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Preliminary Requirements for Thermal Storage Subsystems in Solar Thermal Applications

R. J. Copeland

SERI

Solar Energy Research Institute
A Division of Midwest Research Institute

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PRELIMINARY REQUIREMENTS FOR THERMAL STORAGE SUBSYSTEMS IN SOLAR THERMAL APPLICATIONS

R. J. COPELAND

APRIL 1980

PREPARED UNDER TASK NO. 3528.20

Solar Energy Research Institute

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FOREWORD

This report is a partial documentation on SERI Task 3528, Thermal Storage Requirements, and presents value data for thermal storage in solar thermal electric systems and a method of ranking thermal storage concepts. This effort is being conducted in support of a plan to develop thermal storage technologies for solar thermal applications. The task is joint funded by the Thermal Storage Program and the Advanced Technology Element of the Solar Thermal Program at the Department of Energy.

The report's data were generated by several people. The relative value data were generated by Jim Green. Nancy "Therm" Burnham contributed the value analysis for long-term storage and the data for the ratio of energy delivered from storage to that delivered directly. John Kowalik contributed the data on collector efficiencies. Conversion efficiencies were provided by Harold C. Welz, Project Supervisor at Stearns-Roger. Roger Taylor suggested the method of adjusting the capital cost in determining capital value. Jim Calogeras of NASA-Lewis contributed the cost of the reference thermal storage concepts.

Michael Karpuk was a prime contributor. He helped select the reference systems and provided the cost data on them. He also helped monitor the Stearns-Roger subcontract. Karpuk reviewed the ranking methodology and made several comments that led to improvements.

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SUMMARY

Thermal storage is a very important subsystem in a solar thermal system. With thermal storage, a solar thermal system can operate continuously during periods of variable insolation or operate during nonsolar hours. To facilitate the accelerated development of thermal storage technologies matched to solar thermal system requirements, a comprehensive program plan has been prepared (U.S. DOE 1979). The plan was prepared at the joint request of the DOE Division of Central Solar Technology (CST) and Energy Storage Systems (STOR). This report presents preliminary requirements for thermal storage subsystems in support of the implementation of the joint plan.

The implementation of the CST-STOR plan requires development of thermal storage technologies that meet the following criteria:

- When mature, the cost of the thermal storage must be less than or equal to its value.
- The developed technologies are more cost effective than alternative thermal storage technologies.

Value is a measure of the worth of the thermal storage technologies; i.e., what the user will pay as measured by the cost of conventional fuel systems. These value data are employed to establish program cost goals. Preliminary data on thermal storage value for buffer, diurnal, and long-term storage for solar thermal bulk electric power applications are presented below:

RECOMMENDED PRELIMINARY COST GOALS FOR THERMAL STORAGE IN SOLAR THERMAL ELECTRIC PLANTS
(1976 $/kW$_e$)

Quantity of Storage	High Insolation	Medium Insolation	Low Insolation
Buffer	established by obtainable cost		
Diurnal			
3 Hours	255	120	60
6 Hours	300	180	90
9 Hours	—	225	110
Long Duration			
62% Efficiency	230	210	180
100% Efficiency	695	620	515

The obtainable mature system costs of any candidate thermal storage concept must be less than or equal to the appropriate cost goal for its intended application. Several thermal storage technologies are anticipated to meet one or more of the above cost

goals. The program does not have sufficient funds to develop all such technologies; and a standard method of determining the most promising cost-effective technologies is needed.

The determination of cost-effective thermal storage concepts can only be accomplished by comparing the alternatives. A ranking methodology has been prepared to do such comparisons. The methodology has two versions: Simplified and Computer. The Simplified version is to conduct preliminary screenings; the Computer version is for conducting comparisons over a wide range of system parameters. All data necessary to use the Simplified version are included. An exemplary case is presented to illustrate the use of the methodology.

The ranking methodology compares thermal storage concepts on the basis of unit energy costs of the storage-coupled solar thermal system (i.e., bus bar energy costs). The ranking methodology has been compared with absolute calculations of bus bar energy costs. There are differences, but identical conclusions are reached with both approaches.

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SECTION 1.0

INTRODUCTION

Major considerations impacting the development of solar thermal power systems* for commercial applications are the need to provide continuous operation during periods of variable insolation, to extend operating periods into nonsolar hours, to buffer potentially harmful transients induced into systems by abrupt insolation changes, and to assure the availability of productive capacity in emergency periods. Two options exist for meeting these requirements: conventional backup systems and thermal storage.** Backup systems provide a viable near-term solution; however, as conventional fuel supply becomes critically limited, due to cost or availability, thermal storage will assume an increasingly important role.

To facilitate the accelerated development of thermal energy storage technologies matched to solar thermal system requirements and scheduled milestones, a comprehensive program has been drafted. The plan (U.S. DOE 1979) for this program was prepared at the joint request of the DOE Divisions of Central Solar Technology (CST) and Energy Storage Systems (STOR). The basic strategy of the program is both aggressive and flexible. Reflecting the current direction of the Thermal Power Systems (TPS) Branch, CST, storage for repowering/industrial retrofit, total energy, and small community system applications will be stressed in the early years.

1.1 THERMAL STORAGE TECHNOLOGY DEVELOPMENT PLAN

1.1.1 Objective of the Program Plan

The development goals of the program are to provide:

- second-generation storage subsystems, offering cost/performance improvements over the first-generation storage subsystems currently being developed for solar thermal power applications;
- first-generation storage subsystems for those solar thermal applications that presently have no storage subsystems; and

---

*Solar thermal power systems collect and concentrate the sun's radiant energy to heat a working fluid, i.e., convert the radiant energy to thermal energy. The thermal energy can be used directly for process heat applications or to drive a heat engine, producing mechanical and/or electrical energy. Applications for the latter include, but are not limited to, electric utility power plants, irrigation pumping systems, and total energy systems (cogeneration).

**Backup systems include utility grids, fossil-fueled systems, batteries, pumped hydro, etc. Thermal storage includes sensible heat, latent heat, and thermochemical concepts.

- a technology base to support storage subsystem development for future solar thermal power applications.

1.1.2 Program Elements

Seven elements have been defined in the storage development program: six of the elements are keyed to storage development for specific collector/receiver technologies; the seventh element is advanced storage technologies. These elements are:

1. Storage for water/steam-cooled collector/receiver
2. Storage for molten salt-cooled sensible heat collector/receiver
3. Storage for liquid metal-cooled sensible heat collector/receiver
4. Storage for gas-cooled sensible heat collector/receiver
5. Storage for organic fluid-cooled sensible heat collector/receiver
6. Storage for liquid metal/salt-cooled latent heat collector/receiver
7. Advanced storage technologies

Project applications* for the first six elements have been identified to provide a development focus for the storage technology development.

1.1.3 Role of SERI Systems Analysis

SERI is supporting the joint CST-STOR program plan with Systems Analysis. This activity includes both value analysis and comparisons of thermal storage technologies.

The value of thermal storage in a solar thermal system/application is a measure of its worth, or benefit, to the user. This benefit is measured by the cost of conventional fuel and equipment that is saved by the use of the thermal storage. Clearly, if the cost of a thermal storage system exceeds its value, a user would be expected to avoid the thermal storage. Program cost goals are always set less than that value. This procedure assures that only those technologies that have the potential of meeting (or surpassing) the cost goals will be developed, and furthermore, there will be a market for those technologies when developed.

Several thermal storage technologies are expected to meet the value-derived cost goals. Program resources are limited, and only a few of those that are possible can be developed. Obviously, these should be only the most promising technologies. SERI is supporting the selection process; it will review data being generated by the advocates of

*The repowering/industrial retrofit program may result in two system applications: repowering of an existing electric power generating plant and retrofitting of an existing industrial process heat plant. Storage requirements, which may differ significantly for the two applications, will be further defined pending completion of conceptual design studies in FY80.

each technology and then will compare the technologies on an equal basis. These comparisons are conducted in accordance with the program elements, identifying thermal storage technologies appropriate to each of the solar thermal systems.

1.2 OBJECTIVES OF THIS REPORT

This report presents partial documentation of SERI's Systems Analysis effort. Preliminary value data of thermal storage in electric power applications of solar thermal are included in support of defining cost goals. A standard method of ranking thermal storage concepts is also presented to assist in the selection of promising thermal storage concepts (i.e., technologies).

Section 2.0 presents general requirements for thermal storage. A rigorous definition of value is given as well as a rigorous mathematical expression. Requirements for a fair comparison of thermal storage concepts are also described.

Section 3.0 presents value data for thermal storage. First, the approach and value terms are defined. Methods of specifying backup are delineated, as is how value is determined with the various alternatives. Value data for buffer, diurnal, and long-duration storage are then presented.

Section 4.0 presents a methodology for ranking thermal storage in solar thermal system applications. The conditions are defined. An equation, called the ranking function, is derived. This equation calculates the Ranking Index (RI), which is the ratio of the energy costs with two different thermal storage concepts. Data for the evaluation of this Ranking Index are included. An example case is presented to illustrate the process.

