	123	125	125	85	85	85
C_{pp}	400	418	435	435	416	416	416
Power Related	565	541	560	560	501	501	501
Total $/kW	758	729	902	689	1701	927	626

The peaking Turbine Island cost is found as before from Figure 5-25 to be 400 $/kW for 2.01 MPa throttle pressure. The turnaround efficiency determines L as 39 $/kW.

The costs of the tankage and storage media, C_{TM}, are dependent on the media used, their configuration, and the assumed costs of the media. As indicated above, the selected baseline system uses Caloria HT-43 and rock in packed beds. The assumed 1976 cost of Caloria is 246 $/Mg (223 $/ton; 80¢/gal). The assumed cost of rock as river bed gravel is 16.5 $/Mg (15 $/ton).

For the baseline case the oil required is 57,500 m^3 (54,750 tons; $15.2 \cdot 10^6$ gal). The rock required is 560,000 Mg (615,700 tons). The tankage required is 289,000 m^3 ($10.2 \cdot 10^6 ft^3$). (See Table 5-12.) The cost of these may be totaled: 12.21 M$ for oil, 9.24 M$ for rock, 7.02 M$ for the 16 tanks, totaling 28.47 M$ direct costs. TOTAL costs in $/kW = $28.47 \cdot 10^6 \cdot 2.16/400{,}000 = 154$ $/kW.

The total TESS cost for this case, given in the first column of Table 6-3 is 758 $/kW.

SENSITIVITY TO α AND M_C. The various elements of cost are affected differently by changes in the design value of approach at heat exchanger "pinch-points," and in the ratio M_c of the flow of heat transfer fluid to the flow of charge steam. In general, a decrease in the value of α will increase the cost of the heat exchanger; improve turnaround efficiency, hence reduce L; raise the discharge steam pressure and temperature, hence reduce C_{pp}. A decrease in the value of M_c will decrease the amount of oil, rock, and tankage required, hence C_{TM}; decrease the cost of the heat exchanger because of lower flow rates and a larger ΔT; decrease the turnaround efficiency, hence increase L; and decrease the pressure and temperature of discharge steam, hence increase C_{pp}.

The counteracting trends do not clearly show in a tabular display of many cases because of the non-linear variation of each cost component with the two parameters. Figure 6-2 is a map of the total energy-related costs versus the total power-related costs for three values of M_c; 10, 12.5, and 15. Along each of the curves, the other parameter α is varied in increments of 2.8°C (5°F) from 2.8 to 11.2°C. The circled point is the selected case described in column 1 of Table 6-3. The diagonal dashed lines represent constant total cost of 750 $/kW and 725 $/kW.

A rough optimum, within the range explored, is the point representing $M_c = 12.5$, $\alpha = 8.4$ The values for this case are given in column 2 of Table 6-3. The improvement, from 758 to 729 $/kW is only 4 percent.

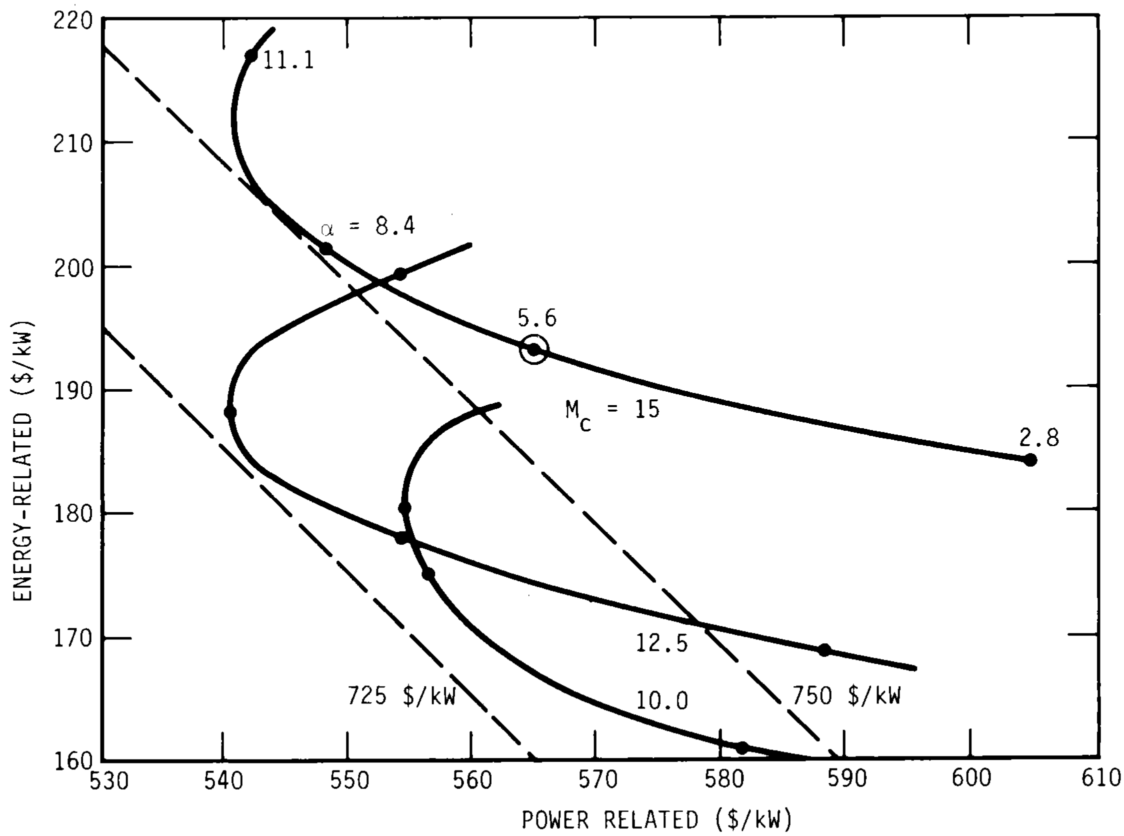


Figure 6-2. Map of effects of parameters M_C and α on energy-related and power-related costs.

SENSITIVITY TO MEDIA COST. The total cost of TESS is also sensitive to the properties of the storage media, including their specific cost in $/kg or $/m^3$. For the selected case, the share of the cost item C_{TM} is 0.429 oil, 0.324 rock, and 0.247 tankage. Use of a more expensive oil such as Therminol, at 10 $/gal versus 0.80 $/gal would increase C_{TM} by a factor of 4.9. Rock costs in most of the United States can be as low as 3 to 6 $/ton for crushed granite or similar rock, washed and screened to a size class, eg 1.9 to 2.5 cm (3/4 to 1 in.). The more rounded river bed gravel can cost 13 to 15 $/ton; $15 was used in the selected case. Special solid materials such as taconite pellets, alumina, or magnesia spheres can be considered more costly; taconite has been estimated at 40 $/Mg (36 $/ton).

If lower cost rock can eventually be used, ie, is found to be compatible with oil over the temperature range of the selected case, for long periods of time with low makeup and maintenance costs, the value of C_{TM} could be decreased. For rock at $5 rather than $15/ton the value of C_{TM} would be decreased by a factor of 0.784.

It should be noted that the changes in C_{TM} in $/kW give only a first estimate of the changes in the total $/kW for TES. First, the analysis must be corrected for the different density and specific heat of the altered fluid or solid storage medium. Second, any major changes in one component would require design changes in the other components to reoptimize the system. For example, with a very costly oil, a lower value of heat exchanger approach and a smaller value of M_c would decrease the amount of oil required but increase the cost of the heat exchanger, the turbine, and the loss term. A stepwise increase in the cost of other components to decrease the cost of oil by a larger amount could be continued until a new optimum was found.

SENSITIVITY TO PACKING VOLUME FRACTION. Deviations from the assumed ratio of a packing volume fraction of 75 percent for rock and 25 percent for oil can be considered. At one extreme the rock packing fraction can go to zero, ie only oil is used. At the other extreme are "drained bed" concepts (Concept Definitions #27 and #35) in which the voids between pebbles are normally filled with inert gas, and the oil is only used as a heat transfer fluid during charge and discharge. Much less oil is required for these concepts; as a limit, the cost of the TESS with 100 percent of the thermal storage in rock can be considered.

Using oil alone, without a packed bed, has been proposed, both with a thermocline, and with separate hot and cold tanks, eg Selections #8 and #10. The relative properties of rock and oil are shown in Table 6-4. They are compared in cost per unit weight ($/Mg), weight per unit volume (kg/m^3), and in specific heat (kJ/kg · °C). The last column in each group is a ratio R showing how oil compares to rock which is taken as 1.0. Oil costs 14.9 times as much per unit weight, is 0.267 times as dense and has 2.66 times the specific heat. The specific

Table 6-4. Summary of media parameters — LVP systems.

	Cost			Density			Specific Heat		
	\$/Mg	(¢/lb)	$R_\$$	kg/m^3	(lb/f^3)	R_ρ	kJ/kg·°c	(BTU/lb·°F)	R_{cp}
Oil (Caloria)	245.	11.16	14.9	711	44.3	.267	2.616	.626	2.66
Rock (RB gravel)	16.5	0.75	1.0	2663	166	1.0	0.982	.235	1.0

heat indicates the relative energy stored per unit weight for a given range in temperature. R_{cp} indicates oil has the advantage by the rate 2.66. The product of R_{cp} and R_p, 0.710, indicates the relative energy stored per unit volume; oil contains less energy than rock. The ratio $R_{cp}/R_\$$, 0.178, indicates the relative energy stored per dollar; oil is inferior to rock by more than five to one.

For 75:25 volume ratio of rock and oil, the energy stored in each are respectively 80.8 and 19.2 percent. To replace the rock by all oil requires 1/0.192 or 5.21 times as much oil. More volume is required, hence more tankage by the ratio 1.305. From the data on page 6-24 the direct costs of media and tankage, which were 28.47 M\$, become 12.21 · 5.21 + 7.02 · 1.305 or 72.77 M\$, or a TOTAL specific cost of 393 \$/kW for C_{TM}. To lower this cost component, one should move in the direction of lower M_c. An example for $M_c = 10$, $\alpha = 5.6$ is given in the third column of Table 6-3. C_{TM} is reduced by more than 100 \$/kW, C_{HX} is reduced by 40 \$/kW, while the components C_{pp} and L, dependent on turnaround efficiency, increase by only 60 \$/kW over their column one values.

At the other extreme, drained rock-beds in which the oil only functions as a heat transfer fluid, the results for the same M_c and α are shown in the fourth column. The cost of rock alone plus tankage makes C_{TM} equal 68 \$/kW, rather than 281 \$/kW. Note that all the other cost components remain unchanged.

Several intermediate values of C_{TM} for other volume fractions of oil were explored. For 0.10, perhaps a more reasonable approximation to a drained bed to allow for filling the pipes and heat exchangers and wetting the rock with oil, C_{TM} is 84 \$/kW. For 0.25, C_{TM} is 100 \$/kW. The relationship is smooth, although not linear, so other values can be interpolated.

OTHER HEAT TRANSFER FLUIDS. Other materials than Caloria HT-43 can be used as the heat transfer fluid. Many are more expensive but have advantages such as less degradation at high temperatures, better compatibility with low cost rock-beds, or better heat transfer capability. Two such fluids proposed are molten salts, such as HITEC or PARTHERM 290 and molten sulfur. A table of values similar to those in Table 6-4 are given in Table 6-5. As before the ratio columns are normalized to rock as 1.00.

Table 6-5. Summary of media parameters (continued).

	Cost			Density			Specific Heat		
	\$/Mg	(¢/lb)	$R_\$$	kg/m^3	(lb/f^3)	R_ρ	kJ/kg·°c	(BTU/lb °F)	R_{cp}
HITEC	605	27.5	36.7	1909	119	.717	1.558	.373	1.58
Sulfur	75	3.4	4.5	1733	108	.651	1.149	.275	1.17

The relative energy per unit volume, $R_{cp} \cdot R_e$, is 1.13 for HITEC and 0.762 for sulfur, ie exceeds rock for the former and is between that for rock and oil for the latter. Three cases for using HITEC as the heat transfer fluid are included in Table 6-3.

The relative cost per unit energy (for a given ΔT) is 23.2 for HITEC and 2.90 for sulfur. It is clear from Table 6-3 that the economics of an all molten salt system (100 percent volume fraction) is not favorable compared to the other LVP systems and many of the HTW systems. On the other hand when a drained bed is assumed, with very low volume fraction of salt, the comparison is favorable with all except the best underground cavity systems considered.

The drained bed case with salt is less costly than the drained bed case with oil because the heat transfer characteristics of molten salt are better than oil. A fouling factor must be included in considering oil as a heat transfer fluid, since the high molecular weight degradation products tend to coat the heat exchange surfaces; HITEC is sufficiently clean that no fouling factor need be assumed. Comparing literature values and those offered by some proponents indicates that the heat transfor coefficient, U, for HITEC and comparable salts may be as much as an order of magnitude better than for oil. This particularly important for the boiler and condenser heat exchanger, when the liquid-side contribution to U dominates, but has appreciable impact on superheaters and subcoolers as well.

The cost of the 25 percent volume fraction case is higher than that for oil, but reasonably comparable. The three salt cases have not been separately optimized. It is reasonable to expect that exploration of the range of M_C and α would improve all three and bring them closer together, by tradeoffs between C_{TM} and C_{HX}, but the effect is not expected to alter the ranking of cases.

FEEDWATER STORAGE. The feedwater storage mode is proposed in Selection #8, using Caloria HT43, and separate tanks for storage of hot oil and cold oil (with inert gas as ullage in empty tanks). The heat exchanger configuration is different as illustrated both in Figure 3-8 and Figure 5-24. For discharge, an oil to water counter flow heat exchanger is used to heat feedwater from 80°C to 227°C, in the case of plant #2. During the charge cycle the steam extraction from the main turbine is increased at all extraction points to heat oil to a temperature higher than 227°C by the approach α to be used in the discharge heat exchanger design.

As in the feedwater storage case with HTW containment, the added heat exchangers for extraction steam during charge must be included, but during discharge less steam extraction is required so the Feedwater Heater account in the incremental Turbine Island costs may be deleted.

For the case analyzed in Section 5 pages 5-58 to 5-62, ie plant #2 and 17 percent swing,the turnaround efficiency is 0.853. The value of α assumed is 11.1°C; since this α determines the inlet and outlet temperature of the oil, the value of M_C is determined. It is about 2 for the oil to water discharge heat exchanger, and about 12.5 for the steam to oil extraction heaters.

The cases studied in Table 6-3 showed that an all-oil system is considerably more costly than one with 25 percent volume fraction of oil in a packed-bed thermocline system. Use of hot and cold tanks instead of a thermocline would make it still more costly. In order to compare feedwater storage most favorably to steam generation systems, the packed-bed thermocline system will be assumed. The added cost for all-oil can be estimated.

The components of C_{TM} are 2.05 M$ for rock, 2.89 M$ for oil and 1.53 M$ for tankage, for a direct cost of 6.47 M$. Conversion to total cost gives C_{TM} = 80 $/kW. This is considerably lower than for the steam generation cases in Table 6-3. As with HTW feedwater storage cases, feedwater storage gives a high specific output in kWh/m^3. The loss component L is 29 $/kW for the stated turnaround efficiency.

The same method used with HTW feedwater storage is used to find the incremental turbine costs. The added capacity is about 0.23 in the IP turbine and 0.77 in the LP turbines. For normal operation the ratio is 0.38 for IP versus 0.62 for LP. As a set of two peaking turbines each operating from a given inlet pressure down to condenser pressure, the outputs required are again found to be roughly equal. Therefore the Turbine Island cost is found as the average of that for 6.72 MPa, IP inlet conditions, and 1.16 MPa, LP inlet conditions, on Figure 5-26. The average of 354 and 484 is 419 $/kW. However a part of this must be deleted which represents the feedwater heating account. For plant #2, in Table 5-15, this account is 15 M$ out of 190 M$ or .0789. Deleting this fraction from 419 gives C_{pp} = 389 $/kW.

The three heat exchangers have direct costs (from Table 5-14) of 3.31 M$ for the charge phase extraction heater, 0.31 M$ for the trim heater and 19.49 M$ for the discharge phase feedwater heater. The sum of these can be converted to 253 $/kW.

The energy related costs are 109 $/kW. The power related costs are 642 $/kW, giving a total of 751 $/kW. It exceeds the comparable steam generation cases principally in the high cost of the discharge heat exchangers. This case has not been optimized; the use of the same α = 11.1°C on all the heat exchangers is arbitrary. For a liquid-to-liquid heat exchanger, the cost is roughly inversely proportional to α. It can be estimated that using α = 22.2°C (40°F) would halve the cost of the discharge heat exchanger. The trim heater must be doubled, and there will be some increase in the loss term, L. The net result is a reduction of total cost to 670 $/kW.

Selection #8 – Oil Storage of Feedwater Heat

This selection features oil, specifically Caloria HT43, used for feedwater storage, use of separate hot and cold tanks, and several variants in the heat exchanger configuration. The analysis above chose as the basis for comparison with other concepts the use of the oil and packed bed of rock in a 25:75 volume ratio as most likely to be competitive.

The effect of using oil only and the extra tankage can be estimated. Five times as much oil will be needed for the same stored energy, the tankage will increase by more than a factor of two. This would increase the cost C_{TM} from 80 $/kW to 217 $/kW. The 670 $/kW total cost would increase to 807 $/kW.

For simplicity in analysis with some expectation it would be most cost effective, the case analyzed above in this Section assumed extraction steam-to-oil heat exchangers for charging. Figure 3-8 and other variants by the proponents assumed increased extraction steam-to-water (feedwater heater) capacity and used the hot feedwater in the same oil-to-water heat exchanger used for the discharge phase. There are both advantages and disadvantages to this use of an intermediate heat exchange during charging. Advantages include the reduced likelihood of oil leaks into the feedwater system. The intermediate loop pressure can assure that leaks can only be in the reverse direction. Also, the technology of steam to water feedwater heaters is more certain than that of steam to oil. The cost of the latter is certain to be greater than that of steam to water for expected heat transfer coefficients with oil. The principal disadvantage is that the intermediate loop will reduce the turnaround efficiency. It was noted above that optimizing the TESS cost by reducing the cost of the oil to water heat exchanger might require an α of 20°C or more. If this ΔT is encountered twice, on charge and discharge, as well as the ΔT in the feedwater heaters, the turnaround efficiency could be down to 70-75 percent.

For comparison purposes the configuration analyzed above in this section will be retained as the recommended form and its cost assumed as 670 $/kW.

<u>Selection #9 – Oil and Packed Bed Thermocline</u>

The features of this selection are the use of a packed bed of rock or gravel in tanks, with the voids filled with oil. Clearly the thermocline mode must be used when rock is the major storage medium. This selection also generates steam for use in a peaking turbine.

The foregoing sensitivity analysis, and the comparison in Selection #8 of LVP systems that use oil only with those using a packed bed, provide the data for comparative evaluation of this selection. From the cases considered, the use of 25 percent volume fraction of oil, 75 percent of rock, a mass flow ratio M_C of 12.5, and an approach of 8.4°C (15°F) appears roughly optimum, at 729 $/kW (Table 6-3, column 2).

For a 50 percent swing, 400 MW peaking in plant #1, the storage tanks and heat exchangers are best built in multiple units. Tables 5-12 and 5-13 indicated 16 storage tanks and 80 separate heat exchangers. No desuperheater heat exchanger is recommended for this selection; an attemperator (or spray desuperheater) is used instead. Comparing cases analyzed with desuperheater and attemperators indicated that the desuperheater heat exchangers added 20-30 $/kW while the reductions caused in the other cost components only totalled 10-15 $/kW.

The postulated void volume of 25 percent for oil is arbitrary. Close packed uniform spheres in various crystal lattice configurations leave a void volume of about 26 percent. This would not be achieved with random packing of non-uniform spheres of one size grade; the void volume might exceed 40 percent, according to the literature, unless great care is taken in assuring maximum settling layer-by-layer as installed. The opening between three touching spheres will pass a smaller sphere with a diameter less than one-third of their diameter, so particles smaller than this can partially fill the voids between spheres. A close packing of one size sphere, say 3 cm diameter, and void filling by smaller spheres, say 0.3 cm diameter, could approximate a void volume of 7 percent, so that even imperfect packing should attain less than 25 percent voids. The proponents of this selection have experimentally used river gravel plus coarse sand as such a two-size mixture.

Another approach to minimize the ratio of oil to rock needed is the drained-bed concept (CD #27 and #35). The first variant uses trickle-charging. The voids in a packed bed contain an inert gas. To charge a tank, hot oil is distributed over the top layer, trickling down, to cause a thermocline to move downward. The cold oil is removed from the bottom; when charged the tank contains hot rock and inert gas. To discharge, cold oil trickles down from the top, and is removed hot from the bottom. The thermocline again moves downward leaving the discharged tank filled with cold rock.

The second variant was a result of this current project. An alternative way to reduce oil requirements is to fill with oil the voids in only three of the many tanks (eg 16 in the case discussed). A filled tank is charged and discharged conventionally, with hot oil going in and out at the top and cold oil out and in at the bottom. While one tank is being charged or discharged, the oil from an already charged or discharged tank is drained and transferred to the next tank to be processed.

These concepts have not yet been tested sufficiently to be considered near term available, but are growth potential directions to reduce cost.

Therefore, the comparison value for this selection will be assumed as 729 $/kW, as in column 2 of Table 6-3.

Selection #10 - Oil and Salt Storage

The features of this selection are use of dual media, both Caloria HT43 and HITEC, primarily to extend the temperature range of storage to a higher temperature than that for which Caloria was acceptable. The molten salt, HITEC, is operated over the range 300-450°C to desuperheat and superheat steam from a central solar thermal system. The oil, usable to about 310°C is the heat transfer fluid in the condenser/boiler and the subcooler/preheater.

No judgment is made about the merit of this combination for the solar application. In this project the concepts being compared do not require temperatures above 300°C in the storage medium because, for reasons already given, the source of energy for storage is the IP inlet steam in plant #1 and the live steam from the nuclear reactor plant #2. Both are at under 300°C. For these steam conditions the degree of superheat to be removed is much less than in the solar application considered by the proponents of this selection. As has been discussed, there was not found to be an advantage in using a desuperheater rather than an attemperator. As shown in Figure 5-23 a superheater is used but only produces about 40°C of superheat.

As the use of molten salt as the sole medium is treated as Selection #11, Selection #10 will not be treated further.

Selection #11 - All Molten Salt

This variant of Selection #10 uses only molten salt for storage. As shown in Figure 3-11, three tanks are used; there are two large ones for a high mass flow ratio M_c in the boiler/condenser and the preheater/subcooler. A smaller tank is used at the highest temperature 482°C, used for desuperheating/superheating.

As this selection does not propose the use of packed beds with the salt, it most resembles the column in Table 6-3 which is headed Salt:1.00. The specific cost for such a system is 1701 \$/kW. This high cost results solely from the high cost of HITEC compared to oil and to rock. At 605 $\$/M_g$ (27.5 ¢/lb)(1976\$), HITEC costs about 4.3 times as much as oil and 24 times as much as rock, per unit of energy stored. To make this selection economically viable, it must be modified in one of three ways.

The first is to use packed-bed/thermocline tanks with HITEC as the heat transfer fluid. Honeywell, Inc. has been considering this (Reference telecon with R. LeFrois), and DOE (V. Berolla: Sandia Livermore) has been conducting static tests of degradation rates of oil and molten salt at high temperature in the presence of various minerals such as granite and taconite pellets. Some catalytic increase in the degradation rates may occur from rock/fluid interactions, but the data does not yet seem adequate for judgment in the range of interest for this project, ie 200-260°C rather than the higher temperatures contemplated for other applications. Table 6-3 indicates that a low void fraction, by use of multiple size grades in packed-beds or use of drained-bed technology, could give attractive costs between 20 and 5 percent volume fraction of salt.

A second approach is a decrease in the cost of the molten salt mix. HITEC, using quite pure sodium and potassium nitrates and sodium nitrite, has a higher cost than some alternative salt mixes. The eutectic of sodium and potassium nitrate, also known as draw salt, is offered by Park Chemical as *Partherm 430* with a melting point of 220°C and 430°F. Park Chemical indicates a cost thirty percent less than *Partherm 290*, their equivalent of HITEC. Both of these are high purity and have been found to cause little corrosion to low carbon steel for many years if used at the temperatures required by the cases here analyzed (References 1, 30). Commercial and fertilizer grades of sodium and potassium nitrate have costs, according to the *Chemical Marketing Reporter*, as low as 65-75 \$/Mg (3-4¢/lb). The costs of corrosion resistant materials compatible with the impure salt grades, or intermediate levels of removal of specific impurities to retain the low corrosion levels of carbon steel, would require a tradeoff study.

The third approach is the use of other inorganic materials that are inherently low in cost at high purity levels. Two chemicals, sulfuric acid and elemental sulfur, are near the top of the list in the annual quantity produced, and cost about 80 \$/Mg (3.6¢/lb). Problems of corrosion, of their use as a heat transfer fluid, and of the feasibility of using them in packed rock beds have not been adequately explored, so they cannot now be considered near-term available.

For this selection the lowest cost version that can be considered as near-term feasible is the same packed-bed thermocline configuration used in Selection #9, but with molten salt rather than oil, as represented by the column headed Salt:0.25 in Table 6-3. The comparative cost is 927 \$/kW.

Selection #12 - Phase Change Materials (PCM)

This selection was included to assure that the merits of phase change materials were considered, despite the fact that many of the proposed materials and configurations cannot be considered near-term available. Two reasons for consideration were stated in Section 3: the possibility of reduced storage media plus containment cost because of higher energy densities stored per m^3, and the possibility of improved thermodynamic performance by latent-to-latent heat exchange with a small differential temperature.

Unless the concept of packed-beds of rock with voids filled with oil or molten salt is found invalid, because of possible problems in media compatibility, or ratcheting (settling) effects in the rock bed which endanger the containment, it is difficult to see a PCM medium matching in cost the rock plus heat transfer fluid (of 0.25 or less volume fraction). Rock cost at 16.5 \$/Mg or less must be compared with salts at 60-200 \$/Mg for commercial grade purity and 400 \$/Mg upward for grades currently used for low corrosivity heat transfer fluids. For a working temperature range of say 50°C as has been found roughly optimum in the cases explored, the energy density per kg of the PCM material (from its specific heat plus latent heat of phase changes) would have to be more than an order of magnitude better than that of rock. This appears unlikely.

The other potential advantage of PCM, thermodynamic improvement, would result if higher temperature and pressure output steam could be achieved by a lower slope of the storage fluid profile as shown in Figure 5-18 and 5-23. Such a slope corresponds to a higher value of the mass flow ratio M_c when sensible heat exchange is used. With true latent-to-latent heat exchange, the storage medium profile could be parallel to the charge steam and discharge steam profiles, so that the discharge steam temperatures could be only twice the approach temperature difference α below the charge steam temperature. The impact of this could be a lower value of L and a lower value of C_{pp}, from a higher turn-around efficiency and higher turbine inlet pressure. Conceivably this could lower the sum of these two terms by 60 $/kW, which the increased cost of the storage medium would negate.

The TESS costs of this selection will be taken as higher than the 927 $/kW for rock and salt and approaching all-salt, say at 1500 $/kW. This plus the doubts on near-term availability due to heat transfer problems places the selection low in ranking.

SECTION 7
DISCUSSION OF SELECTION CONSIDERATIONS

In Section 6 the primary emphasis was on economic comparison of the twelve selected concepts, with sufficient sensitivity analysis to give perspective as to the reasons for higher and lower costs. Summary Table 7-1 indicates the results in $/kW$_e$ for the case chosen to represent each selection. For ready reference, the energy-related and power-related costs are also given in separate columns. Since all cases were treated for six hours discharge, the energy-related costs in $/kWh can be found by dividing C_E by six. The rank ordering by TOTAL cost C_T charged to the TESS concept is given in the sixth column.