Section 5.0 presents conclusions and recommendations.

SERI

SECTION 2.0

GENERAL REQUIREMENTS

The overall objective of the Thermal Power Systems Program is to establish the technical readiness of cost-competitive solar thermal power systems (U.S. DOE 1978). Thermal storage used in those solar thermal systems also must be cost competitive with other technologies that can perform the same mission. These alternatives include both conventional fuels (oil, gas, and coal) and other storage technologies (pumped hydro, batteries, mechanical storage, etc.). Clearly, the thermal storage-coupled solar thermal system must cost less than (or be equal in price to) available alternative systems. The lowest priced alternative defines a quantity known as the "value." Available alternative systems are all of those that have an assured fuel (or energy) supply and can meet the environmental restrictions for the user's location and application. For thermal storage in a solar thermal system, value* is defined as follows: that contribution to the solar thermal system value that is due to the presence of thermal storage.

For an incremental change in thermal storage quantity, this statement may be expressed mathematically as the following partial derivative:

$$\begin{array}{l}\text{THERMAL}\\ \text{STORAGE}\\ \text{INCREMENTAL VALUE}\end{array} \equiv \frac{\partial(\text{SYSTEM VALUE})}{\partial(\text{THERMAL STORAGE})}$$

where all other factors are held constant during the differentiation.

2.1 COST GOALS

Thermal storage technologies generally are not application specific. For example, a latent heat technology could be employed commercially in an electric power plant, a process heat system, or a total energy application. The cost of a thermal storage technology will not be greatly different for each of the various applications. Recognizing that fact, it is convenient to establish cost goals for thermal storage.

Cost goals serve two main functions in the CST-STOR plan: (1) to assure technologies are developed that will have a market; (2) to screen concepts, thereby eliminating those that are unpromising.

2.2 COMPARISON OF THERMAL STORAGE CONCEPTS

Virtually all thermal storage concepts can be designed for use in a solar thermal

*There are many factors contributing to the selection of an energy system; many of these factors are not amenable to economic analysis. This study addresses only those factors that can be directly analyzed.

system. One concept may provide low-cost storage but may impose efficiency penalties (low temperature or high losses). Another concept may be expensive but offer high efficiency. Depending upon the solar thermal system and the application, either one or both of the above types might be acceptable. To determine which concepts should be developed, quantitative criteria are needed that relate the cost and efficiency. Not only must these factors be addressed, but the impacts on the system must be considered. For example, one method of offsetting a low efficiency in a thermal storage concept is to increase the solar collector area. However, a large collector area is also beneficial to a high efficiency thermal storage concept, and such an increase may be even more advantageous (or less). To assure a fair comparison of concepts, each concept must be evaluated over the expected range of all system parameters.

At decision points in the program, several concepts will be under simultaneous development for the same application(s). Sufficient data may not be available to evaluate all parameters (e.g., development cost), but the consideration of a few might show that some of the concepts are clearly more attractive than others. The decision process in the thermal storage program is anticipated to progressively consider the parameters beginning with the most important. The overall process is described below:

1. Comparison to Cost Goals

   Following the establishment of the feasibility of a concept, the obtainable thermal storage costs anticipated for a mature technology are compared to program cost goals. Only those concepts that can potentially meet one or more cost goals are considered.

2. Ranking of Concepts

   Following laboratory experiments, design data are generated. The cost and performance are evaluated, with one or more solar thermal systems for each thermal storage concept. Those concepts are then compared, and the most promising are continued.

3. Selection for LSE

   Following tests of a full-scale or subscale thermal storage concept, the design data are updated. System cost, performance, and other factors are evaluated, and one concept for each program element is selected for technology verification in a solar thermal Large Scale Experiment (LSE). Obviously, this concept also must have the potential of meeting one or more cost goals.

For each of these selections, a quantitative method for relating the importance of the various system parameters is needed. A ranking methodology for that purpose has been derived and is presented in Section 4.0.

A summary of important parameters in the selection of thermal storage concepts for development is given below. Three classes are identified: performance, cost, and program. The performance factors affect the quantity of energy that is delivered by the system. The cost factors affect the cost of equipment. All factors must be considered when narrowing the selection to only one concept.

Important parameters in ranking concepts are:

- Performance factors: collector field size; plant nameplate rating; storage capacity; dispatch strategy; solar thermal plant location; and efficiency (1st Law, 2nd Law, and receiver).
- Program factors: environmental impact; availability of material; safety; time frame for commercialization; usability in several solar thermal systems; program resources (cost and risk of development); and priority areas.
- Cost factors: power- and energy-related storage costs; collector field cost; balance of plant cost; O&M levelized cost; and fuel cost (in hybrid system).

SECTION 3.0

VALUE OF THERMAL STORAGE

This section presents data for the value of thermal storage in solar thermal electric system applications. Section 3.1 presents the approach to the calculation of value. Section 3.2 presents data for buffer storage in a repowered electric power plant; diurnal storage in new electric power plants; and long-duration storage in base load electric power plants. Section 3.3 discusses factors that must be considered when using cost-goal and obtainable-cost data.

3.1 APPROACH TO THE CALCULATION OF VALUE

The value of thermal storage can only be determined from an analysis of the solar thermal system value. First, a mission must be defined for the storage-coupled thermal system. Next, the alternatives for performing the same mission with conventional energy (oil, gas, coal, nuclear, etc.) must be identified. The cost of performing the specified mission by the alternatives is then determined. The lowest cost option available to the user expresses the value of the solar thermal system. Thus, if the solar thermal system had a higher cost, most of the users would be expected to buy the alternative. Conversely, if the solar thermal cost were lower than the value, it would be the system of choice. Thus, value expresses the worth of solar thermal systems. Frequently the term "benefit" is used synonymously for value, and many studies have calculated the cost/benefit ratio of solar thermal systems.

The presence of thermal storage allows solar thermal systems the flexibility to perform many different missions. This capability is generally expressed as a function of capacity factor. With small or no thermal storage present in a solar thermal plant, it can operate only during the daytime. As a result, the capacity factor is limited to approximately 0.3, regardless of the collector area (Iannucci 1978). By adding thermal storage, the plant can be configured to operate continuously, obtaining capacity factors of 0.7 or higher. Thus, the mission of a solar thermal system can be influenced by thermal storage to anywhere from daytime-only operation to a baseload use.

The costs of the alternative energy systems are affected by the mission. This affect is illustrated in Fig. 3-1. The chart presents the same data in two ways: (I) levelized annual cost; and (II) levelized bus bar energy costs (BBEC). The letters A, B, C, D, and E are representative of different types of electrical power generation plants. The data include the capital, fuel, and operating and maintenance (O&M) costs and are somewhat representative for conventional gas turbines, advanced combustion turbines, combined cycles, coal, and nuclear power plants. The data are representative only and therefore do not necessarily present real costs for these plants. The data do show a real trend in the cost of alternative energy: as the capacity factor changes, so does the alternative system of choice. Furthermore, the unit cost of energy is not constant. The determination of value must therefore consider this variation in alternative energy supply for different missions.

The intermittence of the solar resource further complicates the determination of solar thermal system value. Unless the quantity of storage is very large, there will be several consecutive days in which the insolation will be inadequate, and the solar thermal system will not be available. Since users can be expected to need energy during those periods,