Table 7-1. Economic and near-term availability ranking.

Selection Number	Short Title	C_E $/kW	C_P $/kW	C_T $/kW	Rank - Economic	Rank - Availability
1	PCIV-FWS	461	462	923	6	4
2	PCPV-FWS	524	495	1019	9	4
3	STEEL-FWS	1129	495	1624	11	1
4	UG-C-VARP	172	477	649	1	3
5	UG-A-FWS	108	667	775	5	6
6	UG-A-EVAP	180	487	667	2	4
7	AQUIFER	75	855	930	8	6
8	OIL-FWS	132	538	670	3	5
9	OIL/ROCK	188	541	729	4	3
10	OIL/SALT	--	--	--		
11	SALT/ROCK	426	501	927	7	4
12	PCM	>1000	--	$\sim$1500	10	8

Although the economic ranks are numbered sequentially, it is apparent that there are several groups with relatively small cost differences. In sequence, #4, #6, and #8 are all in the 649 to 670 $/kW range; #9 and #5 are in the 725-775 $/kW range; #1, #7, #11, and #2 are in the 900-1020 $/kW range; #12 and #3 are distinctly higher. Within each of the groups little attention should be paid to ranking. The degree of optimization and the certainty level on many of the cost components does not warrant it. An uncertainty in each of the storage system components of ±20 percent is easily credible at the quartile level.

However, for the purposes of this report it should be noted that components common to many of the selections should affect those selections similarly. For example, the peaking Turbine Island is a significant part of all the concepts, ranging from 400 $/kW to 530 $/kW. While revised estimates from detailed design of specific turbine configurations could move these costs upwards or downwards, they would probably move comparably and not affect the ranking among the above groups.

Some of the components with significant cost are unique to one selection or a small subset. They may be uncertain in cost because of uncertainties in technology that have not been resolved by adequate development and testing to date. These uncertainties can be considered as a factor in judging the near-term availability of the selected concepts.

For near-term availability, and other criteria that are in part subjective, ranking should not only indicate the best and the worst, but should indicate groups that are very comparable in rating and places in the sequence where there is judged to be a large gap. The scale of one to ten is used, one best and ten worst, with the same rating on similarly valued selections and omitted numbers where there is a large difference in value.

NEAR-TERM AVAILABILITY

A subjective ranking of near-term availability is made in the last column of Table 7-1. None of the selections is completely available off-the-shelf. The definition of near-term availability used in the ranking judgments is that the technical uncertainties have either been resolved by demonstration, or could be so resolved in the near future by industry or government action, so that an electric utility customer could order a TES system with "reasonable confidence" by 1985, for delivery and operation during the period 1985 to 2000.

The principal purpose of discussing the relative value of the selections on this and other criteria discussed in Section 3 is to assess the impact that particularly good or bad features may have on the preliminary ranking by cost. A major fault could move a selection downward, or a unique advantage move it upward. Minor differences will not be emphasized, nor are they likely to alter rankings unless a confluence of many advantages seems to merit it.

Judgment of near-term availability is mostly concerned with technical problem areas not yet resolved. The principal problem areas, potential solutions that have been proposed, and their status will be briefly discussed as justification of the rank ordering assigned. In most cases it is a key component, not common to the other selections that is discussed.

STEEL TANKS. Steel pressure vessels for containment of materials at temperatures and pressures to and beyond those needed for TES (Selection #3) are state-of-the-art. Design practices are well codified and backed by years of operating experience.

UNDERGROUND CAVITIES. The technology of excavating shafts and cavities is well known, from mining, tunneling, and other industrial applications. Problem areas specific to Selection #4 include:

- Competent rock must be found. This limits sites to specific regions and requires exploratory drilling on specific sites. Until actual excavation some uncertainty remains.

- Applications that keep the rock at high temperature have not been demonstrated for long-life effects. Precautions in rock conditions selected, use of adequate reinforcement near the cavity, use of adequate high-temperature high-strength concrete for stress transfer are suitable solutions.
- Cycling in temperature and pressure on a daily cycle has not been demonstrated for long-life effects. The proposed mode of operation as a variable pressure accumulator with modest swings in pressure and temperature should minimize these effects.
- Underground cavity volume required is larger than demonstrated by current technology. Until moderate size cavities (30,000 m^3) have been thoroughly demonstrated, larger cavities such as 100 m diameter cannot be considered near-term available.* Multiple smaller cavities around a common shaft can be used for larger volumes.

UNDERGROUND CAVITIES—AIR SUPPORTED. Selections #5 and #6 are rated somewhat lower than Selection #4 because of additional problem areas.

- The use of compressed air support for a low pressure containment vessel has not been demonstrated. While there are advantages in accessibility to the cavern components, the problems of air leakage out, water leakage in, pressure seals for access doors, cooling of the compressed air, risk of severe pressure swings despite the equalization tank have more technical risk than the concrete-supported cavity.

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*Correspondence and discussions with Dr. Andrew Merritt (Vice President of Deere and Merritt, Inc., consultants in engineering, geology, and applied rock mechanics, Gainesville, Florida) indicated that 30 m span in weakest rock direction is state-of-the-art; that height of 30-40 m and length up to 100 m or more are usually feasible, but that the problems and costs for 100 m diameter cavities could be much worse than estimated by proponent R&D Associates.

OIL/ROCK. The use of a thermocline tank with oil as a heat transfer fluid, and gravel and sand as the storage medium has been demonstrated for a limited time. Some confidence has been gained, but long-term stability requires demonstration.

- Degradation of the oil by temperature, presence of the rock, or the combination causes maintenance expenses. Removal of solid and vapor decomposition products by sludge removal provisions, vapor recovery systems, and refurbishment or replacement will be required. Heat exchangers where thin film fouling can affect performance must be designed for easy cleaning, eg, oil inside the tubes rather than the shell.

- Uncertainties in heat exchangers. General references on heat exchangers give condensing steam to oil heat transfer coefficients as seven to ten times lower than those for condensing steam to water. Partial reasons are lower heat capacity, lower density, higher viscosity for the oil. It is suspected that a large part of the difference is the assumed fouling in such references. The oil and the fouling conditions assumed are not stated. Caloria HT 43, highly purified may be much better. This report, following Martin Marietta's report (Reference 61), used U = 92 (English units), about one-fourth of that used for water. There may be an uncertainty of two to one in heat exchanger costs for oil.

- Settling behavior of rock beds has been suggested as a problem area. If under daily thermal cycling the rock bed contact with the steel tank increases the tensile stresses in the tank as the rock settles, leakage or failure could occur. The effect was not found in the Rocketdyne test of an oil/rock bed. If found to be a problem, possible solutions include: use a form of solid medium that has an expansion coefficient similar to that of the tank, such as taconite; test whether smooth river-bed gravel and sand is better than other shapes such as random size crushed gravel, several mixed sizes with low void fraction, etc; use partial thermal insulation on the inside of the tank so that tank expansion is reduced to match rock expansion.

PCIV. The prestressed cast iron vessel of Selection #1 has not been demonstrated at pressures and temperatures of interest.

- Current emphasis (Reference 45) is on a hot-going PCIV, with external insulation. Operating with the cast iron hot and prestressing cables and tendons cold has potential problems of fatigue failure and creep under diurnal temperature and pressure cycling. Some of the external insulation must withstand high pressure loading. Ample design margin and periodic test and adjustment of cable stresses is one approach to the solution. Finding a form of thermal insulation suitable for use inside the steel liner of the PCIV is another. It must be compatible with boiler quality feedwater and able to withstand high pressure while retaining low conductivity. Siempelkamp is reportedly working on such an insulation (Reference 175) but has supplied no details. Discussions with GE Corporate R&D Laboratory personnel* indicated one possible approach.

- Expansion accumulator mode gives lowest daily changes in pressure and temperature. This would require, for the feedwater storage mode of operation chosen for greatest economic viability, a cold storage volume comparable to the hot PCIV volume. A displacement accumulator mode would eliminate the large cold

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*Telecon with Dr. F.P. Bundy referenced article, "Flat Panel Vacuum Insulation" by H.M. Strong, Bundy, et al, in *Journal of Applied Physics*, Volume 31, 1960, describes mat of glass fiber layers, alternating in direction, built up to 2 cm thickness with thousands of layers, encased in 0.1 mm stainless steel foil and evacuated. Conductivity approached that of Dewar flask. Was tested to 4-5 bars but not to 40 to 50 bars required for PCIV.

tank, but to operate in a thermocline mode the internal thermal insulation would be required. Such insulation would not only greatly reduce the thermal stress caused in the liner by a thermocline but would greatly reduce the vertical conductivity effects in the internal liner which tend to degrade a thermocline.

PCPV. The prestressed concrete pressure vessel, like the PCIV, has not been demonstrated at the temperatures and pressures of interest. Many very large PCPVs have been used at lower pressures (0.3 MPa and one example at 3 MPa). There can be considerable confidence in the technology and design principles. Hot-going systems are not feasible so some kind of cooling system is required outside the liner and layer of high temperature concrete.

- Cooling systems to maintain desired temperature distribution yet minimize thermal losses from storage must be devised. Active systems can be used in which finned tubes, water-cooled, are embedded at the outer interface of high temperature concrete. Internal insulation can play a role, particularly if the displacement accumulation is to be used.

SALT/ROCK. There has been less reported experimentation on the compatibility of molten salt and rock than that reported for oil/rock. Sandia Livermore is conducting static tests on degradation rates.

- Degradation rates could be excessive with some forms of rock, eg, dissolving of some rock constituents. Exploration of alternative low cost rocks and minerals is one approach. Filter for solid degradation products, refurbishment, makeup, and eventual replacement of the salt and/or rock would be maintenance and operating expenses, to be considered in economic evaluation.

- Heat-exchanger fouling does not appear to be a problem with pure salt, heat transfer is very good, comparable to water. Effect of degradation products from interaction with rock are not known.

- Lower cost forms of molten salt such as impure HITEC and draw salt are not near-term-available until thorough tests on corrosion and materials compatibility are made.

AQUIFER AND PCM. Both of these have been labeled as not near-term-available. They are also low in economic ranking.

SUMMARY. Although Selection #3, STEEL, is most available, it is also most costly. Availability is not considered to overcome the cost obstacle. Four out of the top six in availability are also in the top six in cost ranking. Selection #6 ranks better than Selection #5 on both criteria, suggesting that only #6 of these two similar selections be retained unless other criteria indicate otherwise strongly.

UTILITY OPERATING REQUIREMENTS

Site Flexibility

Of the twelve Selections, four are limited to suitable geologic areas. Selections #4, 5, and 6 require competent rocks, suitable for excavation with minimum reinforcement and minimum risk of catastrophic failure or seismic damage. Granite rocks, and other intrusive igneous and crystalline basement type rocks are preferred (Reference 28). Limestone, marble, and other metamorphic rocks not excessively fractured, and old, well-cemented sandstones are also feasible. The above reference displays a map suggesting that roughly one-third of the United States is underlain by potentially suitable rock formations. Including the major mountain chains, all of New England, Wisconsin/Minnesota/Dakotas, and scattered areas in the rest of the country, the suitable regions probably are included in the utility areas serving well over half of the population.

Reference 28 also suggests massive salt deposits as suitable. While excavation costs can be very low, using solution mining, these have not been included in any selection on the grounds of technical risk. Extensive use of such cavities has been made for storage of natural gas and reserves of petroleum. The salt is somewhat plastic under high pressure, even at ambient cavity temperatures. There have been reported cases of partial closing in before enough gas or oil had been emplaced to match the ambient pressure at depths. At high temperatures, using liners that contain pressurized HTW, the problems of leak-proof containment and avoidance of failure from salt plasticity do not appear to have near-term available solutions.

Selection #7 requires suitable aquifers. Sedimentary geology with potentially suitable groundwater layers underlies about half of the United States. Suitable regions are widely dispersed and probably occur within the utility areas serving over two-thirds of the population. In addition to the existence of aquifers, site selection must consider that they must be deep enough to support the pressure of injected HTW without flashing to steam; must be confined, ie, having a retaining impermeable layer of clay above the aquifer and preferably also below it; and must not interfere with potable aquifers for municipal water supplies.

Other aspects of site flexibility are land requirements and aesthetic acceptability. The underground selections use little land and show little visible profile. Disposal of the muck from an excavated cavern poses an aesthetic problem or disposal problem, but often it is salable or can be used for other on-site construction. The PCIV and PCPV require large arrays of storage vessels. The proposed PCIV module is 70 meters high; two or three would be needed for Selection #1 as feedwater storage. As many as twenty could be required for a 2400 MWh TESS. The PCPV would probably be designed with a lesser height to diameter ratio, but would have very thick concrete walls giving a large total visible volume. Location near populated centers might encounter aesthetic objections.

Operating Flexibility

POWER SWING. In the course of the study, discussions with several utilities indicated less interest in small peaking increments such as 5, 10, or 15 percent of the base load plant capacity, than in larger peaking increments such as 30 to 50 percent. On this basis, large power swing capabilities were emphasized over the limited swing available from feedwater storage. Some of the points discussed were:

- Commonwealth Edison is currently purchasing 500 MW coal-fired cycling plants (shut down but kept hot overnight).
- If TESS are economically viable, introduction into the generation mix would be faster with 400 MW peaking capacity supporting each added base-load plant than with only 100 MW peaking capacity per plant.
- Generation control for short-term load-following is only applied to a few plants. If load-following with a TESS while keeping the Boiler Island at constant output proves advantageous, fewer plants with larger peaking capacity are preferred.

On this criterion, Selections #1, #2, #5, and #8, small swing feedwater storage, would be somewhat downgraded compared to the other selections.

In the course of modifying reference plants for Baseline/TESS, a number of the changes made in Section 4 were for ease of control and transient stability, and for a capability of rapid load-following. These of course apply to all selections.

DISCHARGE HOURS. Operating flexibility is also concerned with the number of hours of discharge at full capacity that is available. The energy-related component of cost is roughly proportional to the hours of discharge whereas the power related component is not. For this study, 6 hours discharge and 8 hours charge were selected as a uniform basis for comparison. Since the relative cost of the energy-related and power-related components differs for the selections, the ranking may be altered for a different design with more or fewer hours of discharge. This is illustrated in Figure 7-1.

The TESS cost in $/kW is plotted against the number of hours of discharge capacity built into a TESS plant. At zero hours, the points represent power-related costs alone from column 4 of Table 7-1. At six

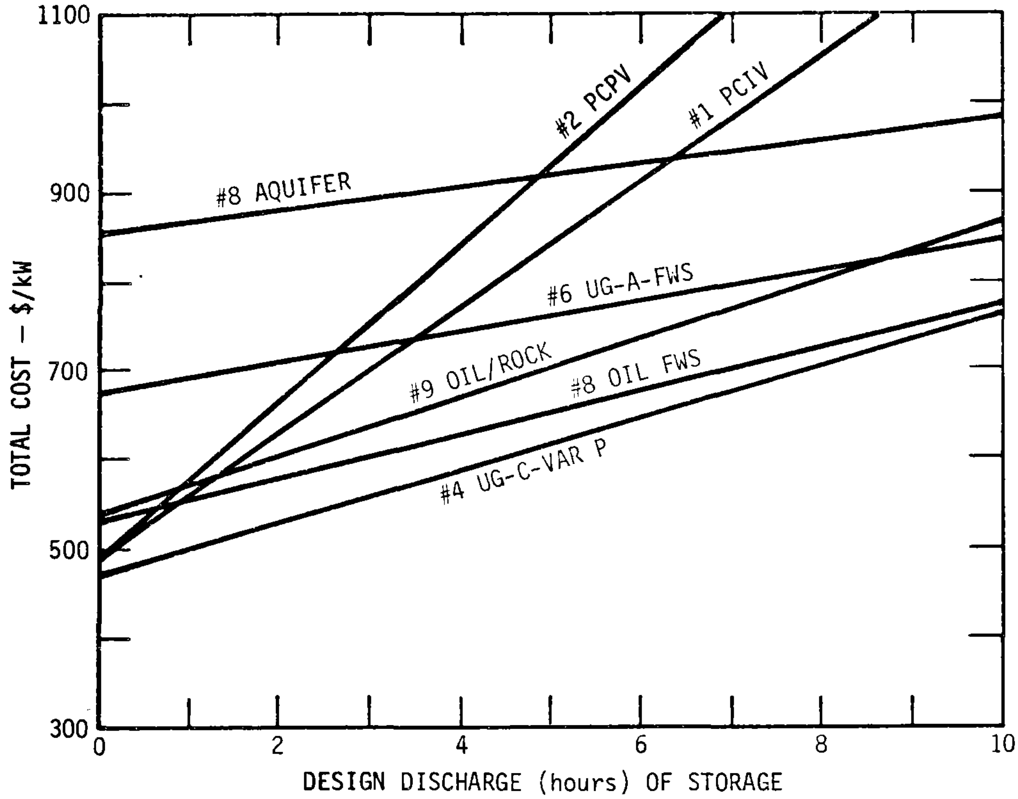

Figure 7-1. Comparison of capital cost of selections from different discharge cycles.

hours the points are the total costs in column 5. Some of the high-cost-per kWh systems such as PCPV and PCIV cross over the oil/rock systems with higher power-related costs at about two hours discharge capability. Aquifer storage, not very attractive for short discharge designs, has a low slope, and would cross all the other lines by 48 hours discharge requirement. It is thus most suitable for long-term or seasonal storage.

The display in Figure 7-1 resembles the "screening curve" in Figure 4-4 in which annual costs per kW are plotted against capacity factor or hours of output per year. The resemblance is deceptive but the differences do not alter the crossover points (in hours of discharge) of the TESS selections.

Figure 7-2 shows the differences in graphic form. Two hypothetical selections are shown, #1 and #2, which intersect. The scale for ordinate can be $/kW as in Figure 7-1 or can be changed to capital costs per year by using the fixed charge rate as a scale factor. The fuel costs or variable costs per year can be added to both selections so the scale represents total $/kW-a. Since the fuel assumed for both is the same and differences in the turnaround efficiency have been included as a capital cost equivalent in C_E, the same amount is added to each as shown by lines 1A and 2A.

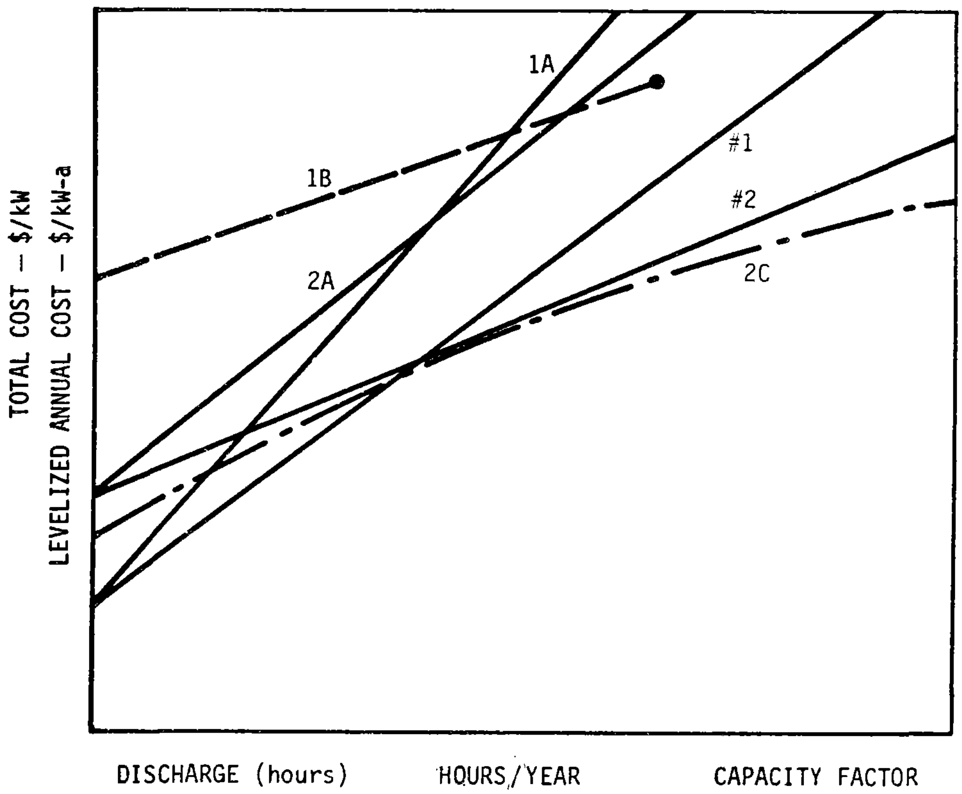

Figure 7-2. Alternate scales for comparisons of TESS selections.

The hours of discharge can be converted into hours per year or a capacity factor if suitable assumptions are made. One can assume that for all designs the TESS operates for its rated discharge period on 250 days a year, ie weekdays but not weekends. This will not alter any of the curves if a scale factor is used such that 6-hour capability equals 1500 hours per year equals 0.17 capacity factor. The lines, however, still represent distinct plant designs with specific maximum discharge capability for each point. A specific plant, rated 6 hours discharge, would have a screening curve indicated by dashed line 1B if for various reasons it were to be operated at more or less than 250 days per year, up to a maximum of 365 days per year at capacity factor equal to 0.25. The capital cost C_E, in the plant is committed; only the fuel cost varies with capacity factor, so the slope of line 1B is less. For designs with other discharge capabilities, the screening curve would be parallel to 1B but higher or lower.

Another point can be made with this generalized figure. As plant designs incorporate more or less hours of discharge capability, the capital cost would not be a straight line if each plant were optimized. One would expect the true curve to be concave downward as shown by dot-dash line 2C.

Reliability

One of the objectives of the use of TESS is to improve the boiler island outage rates by minimizing the output variations required of it. It has also been indicated that reliability could be improved (availability increased) if the peaking turbine can be operated from storage when the boiler island is shut down or from the boiler island steam source if the main turbine is shut down. Both appear feasible at some cost. In any case effects apply equally to all selections except the feedwater storage selections using an enlarged main turbine. Even the feedwater storage selections could use a separate peaking turbine representing the differential capacity that would have been added to the main plant. Turbine design would probably be more difficult and costs higher than shown in Table 7-1.

Reliability can of course be affected by forced outage rates, and the amount of scheduled maintenance required of the TESS components. It is difficult to judge relative proneness to outage except as a function of technical uncertainty, as imbedded in the absence of adequate demonstration of performance. The highly modular construction incorporated in the various selections to use sizes that have least technical risk (eg 3 to 20 PCIVs; 16 oil/rock storage tanks; 5 to 35 parallel heat exchangers) should assure reliable operation providing isolation devices such as stop-valves and control features are adequately designed.

Maintenance requirements of heat exchangers can be expected to be more time consuming in selections using oil than in those using molten salt or HTW. Maintenance in an underground cavity, while hopefully seldom needed could require an outage of many weeks to many months while cavities are emptied and cooling is used to make manned access feasible. Molten salt systems, if shut down and allowed to cool below their freezing point will require electric or steam tracing in all pipes, heat exchangers, and storage tanks to restore flow after repairs. An alternative is to use a system that introduces water as the molten salt cools down, so that a liquid or slurry is maintained down to ambient temperatures (Reference 1). Extra equipment and more tankage would be required.

OPERATING HAZARDS

It can be expected that electric utilities would be reluctant to adopt a TESS concept that potentially endangered the conventional plant components such as boiler or nuclear steam supply, main turbine generator, electrical, and heat rejection systems. Such hazards would most likely occur at the interfaces of the TESS with the main power plant. Precautions must be taken that the quality of boiler feedwater, for example, is maintained at utility standards. Small leakages of foreign materials into it can cause corrosion and scale. Fairly small particles of scale, knocked loose and passing through the turbine can cause blade erosion or even blade failure. Any appreciable incursion of oil or molten salt would make necessary an expensive decontamination outage.

Oil and salt, and potentially granules of rock and sand, interface with HTW and steam in heat exchangers. In case of heat exchanger leakage, it is preferable to arrange the pressure on the water/steam side to be everywhere greater than on the oil or salt side. HTW storage systems will probably have lesser hazards from boiler feedwater contamination, but all parts of the storage systems, tanks, pipes, and pumps must be cleaned and kept clean, and be of suitable corrosion resistant materials.

Avoiding risks to the boiler island in a conventional plant was one of the reasons for opting to eliminate the reheater from Plant #1 as was discussed in Section 4.

ENVIRONMENTAL ACCEPTABILITY

Environmental requirements on the main plant play a major role in site selection, so limit site flexibility. In addition to main plant constraints, unique features of the TESS selections must be considered for their environmental acceptability. All of the aboveground selections require a large volume of tankage. Many tanks can be fairly low and comparable to other structures of the main plant. Of the various selections, the PCIV, Selection #1, probably has the greatest height and visibility, about 70 m, but not in excess of fossil plant stack heights.

Particularly noxious materials, in terms of odor and toxicity, have been avoided in the selections being considered. Sulfur and sulfuric acid, while potentially very low cost heat transfer fluids, may complicate site approvals by environmental objections.

Containment of the storage media in case of a catastrophic failure must be provided for in the case of oil and molten salt, but probably not for HTW. The danger from major release of hot oil is fire. The danger from the release of hot molten salt is less if the area around the tank is kept well cleared of oxidizable material.

CONSERVATION POTENTIAL

Conservation objectives include the saving of energy, and especially the saving of depletable and imported fuels such as petroleum and natural gas. Thermal energy storage and other storage systems do not save energy in that the turnaround efficiency indicates less electric energy is being produced from fuel than could be obtained from the base load plant. Although load-following with a base load plant will give a poorer heat rate at low load operation, it would in general average more efficient than a TESS equipped plant.

However, if a TESS is compared to alternative methods of storage there can be energy savings. The turnaround efficiency of a pumped hydro plant is about 0.65 to 0.70. In the analyses in this study the turnaround efficiency found is quite high for the selected concepts. It is roughly in the range 0.85 to 0.91 for the HTW systems and 0.79 to 0.85 for LVP systems. This is significantly better than pumped hydro and than most of the other storage means can claim.

These results are somewhat higher than those given by some of the proponents in the references cited. This is in part due to selection of a steam source and peaking turbine throttle pressure that do not unduly penalize turnaround efficiency to get a high specific output. It may also be due in part to assumptions that are more optimistic than used by said proponents.

The use of cold reheat steam, or IP turbine inlet steam, as a source minimizes the availability losses from throttling and loss of superheat on charging. Use of a conservative throttling range on variable pressure and multiple evaporator systems keeps the turnaround efficiency up. Use of a large mass flow ratio M_c in LVP systems also gives a higher inlet pressure and temperature at the peaking turbine.

It must be acknowledged that some of the turbine assumptions made may be optimistic. Elimination of reheat in both Plant #1 and Plant #2 for ease of control, and simplicity of analysis will increase the moisture content of the LP steam flow. While the steam extraction points will serve a moisture separation function, the possible reduction of LP turbine efficiency by higher moisture content has only been allowed for qualitatively. Effects of this should affect the selections comparably, without reversals in ranking.

Heat exchanger assumptions may also be optimistic, as has been mentioned. If heat exchanger costs are found to be higher than assumed, reoptimization would suggest a larger value of the approach α, at some penalty in turnaround efficiency.

OIL SAVINGS. When compared not to load-following by base-load plants, but to the alternative peaking means, such as gas turbines which use distillate or low sulfur petroleum fuels, or to compressed air storage which uses some oil fuel during the discharge cycle (about one-third as much as the gas turbine), there is conservation potential in thermal energy storage.