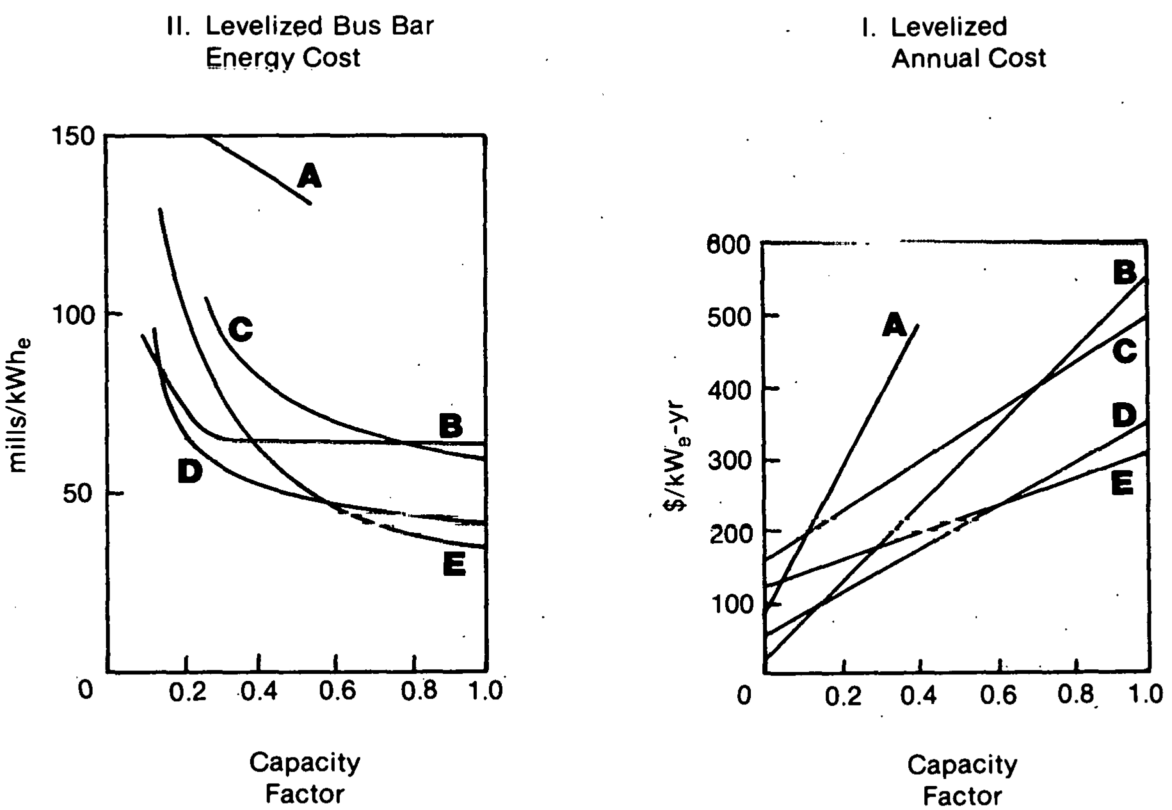


Figure 3-1. Example Screening Curves (EPRI 1978)

some form of backup generation is generally necessary. Several methods of providing the backup have been proposed for solar thermal systems; the following is a partial list:

- Hybrid Capability. Solar thermal and oil, gas, or coal fuel sources share a common energy conversion system. This type may be configured in two ways, as follows:
  - 100% rated, where the conventional unit is sufficient to supply the plant rating continuously.
  - Partially rated, where the fossil-fueled heat source is sufficient to supply the plant rating only with the use of thermal storage, but can do so every day even without a solar thermal input.
- 100% Backup. This type utilizes the solar thermal system only as a fuel saver, and the conventional plant is physically separate from the solar thermal system.
- Remix of Generating Plants. This type is applicable only to large electric utilities. This approach effectively provides backup by an array of different types of generating units in the electric grid. The overall system reliability requirement is thus met, and the total generation capacity and capital cost requirements are less than 100% backup.

The type of backup must be specified when defining the solar thermal mission. For example, if 100% backup is specified, then the value of the storage-coupled solar thermal plant is only the fuel and operations and maintenance (O&M) savings. If the plant is to be a new hybrid, then its value is the cost of the capital, fuel, and O&M of the alternative. If a remix strategy is followed, value is determined as the difference in total capital, fuel, and O&M of the whole utility without solar thermal less the total with solar thermal. Clearly, since the value of the solar thermal system is affected by the choice of backup, the value of thermal storage is also affected. The following paragraphs discuss the approaches to the analysis of thermal storage value for this study.

3.1.1 Remix Analysis Method

This method employs computer simulation models of both a solar thermal system and whole utilities. The solar thermal system model includes hour-by-hour insolation data and simulations for the performance of the electric power plant. The routine calculates the plant annual performance as a function of the collector area and quantity of thermal storage. The model (Day 1978) also calculates the optimum mix of conventional power plants if solar thermal plants are or are not present. In both cases, the reliability (i.e., loss of load probability) is equal. The value of the solar thermal plant is calculated as the difference in the costs of the conventional plants in the two cases. This calculation is expressed as follows:

VALUE OF SOLAR THERMAL SYSTEM	=	COST OF AN ALL CONVENTIONAL PLANT UTILITY	-	COST OF ONLY THE CONVENTIONAL GENERATION PLANTS IN THE REMIX WITH SOLAR THERMAL.

Note that the cost of the solar thermal plant does not appear in the value calculation.

Value data (Melton 1978; Westinghouse 1978) from computer models have been generated in support of the Solar Thermal Program. The data include solar thermal system value as a function of collector area, thermal storage quantity, and location. In this study, those existing value data are analyzed to calculate the value of thermal storage. As previously noted, the value of thermal storage is the partial derivative of the solar thermal value, with all other factors remaining constant. The equation is as follows:

$$\begin{array}{l}\text{THERMAL}\\ \text{STORAGE}\\ \text{INCREMENTAL VALUE}\end{array} = \frac{\partial(\text{SYSTEM VALUE})}{\partial(\text{THERMAL STORAGE})}$$

This equation will provide value, with the units of $\$/kWh_e$. In this form, the incremental value expresses the slope of the system value line and thus reflects the value of thermal storage with small changes in storage capacity. A more useful quantity is the total value of thermal storage and is calculated as follows:

$$\left.\begin{array}{l}\text{TOTAL THERMAL}\\ \text{STORAGE}\\ \text{VALUE}\end{array}\right|_h = \int_0^h \frac{\partial\begin{pmatrix}\text{SYSTEM}\\ \text{VALUE}\end{pmatrix}}{\partial\begin{pmatrix}\text{THERMAL}\\ \text{STORAGE}\end{pmatrix}} \, d\begin{pmatrix}\text{THERMAL}\\ \text{STORAGE}\end{pmatrix}$$

$$= \left.\begin{array}{l}\text{SYSTEM}\\ \text{VALUE}\end{array}\right|_h - \left.\begin{array}{l}\text{SYSTEM}\\ \text{VALUE}\end{array}\right|_{h=0} ,$$

where h is the quantity of thermal storage, usually measured in hours at the nameplate rating of the plant. The value is expressed in \$/kW and is literally a comparison of a solar thermal plant with storage to a solar thermal plant without storage.

The total thermal storage value can easily be compared to the thermal storage cost taken as the sum of the power-related costs and the capacity-related costs. The power cost corresponds to the cost of providing the capacity to accept and deliver thermal energy at given rates. The capacity costs reflect the cost of the maximum energy stored. The total cost of a storage subsystem capable of containing h hours at the system nameplate rating is as follows:

$$C_T = C_P + C_S \times h ,$$

where

C_T = total storage subsystem cost (\$/kW)

C_P = power-related cost (\$/kW)

C_S = capacity-related cost (\$/kWh).

The cost may also be expressed as an average cost per unit of storage capacity as follows:

$$\frac{C_T}{h} = \frac{C_P}{h} + C_S \ .$$

Cost data in both forms are being used (U.S. DOE 1979), and total value may be similarly expressed for a direct comparison.

3.1.2 Next-Plant Analysis Method

For many solar thermal applications, a remix analysis method is either not appropriate or sufficient data or resources do not exist for the more complex analysis. For example, for a total energy system, only one plant is under consideration and a remix is not possible. In other cases, a new application of thermal storage is being proposed (e.g., long-duration storage and transport using a thermochemical storage technology for electric power generation). For such new uses, a calculation of value is needed to determine if the proposed system has promise. For this level, a "Next-Plant Analysis Method" is employed. If sufficiently promising, a more detailed calculation would be performed with a remix analysis method.

The next-plant analysis method takes the point of view of a decision maker, considering the purchase of an energy system for the "next plant." The long-range planning has been conducted and a decision on a certain mission for the energy system has been made: for example, a mission that provides a plant capacity factor of 0.4, with a solar thermal plant. The alternatives are costed following the daily-output and annual-capacity factor desired. In this method, several missions (i.e., capacity factors) are defined. Solar thermal system value as a function of system parameters is obtained by systematically varying those parameters.

A backup energy supply must be specified as part of the solar thermal system. For almost all users, a reliable energy supply is mandatory (the economic consequences of frequent, unscheduled plant shutdowns are generally very great). For large utility applications, a nonhybrid solar thermal plant carries a capacity credit less than its nameplate rating. This fact complicates the analysis in that some method must be devised to account for the capacity credit difference of a nonhybrid solar thermal electric power plant from a conventionally fired plant. One approach is to force the solar thermal plant to be a hybrid or to require 100% back-up (no capital credit). Another approach is to adjust the capital cost of the alternative. In this latter case, a factor (DF) is needed. The use of that factor is illustrated below:

$$\begin{aligned}
\text{VALUE} &= \text{COST OF ALTERNATIVE} \\
&= \begin{matrix}\text{ADJUSTED}\\ \text{CAPITAL}\\ \text{COST}\end{matrix} + \begin{matrix}\text{FUEL}\\ \text{COST}\end{matrix} + \begin{matrix}\text{O\&M}\\ \text{COST}\end{matrix} \\
&= \text{DF} \times \begin{matrix}\text{CAPITAL}\\ \text{COST}\end{matrix} + \begin{matrix}\text{FUEL}\\ \text{COST}\end{matrix} + \begin{matrix}\text{O\&M}\\ \text{COST}\end{matrix}
\end{aligned}$$

The factor DF is a number less than one and accounts for backup generation equipment. Setting the factor equal to the ratio of the solar thermal capacity factor divided by the conventional plant reliability has been suggested by Roger Taylor of SERI:

$$DF = \frac{CF}{R_C} = \frac{(AF)(R_{ST})}{R_C} ,$$

where CF is the solar thermal capacity factor,

R_C is the conventional plant reliability (the fraction of the time that a plant is available to operate),

R_{ST} is the solar thermal plant reliability, and

AF is the fraction of the year that a solar thermal plant could operate if there were no equipment outages for scheduled and unscheduled maintenance.

An example:

$$AF \doteq 0.45$$

$$R_{ST} = 0.90 \quad (CF = 0.405)$$

$$R_C = 0.85$$

$$DF = \frac{(0.45)}{(0.85)} (0.9) = 0.476$$

If the mission is a 100-MW_e plant with a 0.405 capacity factor, the solar thermal plant would have a value equal to the fuel and O&M cost of a 100-MW_e, 0.405-capacity factor [$3.55(10^8)$ kWh_e)/year] plant plus the capital cost of a 47.6-MW_e plant delivering 3.55 $(10)^8$ kWh_e/year. The discount factor approach is clearly very approximate and should only be used when time and/or resources do not allow the analysis by the remix analysis method.

Once backup generation costs have been analyzed, the solar thermal plant value is determined for various missions calculating the costs of all of the probable alternative systems. Once the solar thermal plant value is determined, the thermal storage value is calculated as in the remix analysis method.

3.1.3 Thermal Storage Value Terms

Three terms are employed to express the value of thermal storage:

1. Thermal storage incremental value ($/kWh)
2. Total thermal storage value ($/kW)
3. Average value of thermal storage ($/kWh)

Figure 3-2 illustrates the three terms and the method of evaluation at point X, a certain quantity of storage. The graph plots the total solar thermal system value (for a constant collector area, but variable storage capacity) as determined from its fuel value, O&M value, and capital credit. At point X, the slope of the solar thermal system value is the thermal storage incremental value determined by the partial derivatives at point X. The total thermal storage value is the difference in solar thermal system value with "X" hours of storage less the solar thermal system value with no storage. The average thermal storage value is the total thermal storage value divided by the quantity of storage, "X"; the average value is the slope of the straight line from the solar thermal system value with no storage to the solar thermal system value with "X" storage capacity. All three terms are a function of the quantity of storage. Furthermore, since the solar thermal system value is a function of the collector area, location, application, and market penetration, all three thermal storage value terms are also functions of those items.

3.2 THERMAL STORAGE VALUE IN LARGE UTILITY APPLICATIONS OF SOLAR THERMAL SYSTEMS

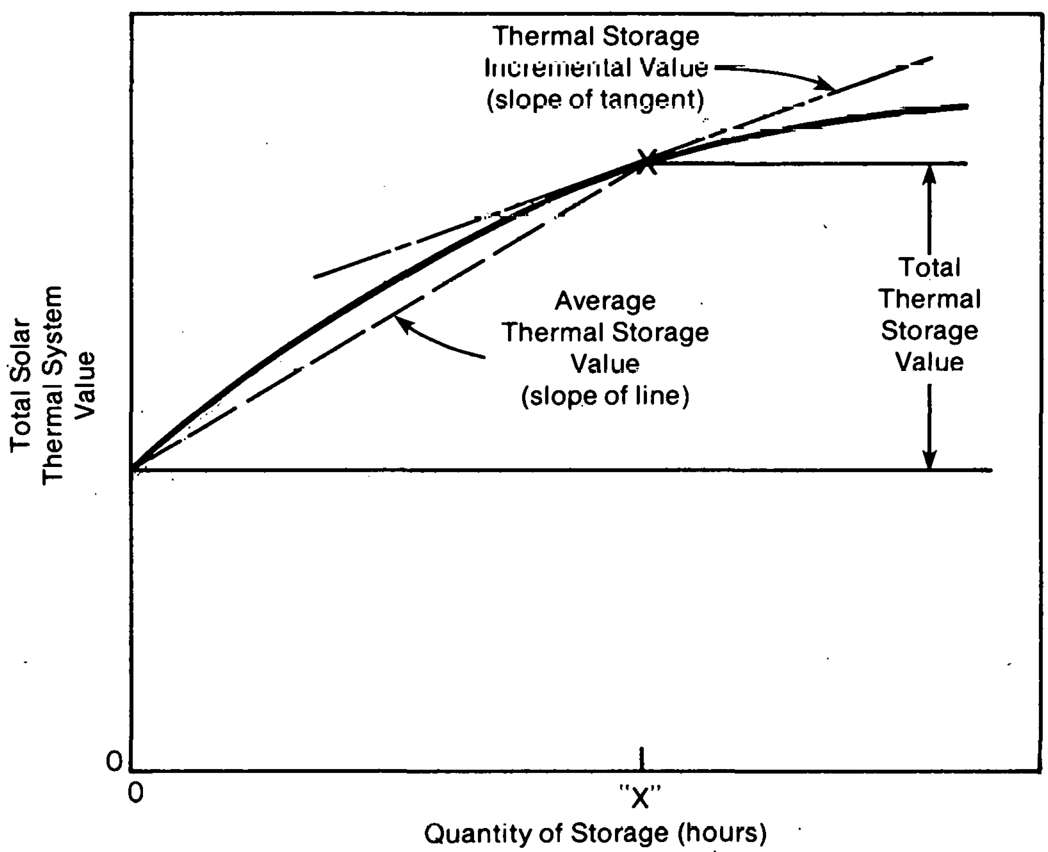
3.2.1 Value of Buffer Storage

Buffering thermal storage is generally required in a solar thermal system. The buffering protects equipment from rapid thermal cycling. In stand-alone solar thermal plants, the buffering is generally provided by the diurnal storage. In hybrid plants, the buffering may be provided either by thermal storage or by burning a fossil fuel. When the latter path is chosen, the fossil fuel must be burned even when insolation is available, to provide buffering; and fuel may also be burned whenever the plant capacity is needed. If there is buffer thermal storage available, the fossil boiler may be left cold most of the time. With buffer thermal storage, the fossil boiler is still used whenever the plant capacity is needed, but otherwise the combustion of fossil fuel is minimized.

A comparison of these two methods of providing buffering has been conducted by Westinghouse (Day 1979) for SERI. The study was conducted for a Barstow technology (water/steam receiver) in a repowered hybrid plant. Two cases were analyzed by a remix computer model: (1) no buffer (or other thermal storage), and (2) the same plant but with buffer thermal storage. The data are summarized in Table 3-1.

The value of a solar thermal plant is the sum of the net fuel savings, operations and maintenance (O&M), and the capacity credit. The fuel value is the cost of fuels that would have been consumed if the plant had not been repowered. Similarly, the O&M and capacity credits are system costs if the plant had not been repowered. The break-even cost is the initial capital cost of the solar thermal plant, which is equivalent to its value. The model calculates these costs while minimizing total expenses in all cases.

The capacity factor without thermal buffer is greater than the same plant with buffer thermal storage. That fact is due to the additional energy delivered from operating the fossil boiler that is required to provide the buffering against transients. However, the fuel value is not proportionally increased. This effect is due in part to thermal losses in cycling the fossil boiler every day and in part to the mixture of fuels that is being replaced. The plant capacity credit is the same in both cases, since both are hybrid plants. The biggest difference in plant value is due to the reduction in fuel cost in the plant with buffer storage. The net result is an increase in plant value with buffer storage, even though the capacity factor is reduced.



For: Fixed collector area, one location, one application, one solar thermal technology

Figure 3-2. Definition of Thermal Storage Value Terms

Table 3-1. **VALUE COMPARISON, 50 MW_e HYBRID REPOWERED SOLAR THERMAL ELECTRIC PLANT, SYNTHETIC UTILITY E, MIDLAND, TX, COLLECTOR AREA - 23,300 m^2**

	No Thermal Buffer	With Series Thermal Buffer
Solar Plant Value,[a] 1985 M$		
Fuel Value	197.5	189.5
Fuel Cost[b]	(-53.7)	(-2.4)
Variable O&M	3.6	2.6
Capacity Credit	22.3	22.3
Total	169.7	212.1
Break-Even[c] $/$kW_e$, 1977 $	992.2	1161.2
Capacity Factor	0.263	0.217

[a]Present worth.

[b]Cost of fuel burned in the hybrid plant is a negative fuel value.

[c]The Westinghouse model calculates the data in 1985 money; the conversion into 1977 money is presented in the table.

The difference in plant break-even cost (with buffer less no buffer) is $232/kW$_e$. The quantity of thermal storage required is about 1/2 hour (25 MWh$_e$) for this case. That quantity is determined by the time to bring a cold, oil-fired boiler up to full operating condition. For 1/2 hour of thermal buffer storage, the value is about $500/kWh$_e$ (1977$). That value is very large and is much higher than obtainable storage costs. Therefore, the author recommends establishing buffer storage cost goals based upon realistic, obtainable costs.