If the TESS charging cycle uses nuclear or low-cost coal as fuel and the peaking turbine output replaces gas turbine power output, oil is conserved. The amount and type of fuel replaced by TESS operation is most accurately determined by an hour by hour simulation of the dispatch procedures used by electric utilities with a given mix of generating capacity types and a given pattern of daily, weekly, and annual demand variation. Some of this simulation is planned for a later task in this project. Some preliminary results indicate that in an assumed utility system with generating capacity that is 27 percent nuclear, 39 percent coal, 19 percent oil/steam, and 15 percent gas turbines, more coal and less oil were burned when TESS was used, even when the TESS was a part of a nuclear plant.

The explanation of what fuel effectively replaces the peaking fuel (oil) arises in the utility dispatch procedure, and is illustrated in Figure 7-3. In most cases the utility dispatch is done on a *production cost* basis, ie when the demand is increasing the reserve unit with the lowest variable cost per kWh (fuel and variable O&M) is started, and when demand decreases the operating unit with highest variable cost per kWh is reduced in load, or shut down. In some cases *environmental dispatch* may override economic considerations on occasion (eg Southern California Edison). Capital costs of units are not considered since they are committed and unchanging. Current fuel costs, not levelized, are used.

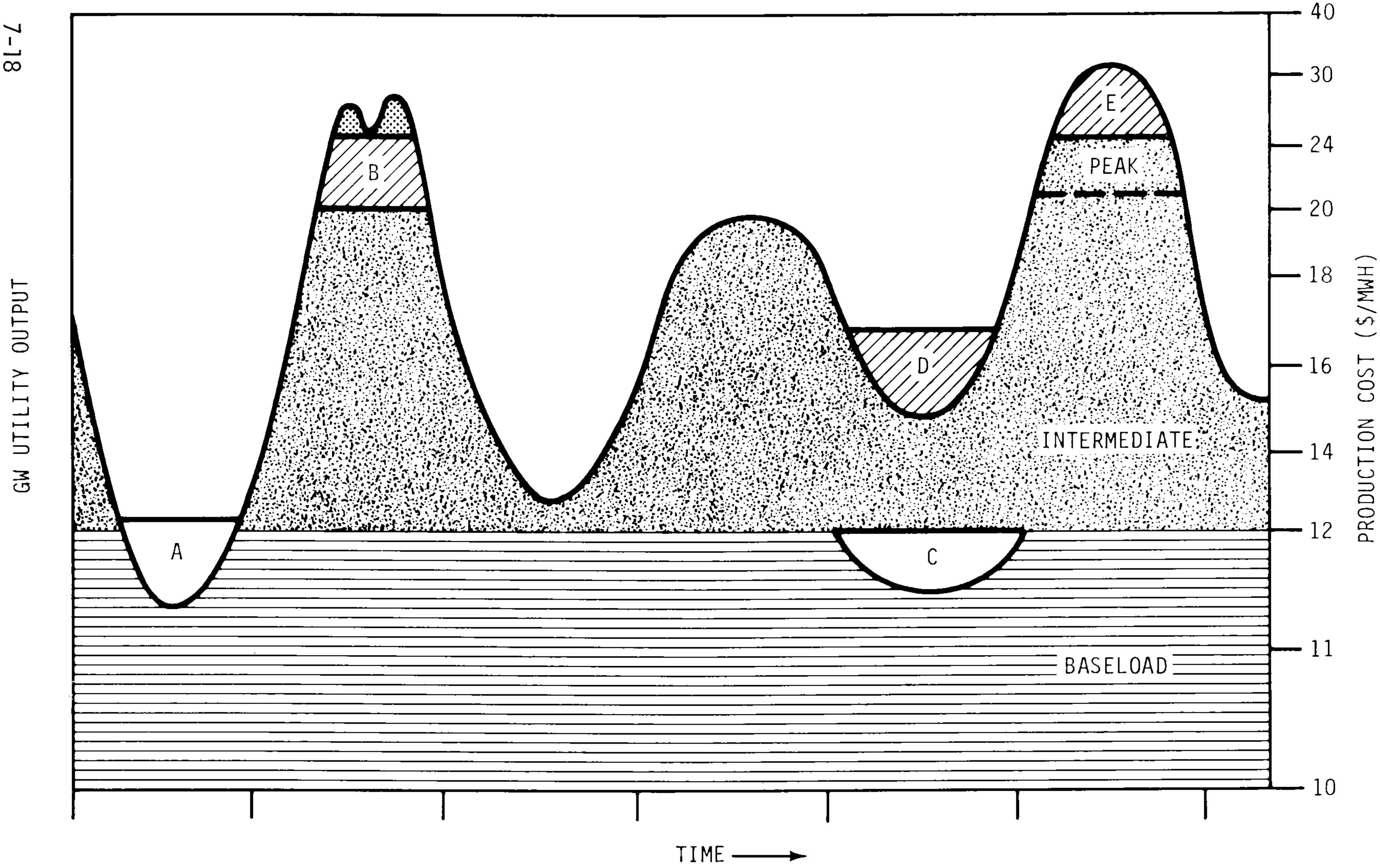

Figure 7-3. Criteria for judging utility dispatch of storage.

The daily demand swings shown in Figure 7-3 are extreme to illustrate several points, and may represent days in different seasons, weekdays, and weekends. At the right is an illustrative set of numbers indicating the production cost in $/MWh of the type of generating capacity normally started or stopped when demand moves through that power level. If the trough in off-peak hours is as low as is indicated at A, area A can represent energy charged into storage at nuclear fuel costs of about 12 $/MWh. It is discharged at B at a level where gas turbine or oil/steam costs are about 20 $/MWh. To be competitive for this application, the turnaround efficiency has to be greater than the ratio of production costs at A and B, 12/20 or 60 percent.

On another occasion, when the trough does not go to the base load level, a nuclear plant TESS could be used, with the energy represented by area C used for charging. However, to meet the demand, this means that a unit at the 16 $/MWh level must be kept operating to deliver the energy in area D. With the energy discharged at area E, the turnaround efficiency must be at least 16/26 or 62 percent to be economic dispatch. Note that in this case the fuel effectively used for charging is probably coal rather than nuclear, and may be oil/steam in areas where coal plants are not prevalent (eg Southern California Edison).

The hour-by-hour simultions have given other preliminary results concerning the usefulness of TESS plants for different levels of penetration of TESS (ie percent of total capacity represented by TESS) and with discharge capabilities longer or shorter than six hours. Figure 7-4 illustrates the effect of penetration, ie the third increment of penetration is not used to generate as much energy per week as the first 5 percent increment, therefore has less value to the utility. The variation with discharge capability in hours shows that the energy generated per year by the TESS varies linearly with the storage capability in hours up to about four to five hours, but is saturated by six hours and increases very little from six to ten hours.

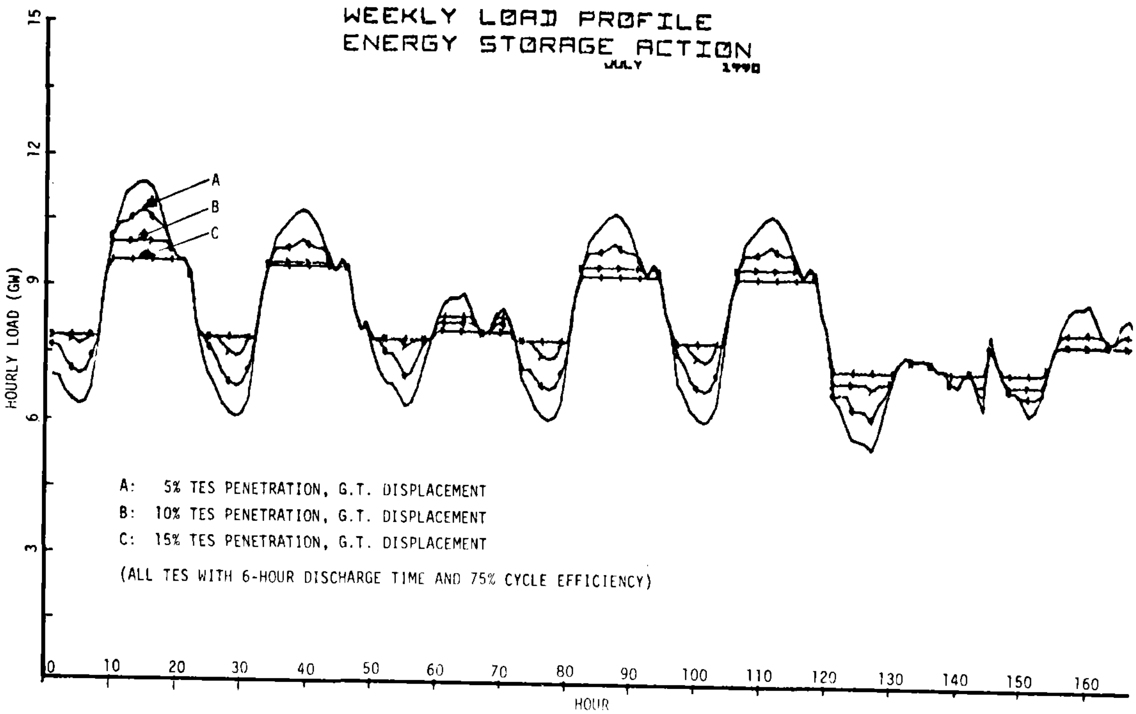

Figure 7-4. Weekly load profile of energy storage action.

Since the power-related costs of TESS are independent of the storage hours capability, the effective cost of electricity produced by TESS will decrease from infinity at zero hours capability to a minimum near six hours, and then rise again as the energy-related costs increase without corresponding energy output.

DIVERSITY

The last criterion, as listed in Section 3, is diversity. Judgment must be used to assure that all selections recommended for further conceptual design are not simply variants of one concept. For example, on the basis of the foregoing discussion, all recommendations should not be underground cavities, though three out of the top ranked five (Table 7-1) are UG cavity concepts. Nor should all be variants of LVP systems with oil as the heat transfer fluid. All should not be regionally limited by geology. Growth potential considerations, frequently mentioned in the preceding sections, should be considered so that selecting the most available does not foreclose future improvement in cost and performance.

Some judgments on the basis of diversity, bearing in mind the other criteria, are fairly easy. Because of geologic specificity, at most one selection should be underground. Since Selection #4, the UG cavity, concrete supported, variable pressure accumulator concept comes out best of all in economic ranking, it should be one of those selected, excluding Selections #5, #6, and #7.

The similarity in all system details except the pressure vessel of Selection #1, PCIV, and Selection #2, PCPV, suggests that at most one of them should be included. Present data favors somewhat the PCIV; if more detailed conceptual design indicates problem areas or major cost revisions, a conversion to the alternative pressure vessel can be made.

LVP systems are fairly similar in configuration, whether oil, molten salt, or another medium is used. All appear relatively unattractive if difficulties are found with the dual media concept of oil/rock/thermocline. At 25 percent or more volume fraction of fluid, oil (Selection #4) appears to rank higher than molten salt (Selection #11) in economics and availability. For drained-tank concepts or for cost reductions of salt through purity/compatibility studies, molten salt offers more promise. As these growth directions are not as near-term, Selection #9 must be preferred to #11.

Although feedwater storage systems are limited in peaking capacity, they are attractive in specific output as illustrated in the comparison of Selections #8 and #9. Although diversity considerations would not indicate that oil/rock systems should be two out of three selections chosen, both could be considered in a group of four choices. If desired, one of these could emphasize oil and the other emphasize molten salt to inject an additional difference. In this case, oil is indicated for the feedwater storage because of the larger temperature swing used, which would extend below the freezing point for HITEC.

COST OF ELECTRICITY

Another sometimes useful economic measure of storage concepts is the cost of electricity (COE) in dollars per megawatt hour ($/MWh) which is numerically equal to mills per kilowatt hour since a mill, in metric terminology, is a millidollar (m$). It will be noted that to this point the emphasis in comparing storage concepts, the primary objective of this report, has been on the capital cost of the storage-related components rather than the COE.

In Section 5 (pages 5-65 to 5-66) the equations for the COE of a baseline plant and the COE of a plant incorporating a thermal energy storage system were compared. Each was defined for the load-following situation in which a peak power level was produced for H hours a day, a minimum power level was produced during off-peak hours, and an intermediate or *normal* power level was produced for the remaining hours per day. The daily load pattern was defined in terms of H and the ratio, p, of the power increment between peak and normal power to normal power. The equations differed in two ways. The Baseline/TESS system included in the numerator a term including the power-related and energy-related capital costs of storage in $/kW, and in the denominator a loss term that is dependent on the turnaround efficiency. This permitted deriving L, a capital cost equivalent of the turnaround efficiency in the various concepts to be compared (page 5-67). A byproduct of the analysis was a value for the capital cost of storage that would give a COE identical to the baseline plant load-following in the same pattern as the Baseline/TESS plant, ie (743 - L) $/kW for Plant #1 and (785 - L) $/kW for Plant #2.

The value for COE of a Baseline/TESS plant can be useful in giving additional perspective in the comparison of TESS plant concepts, or in the comparison of TES with other forms of storage or with other means of peak-load generation. However, great care must be used in assuring that all the economic assumptions made in COE for TESS plants match the assumptions made in the other systems to which they are to be compared. There are many more assumptions involved in the COE than there are in the comparison of capital costs, and correspondingly, chances for error and ambiguity. These will be explicated.

Dedicated Plant Concept

As the storage concepts of interest are dedicated storage attached to a specific power plant and not separately operable (as are battery, flywheel, magnetic, and pumped hydro storage), one obvious way to consider the COE of the concepts considered in this report is as the COE of the entire plant. The COE of the reference Plants #1 and #2 were found to be respectively 44.60 and 43.14 $/MW (page 4-15). This was for the series of assumptions made on financial practices, fuel cost scenarios, and the availability (0.723) as derived from Reference 172 (EPRI, *Technical Assessment Guide*, August 1977).* These base-load plants can load follow, and do so to some extent in practice. If operated in the same load-following pattern as a Baseline/TESS plant with a peaking swing, p, of 0.50, the base-load plant would be at full load 6 hours per day, two-thirds load for 10 hours and at about one-third load for 8 hours corresponding to the discharge, normal, and charge modes of operation. The capacity factor would be less than 0.723 to account for the reduced energy output per day: $F = \frac{6}{24} \cdot 1 + \frac{10}{24} \cdot \frac{2}{3} + \frac{8}{24} \cdot \frac{1}{3} = 0.639$ (see page 5-65). From Equation 5-15, the COE for Plant #1 load-following is 57.00 $/MWh, and from Plant #2, 56.20 $/MWh.