3.2.2 Value of Diurnal Storage

The value of diurnal storage in solar thermal electric power plants is presented below. The data are calculated with a remix analysis method. Two models for the calculation of solar thermal system value have been developed; one, by Westinghouse (1978) and the other by Aerospace Corporation (Melton 1978). There are significant differences in these models and consequently a significant difference in the data. No attempt is made in this study to explain the differences, and data from both sources are presented.

For both cases, the value data are calculated for a stand-alone solar thermal plant employing Barstow technology, which includes a dual-media thermal storage subsystem. That thermal storage subsystem provides buffering. The value of thermal storage is calculated as the difference in the system value with a given number of hours of storage less the system value with zero hours of storage (i.e., buffering only). An hour of storage is defined as the capability to generate electricity with the plant for one hour when operating from storage.

3.2.2.1 Westinghouse Data

Figure 3-3 presents the total capital value of thermal storage in $/kW$_e$ derived from Westinghouse (1978) data. The data are presented as the change in the present worth of the solar thermal plant valued in 1985 money; the second scale is the change in initial capital value in 1976 money. The data are calculated for small market penetration of solar thermal plants (about 1% of the peak generating capacity is from the stand-alone solar thermal plant). Reliable data for larger market penetrations are not available at this time. Similarly, the data are presented for only one collector area, since Westinghouse did not generate data that would allow reliable calculations for other areas.

The data in Fig. 3-3 show the effect of storage and location on capital value. The data are for the EPRI synthetic utilities (Zaininger et al. 1977) with start-up in 1985. Table 3-2 presents the synthetic utilities, the locations, and the code identification for the graphs. The high value of thermal storage for Inyokern is probably due to the fact that most days are clear at that location. For areas outside the very high insolation regions, the thermal storage values are significantly less (for the same collector area and storage capacity).

The average value of thermal storage is simply the capital value of storage divided by the quantity of storage; i.e.,

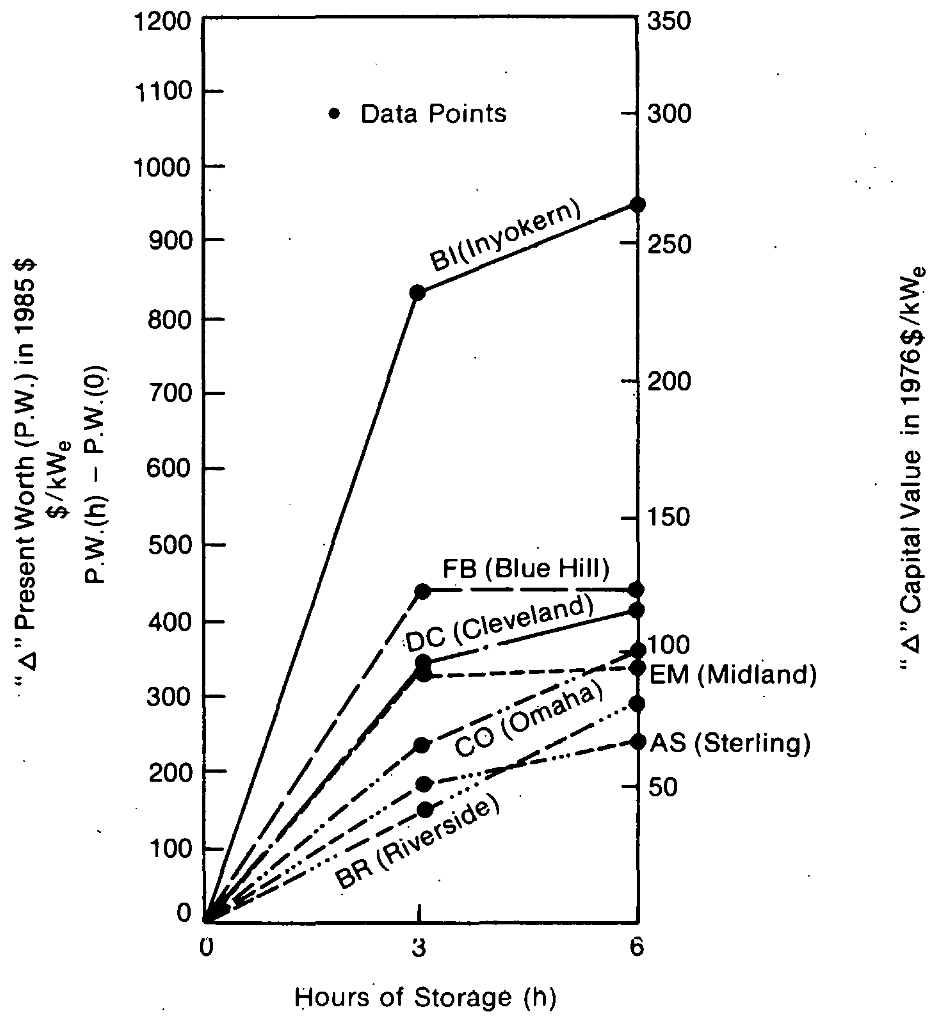


Figure 3-3. Westinghouse Total Thermal Storage Value Data for Barstow Technology 600,000 m² Collection Area 100-MW$_e$ Plant

Table 3-2. WESTINGHOUSE UTILITY AND SITE KEY

UTILITY	SITE	CODE	AREA OF USA
A	Sterling, VA	AS	Mid-Atlantic
B	Inyokern, CA	BI	
B	Medford, OR	BM	West
B	Riverside, CA	BR	
C	Omaha, NB	CO	Plains
D	Cleveland, OH	DC	Northeast
E	El Paso, TX	EE	
E	Midland, TX	EM	South Central
F	Blue Hill, MA	FB	New England
F	Miami, FL	FM	Southeast

$$\begin{array}{l}\text{AVERAGE}\\ \text{VALUE OF}\\ \text{THERMAL}\\ \text{STORAGE}\end{array} (\$/kWh_e) = \frac{\text{CAPITAL VALUE } (\$/kW_e)}{h(\text{hours})}$$

The average value data for Inyokern (a Southwestern site) is 83 $\$/kWh_e$ (1976) at 3 hours and 44 $\$/kWh_e$ at 6 hours. For areas outside the Southwest, the values are much lower: in the range of 14 to 41 $\$/kWh_e$ (1976) at 3 hours and 11 to 20 $\$/kWh_e$ at 6 hours. For locations outside the Southwest, the value does not increase greatly for storage quantities greater than three hours. This fact is due to the finite quantity of heat available from a fixed collector area. Table 3-3 presents the annual quantity of electrical energy calculated by Westinghouse for the Midland, Texas location. For a collector area of 600,000 m^2, the electrical energy delivered was 208.8 GWh/year with no storage, approximately 222 GWh/year with 3 hours, and (also) 222 GWh/year with 6 hours. A substantial increase in delivered energy occurs with the first three hours of storage. However, the next three hours (total of six hours) did not increase the quantity of electrical energy delivered, and thus the value. For larger collector areas, the delivered electrical energy, and thus the value for the higher storage quantities, does increase (see Table 3-3 at 800,000 m^2 collector areas).

3.2.2.2 Aerospace Corporation Data

Figure 3-4 presents total thermal storage capital value from Aerospace Corp. (Melton 1978). These data are calculated for a stand-alone solar thermal plant with Barstow technology. The data are analyzed for a 1990 plant start-up date with approximately 10% market penetration of solar thermal. Aerospace Corp. calculated the value of the storage-coupled solar thermal system employing a remix analysis method. The calculations were also performed employing constant dollar economic assumptions and "snapshot"* fuel prices. SERI converted the data to a current-dollar economic analysis method (same as Westinghouse) employing a 17% fixed-charge rate (rather than 10% with constant dollar) and levelizing the fuel and O&M costs (rather than snapshot). The resulting data are for total thermal storage value with a current-dollar analysis methodology.

Aerospace Corp. presented solar thermal system value for five locations and three collector areas: 500,000 m^2, 1,000,000 m^2, and 1,500,000 m^2. Figure 3-5 presents the total storage value for only the two largest areas. The data for the 500,000 m^2 illustrated a very small value for storage; the range is 12 to 34 $\$/kW_e$ (1977) at three hours with no increase for larger quantities of thermal storage. The small value of storage is due to the limited quantity of heat that is available from the collector field; most of the thermal energy is used to generate electricity directly, with less excess available to charge storage. For larger collector areas, the field provides increased quantities of heat to charge storage, and the value of thermal storage is increased. The largest collector area always gives thermal storage the highest value.

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*A snapshot analysis considers the cost of energy at a given year without levelizing that cost over the life of the plant.

Table 3-3. **STC ENERGY COLLECTION DATA[a] FOR EM (MIDLAND, TEXAS)**

	STC Electric Energy, GWh/yr		
	Storage Capacity		
Collector Area	0 Hour	3 Hour	6 Hour
500,000 m^2		191.2	
600,000 m^2	208.8	(222)[b]	(222)[b]
700,000 m^2		259.3	
800,000 m^2		(277.6)[b]	293.9
900,000 m^2		295.9	

[a]Data from Westinghouse, EPRI 648 study, July 1978 and September 1978.

[b]Estimated by the author.

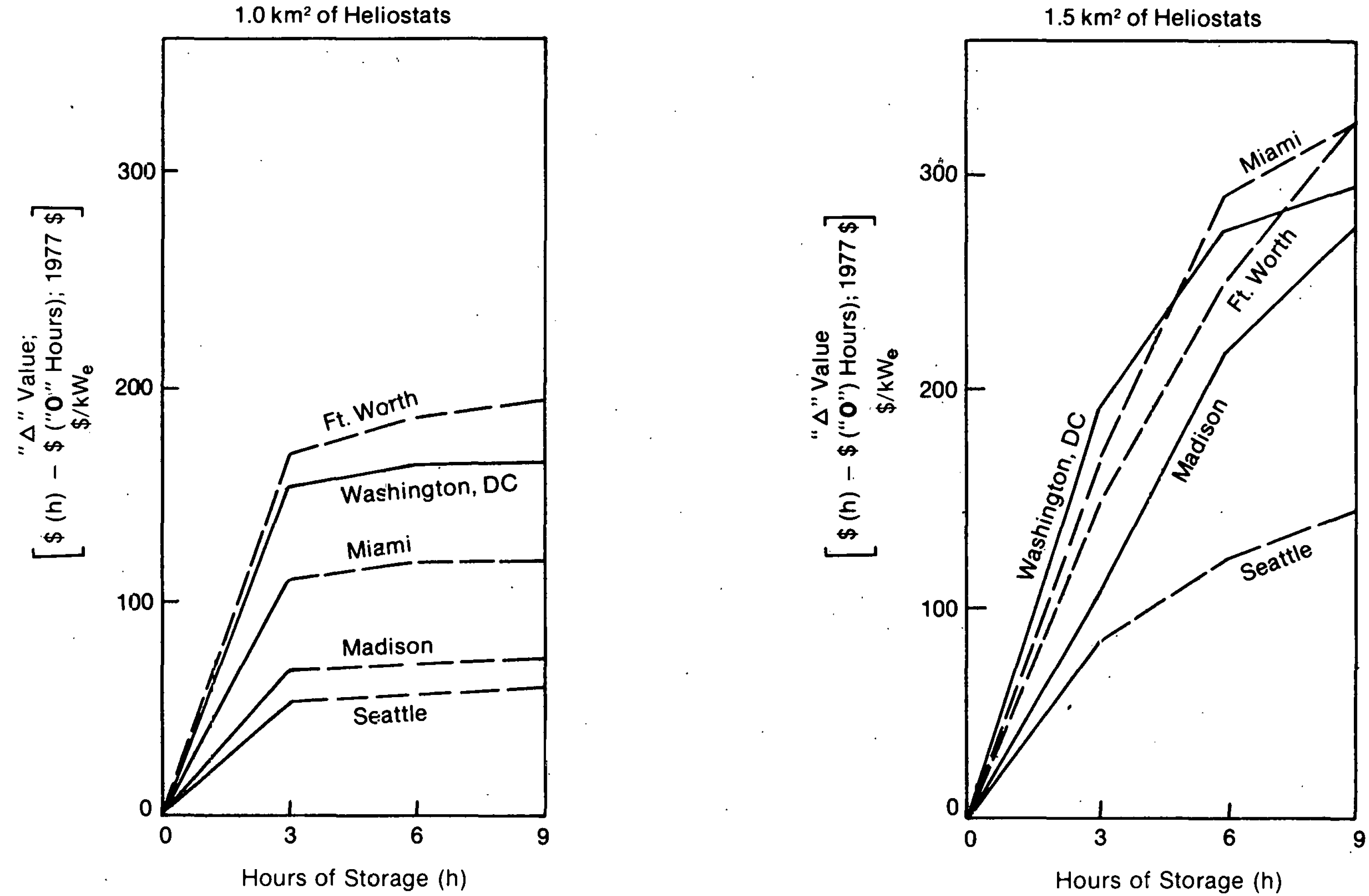


Figure 3-4. Aerospace Total Thermal Storage Value Data for a 100-MW$_e$ Plant: Barstow Technology for 1990 Start-up, 1977$

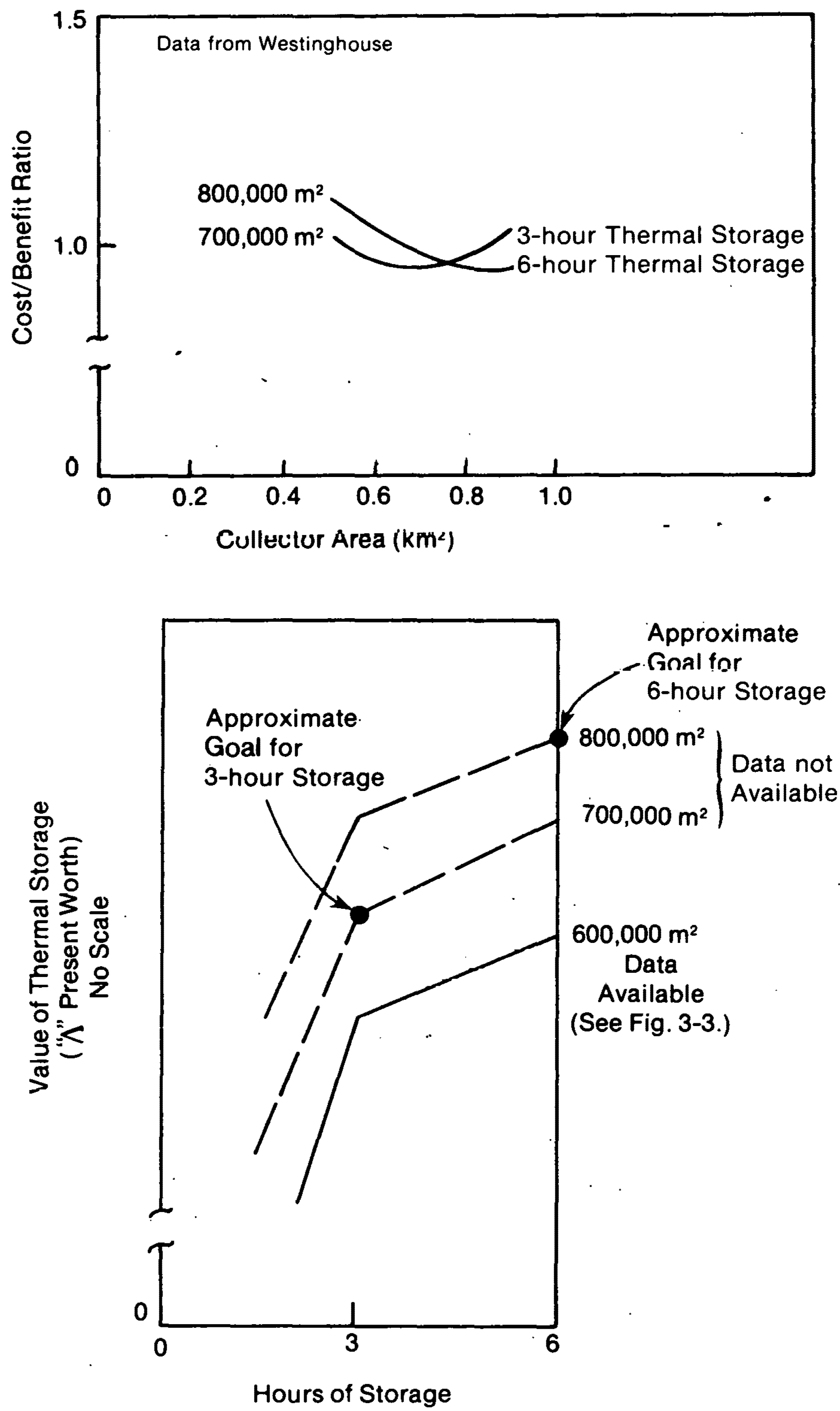


Figure 3-5. Example Case for Identifying Collector Area in Determining Thermal Storage Value

3.2.2.3 Recommended Value Data for Diurnal Storage

The previous paragraphs have presented the value of thermal storage as a function of a range of parameters; thermal storage value (for a given quantity of storage) is strongly dependent upon the collector area. In selecting cost goals (based on value) one must face the issue of selecting an appropriate collector area.