A Baseline/TESS plant, with the same load-following cycle and same p would have the same COE if the capital cost in $/kW of the added TES components equaled the baseline plant capital cost less the loss factor. A higher cost than this for TES would give a higher COE and vice versa.

As an example, for two extreme cases in Table 7-1, Selections #3 and #4, the COE derived from the equations on page 5-67 and the data on page 6-11 are 51.70 and 53.80 $/MWh, even though the capital costs of storage for these cases are respectively 1624 and 649 $/kW. The inversion,

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*Note: The newly available June 1978 version raises the estimated availability by 10 percent, and makes minor changes in many other parameters.

lower COE for the more expensive storage, results from the fact that Selection #3 is a feedwater storage case with only 15 percent peaking while Selection #4 has 50 percent peaking. In both cases the economic comparison of storage concepts is strongly diluted by the base load output of the plant which greatly exceeds in energy the peaking output. The dilution is greater for the smaller peaking output.

Incremental Costs of Storage

For comparison of TES concepts with each other and with other sources of electricity a better approach to COE is the segregation of the TESS-related capital costs and fuel costs as though it were a stand-alone system. This also has its pitfalls.

Table 7-1 gives both the power-related, C_P, and energy-related, C_E, capital costs of the TESS for the selections considered. These include the cost of the peaking Turbine Island, C_{PP}, and the loss term, L. The former is appropriate to retain; the latter, as given in Tables 6-1, 6-2, and 6-3, should be subtracted, and turnaround efficiency included in fuel costs.

As in Equation 5-14 (page 5-65), the COE is found for such an incremental system by combining the fixed and the variable costs per year and dividing by the amount of energy produced during the year.

$$\text{COE} = \left(\frac{C_S \cdot 1000 \cdot 0.1857}{8760 \cdot 0.723 \cdot 6/24}\right) + \frac{7.09 \cdot 1.158}{0.36 \cdot \eta} \quad \$/\text{MWh} \qquad (7\text{-}1)$$

The first term represents fixed costs, C_S is the TOTAL cost of the storage system in \$/kW. The factor 1000 converts it to \$/MW; 0.1857 includes the fixed charge rate on capital and the fixed O&M (Table 4-5). The denominator represents hours of operation per annum: 0.723 · 8760 available hours out of which 6 hours out of 24 are for storage discharge. This product, 1583 h/a is similar to the 1500 h/a often assumed. The second term is variable costs. The levelized cost of coal, 7.09 \$/MWh, a factor 1.158 to include variable O&M, and 0.36, the Plant #1 cycle efficiency, are also from Table 4-5. The turnaround efficiency, η, is included in the denominator to increase the fuel cost incurred per MWh electric.

The two parameters that characterize the selections in Table 7-1 are C_S and η. Using the values from the tables in Section 6, the last column of Table 7-2 shows the range of COE when the dilution effect of an associated baseline plant is removed. It now ranges from 99 to over 200 $/MWh.

Table 7-2. Cost-of-electricity comparisons.

Selection Number	Short Title	C_S (C_T - L) $/kW	COE in $/MWh Fuel:Coal
1	PCIV-FWS	900	129.70
2	PCPV-FWS	993	142.44
3	STEEL-FWS	1598	213.44
4	UG-C-VARP	611	98.99
5	UG-A-FWS	749	113.80
6	UG-A-EVAP	617	101.13
7	AQUIFER	904	132.00
8	OIL-FWS	640	102.58
9	OIL/ROCK	674	109.35
11	SALT/ROCK	871	131.57

Which Fuel is Used?

Although physically the storage system of one of the TESS plants considered is charged by energy derived from the fuel used by that plant (coal in Plant #1, nuclear fuel in Plant #2), the discussion on pages 7-17 to 7-19 indicates that in effect another more expensive fuel may control the actual production cost of charging. This applies when the utility's base load capacity with less than a particular production cost is not large enough to have idle generating capacity during the off-peak hours. If the nuclear plus coal base load capacity is greater than the minimum demand, as shown at A in Figure 7-3, then column 4 in Table 7-2 is a reasonable cost estimate of COE.

If, at some future time, the nuclear capacity alone is greater than the minimum demand troughs, TESS systems associated with Plant #2 can give the lower COE associated with nuclear energy production costs. For illustrative purposes, in Table 7-3, the same TESS costs are combined with the nuclear fuel costs of Table 4-5 in the third column for selected cases. The combination of lower current nuclear fuel costs, a higher escalation rate and levelizing factor and slightly smaller variable O&M costs makes the nuclear plant production costs about 13 percent smaller than the coal plant costs in Table 7-2.

Table 7-3. Cost of electricity: variations with assumptions.

Selection Number	Short Title	Cost of Electricity — $/MWh TESS Plant Alone Nuclear	Oil	Baseline/TESS	
1	PCIV-FWS	126.53	199	47.35	#2
4	UG-C-VARP	95.71	172	54.90	#1
8	OIL/FWS	110.69	183	46.39	#2
9	OIL/ROCK	105.71	190	55.50	#1
11	SALT/ROCK	127.93	212	59.53	#1

Even if the TESS is applied to Plant #2, if the nuclear capacity is insufficient, and base-load coal plants are at the minimum demand level as at D in Figure 7-3, Table 7-2 would apply. If, as is currently the case in the Northeast and in California, the marginal capacity dispatched at the minimum demand level is oil-fired steam plants with less than the efficiency of the most modern plants, the COE in column 4 of Table 7-3 would apply. This is derived from the TAG assumptions on oil of 2.84 $/million Btu by 1990 in 1976 dollars, and a levelizing factor of 2.6. An efficiency of 0.30 for such an older intermediate plant gives 84 $/MWh as the production cost used for column 4.

For ready reference, the *dedicated plant* COE for these selected cases is shown in the last column of Table 7-3. These were derived, as described on page 7-23 using Plant #1 (coal) with Selections #4, #9, and #11, and Plant #2 (LWR) with Selections #1 and #8.

Thus it can be seen that the COE is very much a function of many utility parameters and not a unique number. Other forms of storage and peaking capacity will also involve these parameters so comparison with TESS plants may be made, but must be done very carefully and explicitly to avoid ambiguity and error.

CONCLUSIONS

On the basis of the considerations discussed in this section, the selections in Table 7-1 are ranked as follows:

A. <u>Selection #4</u>

Underground Cavity in Hard Rock; high strength concrete stress transfer from liner to rock. Use in variable pressure accumulator mode. Apply to Plant #1, 800 MW high sulfur coal. Design for peaking capacity ~400 MW.

B. <u>Selection #9</u>

Oil/Rock-Bed with Thermocline; heat exchangers with oil on tube-side. Oil is Caloria HT 43, rock is riverbed gravel of at least two sizes for < 25 percent void fraction. Apply to Plant #1, 800 MW HSC coal. Design for peaking capacity ~400 MW.

C. <u>Selection #1</u>

PCIV as expansion mode accumulator. Use for feedwater storage configuration. Apply to Plant #2, 1140 MW LWR. Design for peaking capacity of 180 MW.

D. <u>Selection #8</u>

Oil/Rock in feedwater storage configuration. Apply to Plant #2, 1140 MW LWR. Design for peaking capacity of 180 MW.

E. <u>Selection #11</u>

Molten Salt/Rock, similar to B except that HITEC or equivalent is used.

The first three, A, B, and C, are recommended as a minimum, balanced set of concepts warranting more detailed conceptual design. The fourth one is an alternate that is attractive in its cost per kilowatt unless heat exchanger assumptions made are too optimistic. The feedwater storage mode with low peaking capacity but high turnaround efficiency and specific output would be explored with both HTW and LVP storage modes if this were included.

For diversity, D is considered as applied to a nuclear plant, as is choice C. While the use of increased main turbine capacity is generally assumed with feedwater storage, the use of a separate shaft turbine generator for the increased capacity should also be examined. Nuclear plant turbines approach the limit of current technology in present sizes, so that three duel-flow LP turbines are currently necessary to achieve the desired capacity. Rather than an eight-flow turbine or development of increased flow-area turbines, a separate turbine might minimize development time and the lengthy approval time for any changes in nuclear systems.

The fifth selection named could be an alternative to B if it were decided that two oil/rock systems did not give enough diversity. Molten salt is currently somewhat behind oil in proven availability and in cost of the storage medium but the growth potential is attractive.

The cost information on these recommended selections is summarized in Figure 7-5.

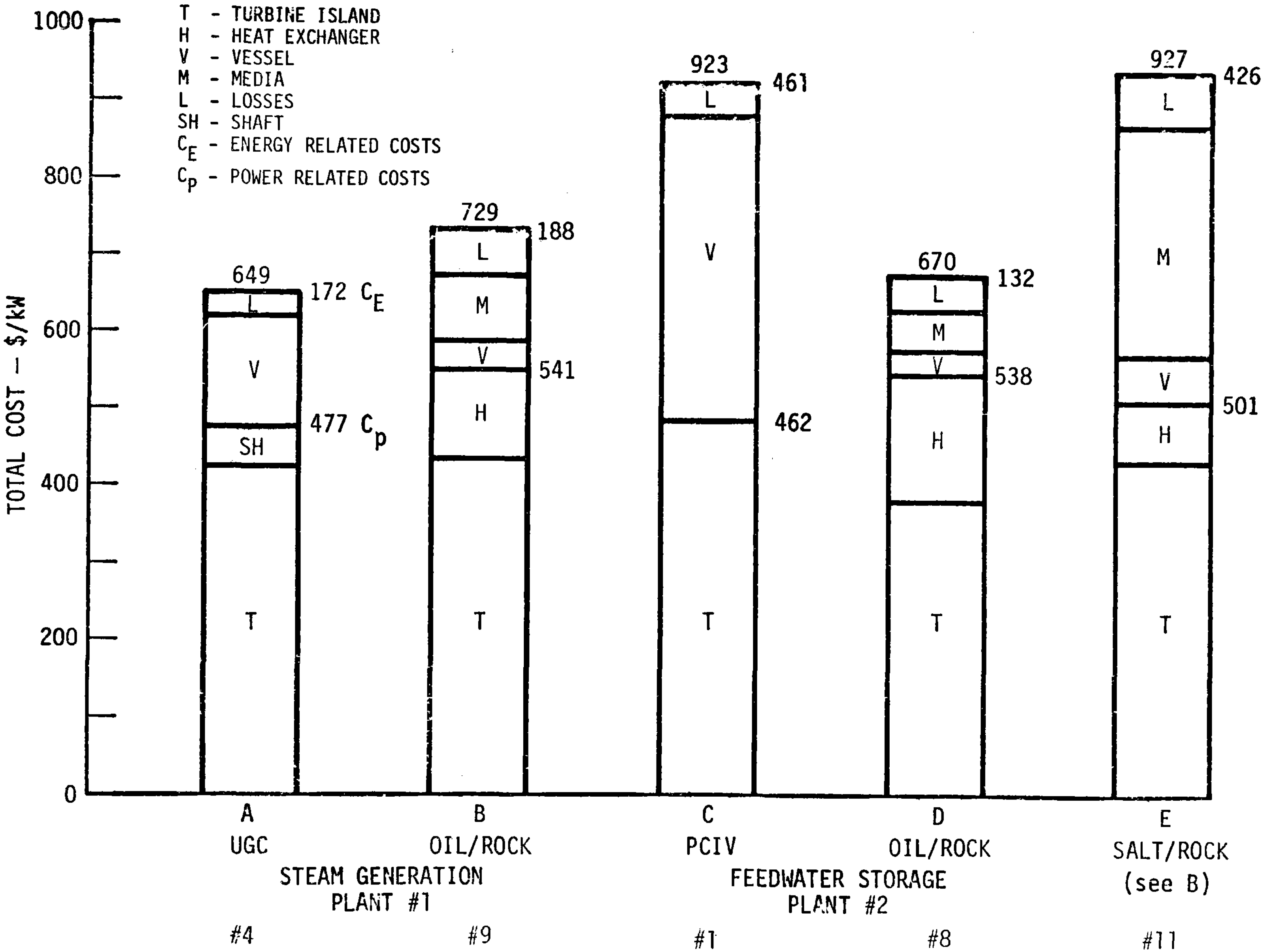

Figure 7-5. Summary of data on recommended choices for further study.

SECTION 8
SUMMARY OF RESULTS

The project task reported is the identification and screening of many thermal energy storage concepts for their relative merit for electric utility applications. Criteria for evaluation emphasized cost, near-term-availability, ability to meet utility operational requirements, and conservation potential. Geographic applicability, environmental requirements, growth potential and diversity of type were considered in the screening process.

PRELIMINARY SCREENING

From literature search and followup contacts, over forty concepts and variants thereof were identified and described. Additional references without system concepts supplied data on materials and components. The distinctive features of the concepts defined were classified as to storage media, form of containment, source of the thermal energy and its properties, and means for converting the stored energy to electric energy, so that other combinations of elements could be synthesized if advantageous.

Preliminary screening of these concepts and their elements, primarily for applicability to the electric utility application and for potential near-term availability, ie commercialization in the period 1985-2000, was used to synthesize a preferred set of twelve selections that incorporated the most promising concepts and component elements. These twelve included as means of containment of high temperature water (HTW) at high pressure the following:

- Prestressed cast iron vessels (PCIV)
- Prestressed concrete pressure vessels (PCPV)

- Steel tank pressure vessels
- Underground excavated cavities, steel lined, with high-temperature high-strength concrete for stress transfer between liner and rock
- Underground excavated cavities with free-standing steel tanks, surrounded by compressed air for stress transfer to the rock
- Underground aquifers of water-saturated sand and gravel confined by impermeable clay layers.