The appropriate choice is obviously the condition that an end user would select: maximizing value while minimizing costs. A quantitative expression for this statement is to minimize the cost/benefit ratio (i.e., cost/value). This condition is a function of the whole storage-coupled solar thermal system economics. Figure 3-5a presents cost/benefit data for one location for the storage-coupled solar thermal system as a function of collector area and storage. For the data in Fig. 3-5a, approximately the same cost/benefit occurs with either 3-hour storage and 700,000 m^2 collector area or 6-hour storage and 800,000 m^2 area. Whether the user chooses 3, 6, or 9 hours or another quantity of storage is not the issue. The issue is what collector area goes with a specified quantity of storage. In this example, the areas are obviously the ones that minimize the cost/benefit. The appropriate values of thermal storage are noted in Fig. 3-5b. Unfortunately, the value data for several collector areas from Westinghouse are not available at this time.

Table 3-4 presents recommended value data for diurnal thermal storage in electric power utilities. Thermal storage value data based upon the Westinghouse and Aerospace studies are separately reported. Those two studies are significantly different in the assumptions for insolation, fuel costs, time frame, market penetration, and remix strategies; and no attempt is made herein to reconcile the differences. By the procedure noted above, the appropriate collector area is that one with the most favorable cost/benefit for the specified storage. With the Aerospace data, determination of that area is possible. The Westinghouse data are all for 600,000 m^2, the only data available. The true value of thermal storage is higher than that reported in Table 3-4, but by an unknown amount.

The value data are reported for three different insolation levels by the author, based upon his expectation for the solar thermal systems. The first applications of storage-coupled solar thermal plants are expected to be in the Southwest. Inyokern data were used to express the value of thermal storage for those high-insolation areas. As the market develops and costs are reduced, solar thermal systems are expected to be competitive in sunny, but lower-insolation areas. Data from Texas sites were employed for those medium-insolation sites. Finally, solar thermal may be used in the lowest-insolation areas of the country. In all cases, the data reflect the total value of the thermal storage system, that value being determined by the sum of the fuel savings, O&M savings, and capital credit for conventional generation systems.

The value of diurnal thermal storage calculated is for Barstow collector and storage technologies, which has inherent limitations with that technology. Specifically, the overall efficiency of storage is low, about 70%. A more efficient storage technology would deliver more usable energy with the same thermal input. Thus, the value of a high-efficiency storage concept is greater than that of a low-efficiency concept, and the reader is cautioned to consider such an effect when using the data presented.

Table 3-4. RECOMMENDED VALUE DATA FOR THERMAL STORAGE

Storage Capacity	Appropriate Collector Area (km²): Westinghouse (W)	Appropriate Collector Area (km²): Aerospace (A)	Average Thermal Storage Value (1976 $/kWh$_e$): High Insolation: W[a] Inyokern	Medium Insolation: W[a] Midland	Medium Insolation: A[b] Ft. Worth	Low Insolation: W[a] Sterling	Low Insolation: A[b] Seattle
3 Hours	0.7	1.0	>83	>30	48	>18	24
6 Hours	0.8	1.0[c]	>45	>15	29	>11	14
		1.5	--	--	36	--	18
9 Hours	ND[d]	1.5	ND	ND	32	ND	15

[a]Based on Westinghouse data for 0.6 km^2 collector area, value is probably higher than these data.

[b]Based on Aerospace data, converted to 1976 money assuming 10% inflation.

[c]The appropriate area is between 1.0 km^2 and 1.5 km^2, but insufficient data exists at this time to determine the areas and the thermal storage value.

[d]ND = No Data Available.

3.2.3 Baseload Storage Value

Long-duration storage is being considered to provide a baseload capability in a Solar Thermal Electric Plant (STEP). A very preliminary analysis was conducted to assess the value of long-duration storage. A next-plant analysis method was employed to compare two cases: one with long-duration storage, the other without.

3.2.3.1 Mission

The mission was defined as a baseload power generation continuously providing power 8760 hours in an average year. Two cases were defined as shown in Fig. 3-6, and have the following costs:

Long Duration Storage: Case 1	Hybrid: Case 2
Short-duration costs	Short-duration storage costs
Capital costs of the STEP	Capital costs of the STEP
Capital costs of the coal plant	Capital costs of the coal plant
Operational costs of the STEP	Operational costs of the STEP
Operational costs of the coal plant	Operational costs of the coal plant
Backup generation and fuel costs of an oil-fired plant	Backup generation and fuel costs of an oil-fired plant
Long-duration costs	

In both cases, exactly the same quantity of energy is delivered during the year at exactly the same time of every day. Furthermore, the nameplate rating for the solar thermal plus coal-fired plants were the same, exclusive of the oil-fired backup plant (which was included to assure that each case could meet the mission, including the mechanical reliability of the plants involved). In Case 1, the nameplate rating is the sum of the plant ratings for the solar thermal plant and the coal-fired plant. In Case 2, the storage-coupled solar thermal plant is not a hybrid and is therefore subject to periods of forced outages due to nonavailability of insolation with only short-duration storage. In Case 2, the solar thermal and coal plant ratings were equal to each other (the sum for Case 1). Case 2 has thus a 100% backup with a coal plant. Case 1 obviously has much lower capital cost since less generation equipment is needed.

As required by the definition of thermal storage value, the solar thermal collector field was constrained to be the same size in both cases; furthermore, the solar thermal collection efficiencies (insolation to heat) were equal. However, the thermal storage efficiency for long duration was allowed to vary up to the efficiency when operating direct. The long-term thermal storage efficiency accounts for both 1st Law (heat losses to the environment) and 2nd Law (conversion cycle). Consequently, the quantity of energy delivered from the solar thermal system in Case 1 was always equal or less than that in Case 2. Since the purpose of long-term storage was to operate the solar thermal plant as a baseload unit, the solar thermal plant rating in Case 1 was reduced to adjust for the long-duration storage efficiency. The difference in solar thermal energy delivery was supplied by increasing the coal plant rating and the total (oil and coal) fuel usage in Case 1.

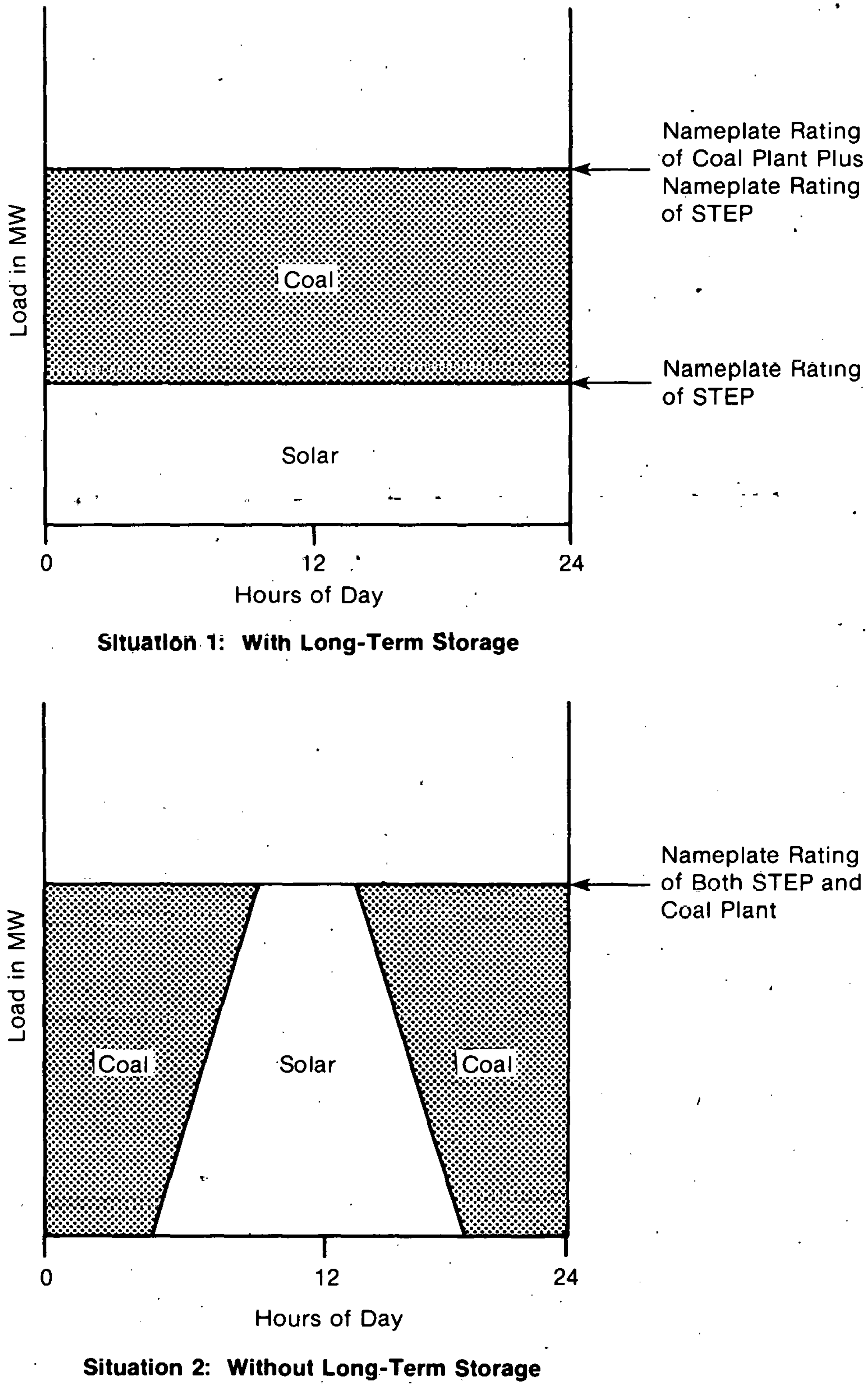


Figure 3-6. Daily Load in Baseload Storage Value Analysis

3.2.3.2 Method of Calculating Long-Duration Storage Value

The value of long-duration thermal storage was calculated as that cost of long-duration storage that makes the total costs in both Case 1 and Case 2 equal; i.e.,

$$\sum \text{Costs}\Big|_{\text{Case 1}} = \sum \text{Costs}\Big|_{\text{Case 2}}$$