In addition to HTW as a storage medium the twelve selections included as low vapor pressure (LVP) media:

- High temperature oils
- Molten salts for their sensible heat
- Phase change materials (PCM) for their latent heat of melting, such as salt eutectics
- Rock or minerals as low cost media which require oil or molten salt as a heat transfer medium.

The containment of these LVP media included:

- Separate hot and cold tanks.
- Single tanks in which hot fluid (oil or salt) floats on top of cold fluid, and the boundary between them (thermocline) moves up and down with the storage discharging and charging cycle.
- Dual-media thermocline tanks in which packed rock-beds fill the tank and oil (or salt) fills the voids and is pumped as a heat transfer fluid.

A reference nuclear plant and both a large (800 MW) and a small (225 MW) coal-fired plant were considered as sources of thermal energy. Within them, various points can be the thermal energy source:

- High pressure (HP) turbine inlet steam
- Intermediate pressure (IP) turbine inlet steam
- Low pressure (LP) turbine inlet steam
- Intermediate steam extraction points and feedwater heater (FWH) outputs in the FWH system to raise condensate back to boiler inlet temperature.

Means of conversion of the stored energy to steam included:

- Flashing HTW to steam and lower temperature water by throttling the pressure, then passing steam through a peaking turbine (steam generation system)
- Using the HTW as boiler inlet feedwater, thus reducing the energy diverted for feedwater heating from the main turbine, increasing its output (feedwater storage system)
- With LVP storage media, using heat exchangers to transfer the energy to cold feedwater, producing either superheated steam, or hot feedwater.

FINAL SCREENING

For performance analysis of the twelve selections, computer programs were prepared to simulate the thermodynamics of the reference plants, modified to best interface with thermal energy storage systems (TESS), and of the charging and discharging cycles of TESS operation.

In a preliminary analysis it was found that use of IP turbine inlet steam had advantages over the other steam sources for both the coal-fired and the nuclear plant. As the nuclear plant has no HP turbine (pressures of 13 to 24 MPa or 2000-3500 psig) the analyses for the two plants were then comparable. For more reliable boiler and nuclear steam supply operation when used with TESS for load-following, the reheater section of the coal-fired boiler and the reheat heat exchanger for the LWR were deleted because of probable operational problems. Plants so modified were called baseline plants, Plant #1 for 800 MW HSC, Plant #2 for 1140 MW LWR. For comparison of the selections a cycle of eight hours of storage charging and six hours of storage discharging per day was used.

Consistent economic procedures were adopted for comparison. Direct costs (equipment plus installation) of the TESS components were derived from the references supplied by the proponents, consultants, and other industry sources. For cost information and methodology applicable to the conventional reference plants the EPRI *Technical Assessment Guide* (August 1977) and detailed cost estimate documents on PWR and high-sulfur coal (HSC) plants by United Engineers and Constructors for the Nuclear Regulatory Commission and for ERDA (DOE). To *direct* costs must be added overhead costs, cost of spare parts, contingency allowance, allowance for funds during construction (or interest during construction), consultant fees, site selection costs, etc. For the two reference plants the ratio between TOTAL costs, including all these adders, and direct costs is over 2.1. For comparability, all direct costs of TESS components were raised in the same proportion to give TOTAL costs, where the capitalization is used to indicate this specific meaning and not just the sum of component costs.

Also described in *Technical Assessment Guide* are the levelizing factors to express an escalating set of annual fuel costs over the life of a plant (eg 30 years) as a uniform set of fuel costs over the period that has the same present worth. Such a levelizing factor more than doubles the actual fuel cost in the first year of operation. Thus, the conservative costing, consistent with utility planning practice, gives capital costs and fuel costs roughly twice as great as used in some other studies not following these practices.

The cost of the TESS components, and the modifications they required to the baseline plant (eg peaking turbine, or enlarging main turbine and modifying feedwater heaters), were determined in terms of dollars per kilowatt electric of increased or peaking output ($/kW). The energy-related costs, C_E, and the power-related costs, C_P, were also separated to permit some extrapolation of design for longer or shorter discharge periods. The often used energy-related cost in $/kWh can be found approximately by dividing C_E by six hours.

Figure 8-1 summarizes the conclusions reached from comparing the twelve selections, and variants thereof tested as sensitivity analysis. The basis for choice was not only cost but technical risk (or near-term availability), ability to meet utility operating requirements, environmental soundness, conservation potential, and prospects for future improvement. Except for cost the comparative judgments are largely subjective.

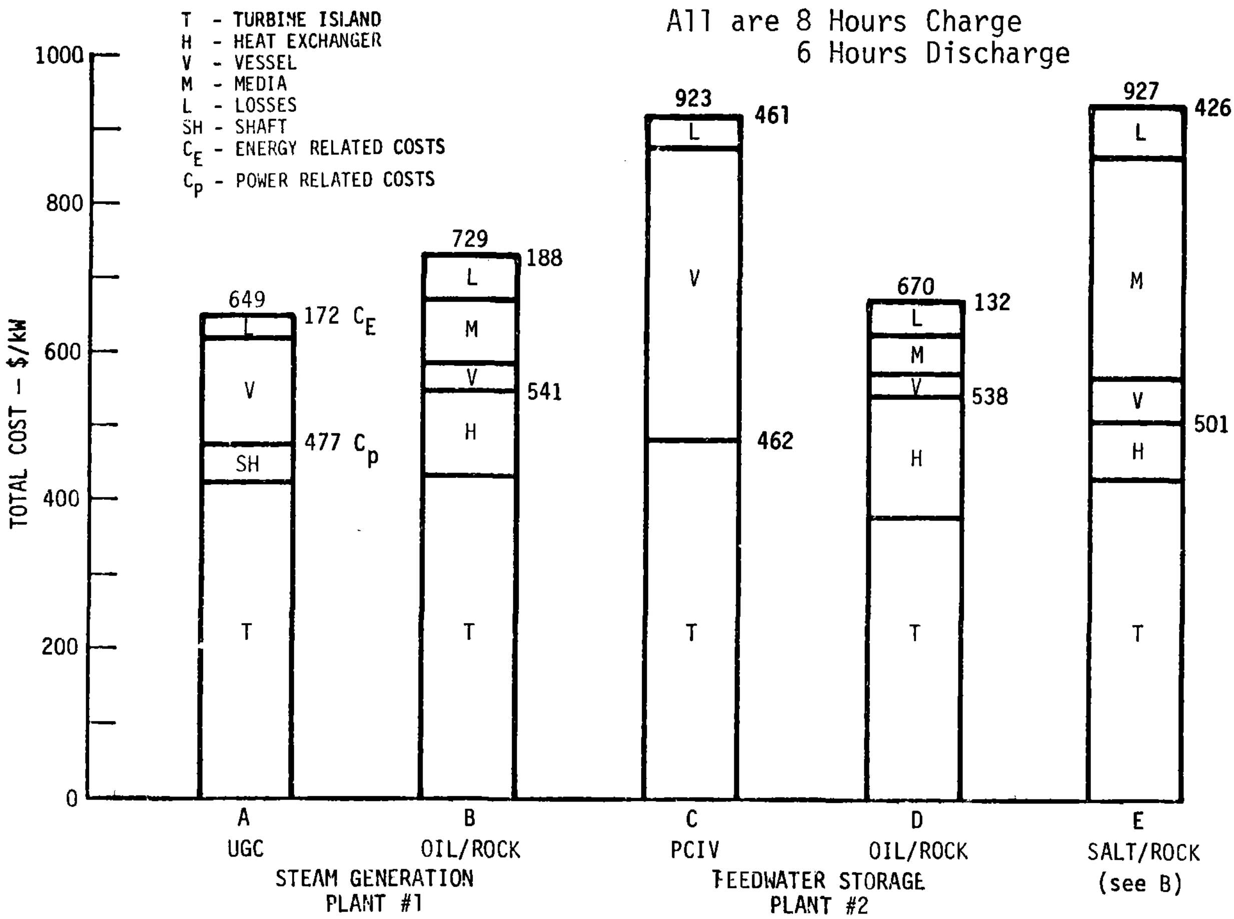

Figure 8-1. Summary of data on recommended choices for further study.

The five bars shown in Figure 8-1 represent the recommended choices and alternates presented to DOE/EPRI/NASA personnel on May 22, 1978 as the conclusion of this task. The left-hand scale of TOTAL cost in $/kW applies to all the horizontal divisions. At the top of each bar is the total cost of TESS in $/kW. At the right of each bar are the separate costs, C_E the energy-related cost at the top, and C_P the power-related cost at the bottom.

Each of these has major components, the relative size of which is of interest. The legend at the top identifies the space marked T as being the $/kW cost of the peaking Turbine Island: turbogenerator plus all related accounts allocable. Storage vessels costs are labeled V, heat exchangers H, and storage media M. In underground concepts at least part of the drilled shaft is considered a power-related cost and is labeled SH.

These bars all include L for the losses due to the turnaorund efficiency: the ratio of the extra electric energy out in a discharge cycle to the reduction in electric energy out during the charging of the TESS. The value of the losses per kilowatt of peaking has been capitalized, ie the present worth of the extra fuel and O&M from turn-around efficiency losses gives a cost component to make comparable TESS of different efficiencies. It was derived by comparing the cost of electricity in $/MWh (mills per kWh) of the TESS plus baseline plant with the defined reference plants operating with the same load cycle: 6 hours at peak, 8 hours at low load, and the balance at the *average* load. On this basis a rough comparison can be made between the cost of each TESS, and the $/kW cost of the reference plants which are 743 to 785 $/kW, the former for the coal-fired plant and the latter for the nuclear plant, all in 1976 dollars.

Power plants are also compared in terms of the cost of electricity (COE). Table 8-1 shows the choices A, B, C, and D and the levelized cost of electricity for combination baseline plant and TESS operating as defined in the Conclusions which follow. If the capital costs of the TESS including peaking Turbine Island or main Turbine Island modifications are used as though the plant were able to operate standing alone, an incremental COE can be found. The variable fuel costs are used of the plant(s) being dispatched or shut down during the storage charge hours, and may be nuclear, coal, or oil/steam plants depending on the load pattern and generation mix available to a utility. Such incremental costs are higher than the *dedicated* Baseline/TESS plant in which the baseline plant energy dilutes the COE measure.

The values of COE for the nuclear and coal base-load reference plants are shown for comparison. The availability or maximum capacity factor (CF) of each is assumed to be 0.723. If the reference plants load-follow with a peak power swing 15 percent greater than the average output, as in choices C and D, the CF is reduced to 0.605. If the peak

Table 8-1. Cost of electricity—alternative approaches.

Selection	Plant	COE in $/MWh Baseline /TESS	TESS Plant Alone Nuclear	Coal	Oil
A	Coal/Steam	54.90	96	99	172
B	Coal/Steam	55.50	106	109	190
C	LWR/FWS	47.30	127	130	199
D	LWR/FWS	46.40	111	103	183

Comparative Reference Plant Values:

	CF	Nuclear Plant #2	Coal Plant #1
All energy *available*	0.723	43.14	44.60
15% swing (as in C & D)	0.605	47.60	--
50% swing (as in A & B)	0.462	--	57.00

power swing is 50 percent, as in choices A and B, the CF is reduced to 0.462. It can be seen that the Baseline/TESS plant has a lower COE than its reference plant counterpart in a load-following mode.

CONCLUSIONS

Choice A is an underground cavity at a depth of about 300 m (1000 ft) to contain HTW at 4.65 MPa (675 psia), for the Plant #1, coal-fired 800 MW. Energy stored permits an additional 400 MW out for six peak hours. The underground cavity has a steel liner connected to the rock by a layer of high-temperature, high-strength concrete. For charging, steam is diverted from between the HP and IP turbine and condensed in the cooler HTW remaining in the cavity. During discharge the pressure is reduced so part of the HTW flashes to steam which goes to the surface, is throttled to 1.72 MPa (206 psia) and drives the peaking turbine.

This choice is most favorable economically at 649 $/kW, is believed to be well within the state-of-the-art in drilling, excavating, and lining, and is quite efficient. A disadvantage is that it is viable only in areas with suitable geology.

Choice B, also applied to Plant #1, 800 MW, and with the same peaking swing of 400 MW, uses Caloria HT 43, a high temperature heat transfer oil as both the heat transfer fluid and part of the dual-media storage. The other storage medium is packed beds of rock, such as riverbed gravel, in atmospheric pressure tanks. About 80 percent of the energy is stored in the low cost rock. Again IP turbine inlet steam is used, heat exchangers to oil transfer energy storage, and on discharge separate heat exchangers convert the stored energy to superheated steam to drive a peaking turbine.

This choice, 729 \$/kW, is somewhat higher than choice A but still below the comparative value, 743 \$/kW, for the reference plant. Some demonstration has been done, confirming technical feasibility, but long-term materials stability and compatibility are not yet proven. It is not geographically sensitive.

Choice C applies the PCIV as the storage containment for HTW. The cost of the PCIV per unit volume is higher than choices A and B. The lowest cost application is feedwater storage for the Plant #2, nuclear at 1140 MW_e. Output for this mode of conversion to electricity is limited to 180 MW. The total cost is 923 \$/kW, higher than A and B and than the reference cost of 785 \$/kW for nuclear base load capacity. It represents a choice that is above ground and not geographically sensitive and that confines storage to HTW of boiler feedwater quality so there is no possibility of contamination of feedwater by other media such as oil or salts.

Choice D is also a feedwater storage system applied to Plant #2. However, it uses the dual-media packed rockbeds and Caloria HT 43 for storage, as did choice B. It is lower in cost, 670 \$/kW, than B with the same storage media, and than C with the same turbine configuration. Like C it is limited to 180 MW peaking output.

Choice E is similar to choice B except the dual media are rock and a molten salt such as HITEC or PARTHERM 290, both trade names for a eutectic mixture of sodium and potassium nitrates and nitrites. E has a considerably higher cost, 927 $/kW, because of the current cost of the salt eutectic.

Conclusions presented on May 22, 1978 were that choices A and B were strongly recommended as warranting more detailed conceptual design and analysis in the next tasks. As an addition to these two, the preferred third choice was choice C, on the grounds of diversity. It would permit exploration of the feedwater storage mode of operation and storage of HTW in pressure vessels above ground. While the PCIV was selected, the major design considerations would be similar for the PCPV.

Choices D and E were offered as alternates rather than additions to the task goal of recommending three systems for further study. If primary emphasis were placed on economic viability, it is superior to C as a feedwater storage system. However, if it were selected instead of choice C, there would be two oil/rock dual media systems. In that case it was suggested that molten salt be considered despite its current high cost; that is replace B with E.

There are cost improvement directions that have been suggested for both oil and salt systems that could reduce the cost of the media or the quantity of heat transfer fluid needed. They are not considered near-term available but have future promise meriting research and development with both media.

Upon deliberation by the attending DOE, EPRI, and NASA personnel, approval was given to perform the continuing study on choices A, B, C, and D under DOE/NASA contract DEN3-12 and parallel EPRI contract RP1082-1. By the proviso that B and D be studied as dual-media systems, the alternatives of oil/rock and salt/rock were both retained.

SECTION 9
ADDENDA

LIST OF SYMBOLS

a — Annum (year)

A — Area (m^2) x 10.76 = (f^2)

A_e — Turbine exhaust area

c_p — Specific heat capacity (kJ/kg · °C) x 0.239 = (Btu/lb · °F)

C — Heat capacity rate (in heat exchanger design, $\dot{W} \times c_p$) (kJ/h) x 0.95 = (Btu/h)

C_h, C_c — Heat capacity rate of the hot and cold streams respectively

C_{min}, C_{max} — Heat capacity rate of the smallest and largest of hot and cold streams

C — Specific cost: (\$/kW) for power, (\$/kWh) for energy, (\$/m^2) for volume, etc

C_P — Cost of power-related TESS components (\$/kW)

C_{PP} — Cost of peaking turbine (\$/kW)

C_{PS} — Cost of power-related storage components (heat exchangers, pumps, etc) (\$/kW)

C_{HX} — Cost of heat exchangers (\$/kW)

C_E — Cost of energy-related TESS components (\$/kWh)

C_{ES} — Cost of storage media and containment (\$/kWh)

C_{TM} — Cost specifically of dual-media plus tankage for 6 hours discharge (\$/kW)

C_L — Cost representing capital equivalent of turnaround efficiency (\$/kWh)

C_T — Total cost of TESS components ($C_P + C_E \cdot H$) (\$/kW)

C_S — C_T less the term C_L

CF — Capacity factor (annual average power/rated power)

COE — Cost of electricity (\$/MWh) = (mills/kWh)

e_o — Specific output (from storage medium) ($kWh_{electric}/m^3$)

F — Load-following factor (fraction of *available* energy that is produced)

h — Specific enthalpy (kJ/kg) x 0.430 = (Btu/lb)

$\Delta h_{e_{sat}}$ — Leaving loss (turbines) correction for saturated vapor

Δh_e — Leaving loss correction for wet steam

H — Enthalpy (kJ) x 0.948 = Btu

ΔH — Hours of storage discharge capacity

L — Loss factor, defined as ($C_L \cdot H$) (\$/kW)

M_c — Mass flow ratio: (kg_{oil}/kg_{steam})

N_{tu} — Number of thermal units: a dimensionless ratio used in heat exchanger design

p — Power swing, a ratio of the *added* peaking power output to the normal or rated output of the plant without TESS

P — Power (MW)

P_n — The *normal* power level, ie power output when storage is neither charging or discharging, and by analogy output level in a load-following plant that is intermediate between the peaking and off-peak-hour level. Approximately, average power.

P_d — Plant power output during storage discharge hours

P_c — Plant power output during storage charging hours

P — Pressure-megapascals (MPa) x 145 = (psi)

P_{STOR} — The pressure in storage containment at end of charging cycle

P_{THR} — The throttle pressure of storage discharge steam that is admitted to the peaking turbine

R — Ratio (dimensionless)

R_C — Ratio of charge steam mass to original HTW mass

R_D — Ratio of discharge steam mass to original HTW mass

$R_\$$ — Ratio of cost per kg of other storage media to rock

R_ρ — Ratio of density of other storage media to rock

R_{cp} — Ratio of specific heat of other storage media to rock

t — Time - hours (h)

t_d — Time duration of storage discharge
t_c — Time duration of storage charge

T — Temperature (°C) x 1.8 + 32 = (°F)

U — Heat transfer coefficient ($W/m^2 \cdot °C$) x 1.75 = ($Btu/f^2 \cdot °F \cdot h$)

v_e — Specific volume of saturated vapor (m^3/kg) x 16.1 = (f^3/lb)

V_e — Exit velocity from turbine (m/s) x 3.28 = (f/s)

V — Volume (m^3) x 35.31 = (f^3)

V_s — Volume of storage medium

$\dot{W}$ — Mass flow rate (kg/h) x 2.2 = (lb/hr)

$\dot{W}_c$ — Charge steam flow rate
$\dot{W}_p$ — Discharge steam flow rate

x_e — Quality of steam (percentage as vapor)

x_r — Weight fraction of rock in dual-media storage

y_r — Volume fraction of rock in dual-media storage

<u>Greek Alphabet</u>

α — Temperature approach (in heat exchangers), minimum ΔT between input and output streams (°C) x 1.8 = (°F)

ε — Effectiveness (in heat exchangers). Measure of performance as fraction of the theoretical maximum heat transfer rated actually achieved.

η — Turnaround efficiency. Measure of the degradation and loss of thermal energy in the TESS processes. Ratio of peaking electric energy produced during discharge hours to the reduction in electric energy output during charge hours (for uniform boiler output rate).

ρ — Density (kg/m^3) x 0.0624 = (lb/f^3)
ρ_r — Density of rock in dual-media storage
ρ_f — Density of heat transfer fluid in dual-media storage

GLOSSARY

This list collects terms that which are defined at least once in the text but which are repeated a number of times in different sections. It concentrates on terms where a particular meaning is used in this report that may not be generally familiar. No attempt at defining generally familiar terms is intended.

Accumulator — A pressure vessel to contain high temperature water (HTW) for later conversion to steam (see variable pressure, expansion and displacement accumulators).

AFDC — Allowance for Funds During Construction (also called interest during construction). A component of TOTAL costs (q.v.).

Boiler Island — Those components and cost accounts of a power plant that produce the steam supplied to the Turbine Island (q.v.). The components include the fuel processing, the boiler (in nuclear plants called the Nuclear Steam Supply - NSS), fans, stacks, and stack gas processing.

<u>Capital Costs</u>

Direct Costs — The cost (in M$ usually) of purchased equipment plus on-site labor and materials costs needed for installation.

Base Costs — Direct costs plus on-site and home-office overhead costs.

TOTAL Costs — Base costs plus other capital investment allowances necessary for initial operation, including spare parts, contingency allowance, allowance for funds during construction, site selection and approval costs, etc.

Specific Capital Costs — Any of the above expressed per unit of power out, ie $/kW.

Levelized Annual Capital Costs — The present worth of the capital investment required as of the year of initial operation (ie the construction costs antedating said year are discounted forward to said year and any required periodic capital replacements are discounted back to said year) is multiplied by a fixed charge rate (FCR) to convert the capital costs to uniform annual amounts over the life of the plant (eg 30 years). Certain taxes and insurance annual payments that are capital related are usually included in the FCR, as is the allowable depreciation schedule, debt/equity ratio, investment credits, etc. The FCR must be compatible with the scenario of assumed future general inflation and discount rates.

Fixed Costs (Annual) — This includes the Levelized Annual Capital Costs plus Operation and Maintenance costs that are capital dependent rather than fuel dependent, ie are independent of the capacity factor of the plant. They are levelized, ie converted to uniform annual cost with the same present worth as the expected actual non-uniform costs.

Variable Costs

Levelized Fuel Costs (per MWh) — The cost of fuel in the year of initial operation in whatever thermal units are convenient is converted to \$/MWh (thermal) and multiplied by a levelizing factor which is a function of the expected fuel escalation rate over the life of the plant. Dividing by the plant efficiency gives a uniform annual fuel cost (\$/MWh electric) that has the same present worth as the escalating actual cost of fuel.

Levelized Annual Fuel Costs — These are the total annual costs incurred for fuel for the rated plant capacity in MW and the annual equivalent number of hours of rated power output.

Variable Annual Costs — To the Levelized Annual Fuel Costs the variable Operation and Maintenance costs (O&M) are added. These are O&M costs that are roughly proportional to annual hours of operation. These O&M costs are also levelized.

Total Annual Costs — The sum of annual fixed and variable costs.

Specific Annual Costs — All of the above annual costs may be specified per kW_e by dividing by the rated kW_e output.

Production Cost — The specific variable costs of a plant per MWh (electric), ie, fuel costs plus variable O&M. These are used for dispatching plants so are actual costs for the current year, not levelized.

Displacement Accumulator — A pressure vessel containing HTW, with no steam cushion. When fully charged it contains all hot water. During discharge HTW is removed from the top and an equal volume of cold water enters the bottom. The thermocline (q.v.) rises until when fully discharged it contains only cold water. Charging reverses the flow.

Expansion Accumulator — A pressure vessel almost completely full of HTW at high pressure when fully charged. The mode of discharge is to extract HTW from the bottom, allowing a small amount of the remaining HTW to flash to steam and fill the volume.

FWH - Feedwater Heaters — Steam condensed in the condenser is returned as water to the boiler inlet by passing through a train of feedwater heaters. Each is fed by steam from spaced extraction points in the turbine train, so that the temperature of feedwater is raised in steps, for efficiency, to the desired boiler inlet temperature.

FWS - Feedwater Storage — One mode of use of TESS is to extract excess steam to heat excess feedwater during off-peak hours. This is stored and during peak hours permits less steam extraction hence more power output from the turbines.

LVP - Low Vapor Pressure storage media — A generic term for those media, liquid or solid, that do not require pressurized containment.

PCIV - Prestressed Cast Iron Vessel — A form of pressure containment comprising cast iron blocks that can be assembled into rings and stacked for the desired height. A cylindrical steel liner contains the pressurized fluid. Steel cables around the exterior, and exterior tendons connecting to top and bottom end caps, ensure that all parts of the cast iron structure are always in compression.

PCPV - Prestressed Concrete Pressure Vessel — Similar in principle to PCIV. A steel liner is surrounded with a layer of high-temperature, high-strength concrete, then the required additional thickness of reinforced concrete is built around the core. Reinforcing bars and cables within or exterior to the concrete keep it in compression.

<u>Plant</u>

An electricity generating unit, from fuel input to electricity output.

Reference Plant — One of the described current technology base-load plants.

Baseline Plant — A reference plant modified to better interface with a TESS.

Baseline/TESS Plant — The combination of a baseline plant and a thermal energy storage system.

Plant #1 — 800 MW high sulfur coal plant.

Plant #2 — 1140 MW Light Water Reactor plant (PWR).

Base-Load Plant — A plant with low variable costs, designed to operate at rated load for as many hours per year as it is available (6000-8000).

Intermediate Plant — Plants with higher production costs than base-load plants and generally operated for fewer hours per year (2000-6000).

Peaking Plant — Plants specifically designed for supplying capacity during peak hours of peak days (<2000 hours per year). Low capital cost is emphasized over low production cost.

Load-Following Plant — A plant that varies its output in a pattern similar to the utility load variations. (N.B. — Usually intermediate plants but can be base-load plants. Some follow load in detail, with elaborate controls; others operate principally at full load or no load, as dispatched. The latter are often called cycling plants.)

Specific Output — A measure of performance of a TESS configuration that gives the number of kWh (electric) produced during discharge per cubic meter of storage media.

Thermocline — A steep vertical temperature gradient between hot and cold fluids (or dual-media). By control of convection, the hot fluid floats on the cold fluid without much mixing.

Turbine Island — That part of a power plant that encompasses the turbo-generator, the electrical equipment, and associated cost accounts. The interface with the Boiler Island is the live steam inlet pipes; the output of the Turbine Island is electricity to the network.

TES — Thermal Energy Storage (in fluids such as high temperature water (HTW), oil, or molten salts and/or in solids such as rock).

TESS — Thermal Energy Storage System. The aggregation of components for thermal energy storage including the storage media, the containment, heat exchangers and pipes for energy conversion and transport, and the peaking Turbine Island to convert the stored energy to electricity. Where necessary to speak of TESS less the peaking turbine it may be called Thermal Energy Storage Subsystem.

Turnaround Efficiency — The measure of losses of energy and availability during a charge/discharge storage cycle. It is the ratio of the peaking electric energy produced during the discharge cycle to the reduction in electric energy production during the charging cycle.

Variable Pressure Accumulator — A pressure vessel containing HTW except for a small steam cushion at the top. When fully charged the steam cushion is smallest and the temperature and pressure are at their maximum. During discharge, steam is withdrawn causing some of the HTW to flash to steam to fill the steam cushion volume. The pressure, temperature, and HTW level continue to drop during discharge until withdrawal is stopped.

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16. Abstract

Over forty thermal energy storage (TES) concepts gathered from the literature and personal contacts were studied for their suitability for the electric utility application of storing energy off-peak for discharge during peak hours. Twelve selections were derived from the concepts for screening; they used as storage media high temperature water (HTW), hot oil, molten salts, and packed beds of solids such as rock. HTW required pressure containment by prestressed cast-iron or concrete vessels, or lined underground cavities. Both steam generation from storage and feedwater heating from storage were studied. Four choices were made for further study during the project.

Economic comparison by electric utility standard cost practices, and near-term availability (low technical risk) were principal criteria but suitability for utility use, conservation potential, and environmental hazards were considered.

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