Moving all items but long-duration storage from the left-hand side of the above equation reveals that the value of long-duration storage is the difference in costs for the following:

- short-term storage;
- capital of the STEP;
- capital of the coal plant;
- operation of the STEP;
- operation of the coal plant; and
- backup generation and fuel for oil.

Capital cost, O&M cost, and fuel costs were evaluated using data from the EPRI (1978) Technical Assessment Guide. To determine the capital cost of the solar thermal plant with long-duration storage, some assumption on the availability factor was necessary. The assumption was to force the long-duration storage solar thermal plant capacity factor to be equal to that of a baseload coal plant; i.e.,

$$CF_{ST} = CF_{coal}$$

$$(AF)_{ST}\, R_{ST} = R_{coal} = 0.788$$

where

CF_{ST}	is the capacity factor of the solar thermal plant,
CF_{coal}	is the capacity factor of the coal plant,
$(AF)_{ST}$	is the availability factor of the solar thermal plant if there were no equipment outages (scheduled or unscheduled),
R_{ST}	is the reliability factor of the solar thermal equipment,
R_{coal}	is the reliability factor for a coal plant.

Based upon the EPRI (1978) data, the availability factor $(AF)_{ST}$ was 0.94.

3.2.3.3 Long-Duration Value Data

Table 3-5 presents the calculated value of long-duration storage. Four locations and three storage efficiencies are included. For all locations with 100% efficient long-term storage, the quantity of energy delivered from the solar thermal plant is equal in both

Table 3-5. **TOTAL THERMAL STORAGE VALUE FOR LONG-DURATION STORAGE[c]**

Location Hours Storage[d]	AF_{ST}[a]	Total Thermal Storage Value $/kW (1977$) 1.00[b]	0.62[b]	0.41[b]
Albuquerque	.50	765	252	-31
Madison	.37	587	207	-2
Miami	.44	683	231	-18
New York	.35	566	200	2

[a]Availability factors from Iannucci (1978) with the critical solar multiple (Note: what is identified here as AF is called "percent solar" by Iannucci).

[b]Efficiency of long-term storage; the ratio of the electrical energy delivered in Case 1 to Case 2 from the solar thermal system.

[c]For application in the 1990s.

[d]All short-duration storage capacities have 3.0 hours of storage in Case 2.

cases. That fact is true since the change in the STEP nameplate rating (Case 1 vs. Case 2) is exactly proportional to the capacity factor. The quantity of energy delivered from the coal plant and backup oil are also equal (in the two cases) when there are no losses in long-duration storage. The large value is thus due to the capital equipment savings. As the long-duration storage efficiency decreases, the value also decreases sharply. Due to increased costs for a higher nameplate rating on the coal plant and fuel for that coal plant at low efficiencies, the value can actually become negative.

The quantity of long-duration storage required is not known. The assumption in this analysis was to maintain the same capacity factor in a long-duration storage-coupled solar thermal plant as in a coal plant, with both equal to the reliability of a coal plant. That assumption required a 94% availability factor for the solar thermal system. That availability factor required storage quantities in the range of 250 to 830 hours (based on data from Iannucci 1978). However, if the availability factor had been lower, say 90%, the quantity of thermal storage would have been reduced (40 to 100 hours) (Iannucci 1978). The total value of the long-duration storage would be slightly reduced, but would still be very near the value reported in Table 3-5. However, the average value of thermal storage ($\$/kWh_e$) would be increased by a factor of about six by changing the availability factor from 94% to 90%. Additional study is needed on this issue and probably a remix model analysis method will be required.

3.3 ON THE USE OF COST GOALS

3.3.1 Caution on Value Data

The value data have been calculated as the cost of conventional equipment, fuels, and O&M. The equipment and O&M costs are comparatively well known, being based on recent experience with conventional generation equipment in commercial application. The fuel costs are estimates, which are subject to significant uncertainties. The causes are international (OPEC), domestic policy (regulation), transportation (railway rates), synthetic fuels (oil and gas), and new discoveries of oil and gas. These uncertainties cast a proportional uncertainty on the value of solar thermal and thus thermal storage. Table 3-6 illustrates the uncertainty level in fuel costs. Not only is there a significant variation in fuel prices, there is also a wide spread in the escalation ratio. Data from EPRI and Westinghouse are the most conservative, i.e., lowest-priced fuels. The data from EIA and Aerospace are the most liberal. The value data for thermal storage have been calculated from the Aerospace and Westinghouse fuel price assumptions. The results are thus representative of the range of expected values. However, neither the Aerospace nor the Westinghouse fuel price scenario are an extreme. The value of thermal storage may actually be less than or greater than those reported herein. The readers are cautioned to recognize these uncertainties and to use the data accordingly.

3.3.2 Calculation of Obtainable Cost

Cost goals are employed to identify promising concepts for development. The obtainable cost for a thermal storage concept is the factor that is compared to the cost goal. Obtainable cost must include all costs associated with the thermal storage subsystem. Costs associated with storage include direct capital, nondirect capital, and O&M.

SERI

Table 3-6. FUEL COST PROJECTIONS FOR 1990[c]

Fuel	Aerospace (Melton 1978) Cost[a]	Rate[b]	EIA (1977) Cost[a]	Rate[b]	EPRI (1978) Cost[a]	Rate[b]	Westinghouse (1978) Cost[a]	Rate[b]
Nuclear	—	—	0.64	0%	0.66	2.2%	0.78	3.4%
Distillate	4.60	1.2%	4.89	3.5%	3.38	1.0%	3.42	3.7%
Residual	4.12	1.2%	4.18	3.8%	3.03	1.1%	3.11	3.7%
Coal	1.55	2.6%	1.52 to 2.29	1.01%	1.07	0.4%	1.24	1.9%

[a]Costs are in 1978 $/MBtu for a Southwestern site.

[b]Rates are real escalation rates and are equal to the annual price increase rate less the inflation rate.

[c]References noted for each source.

Direct capital includes the power-related costs and the capacity-related costs for the quantity of thermal storage being considered (materials and installation costs). Nondirect capital costs are contingency and spares (A), indirects (B), and interest during construction (C). These three are generally calculated as a percentage of the direct capital. Based on data from Westinghouse (1978), the nondirects are calculated as follows:

	PERCENT OF DIRECT
A. Contingency and Spares	15%
B. Indirects	10%
C. Interest During Contruction (15% of Direct Plus Contingency and Spares Plus Indirects)	19%
Total Nondirects	44%

The nondirect costs are added to the direct for the total capital cost. To that cost must be added the levelized capitalized equivalent of the O&M cost. O&M costs are normally calculated as an annual expenditure. Typical costs are generally in the range of 1 to 3 percent of the direct capital costs per year. For a nominal 2%, the capitalized costs are calculated as follows:

$$\text{CAPITALIZED O\&M} = \frac{\text{PC} \bullet \text{DIRECT} \bullet \text{LF}}{\text{FCR}}$$

where

PC is the percentage of the direct costs,

DIRECT is the direct capital cost of thermal storage,

LF is the levelizing factor,

FCR is the fixed-charge rate of the user.

For nominal factors based upon data from EPRI (Day 1978), the capitalized O&M for electrical utility users is as follows:

$$\text{CAPITALIZED O\&M} = \frac{(0.02\ \$/\$\text{-YEAR})\ \text{DIRECT}\ (1.881)}{(0.17\ \$/\$\text{-YEAR})}$$

$$= 0.2219\ \text{DIRECT}$$

Obtainable costs for thermal storage concepts must be less than the cost goals and are calculated as follows:

$$\text{VALUE} = \frac{\text{COST}}{\text{GOAL}} \geq \frac{\text{OBTAINABLE}}{\text{COST}}$$

$$\geq \text{DIRECT} + \text{NONDIRECT} + \text{O\&M}$$

$$\geq (1 + 0.44 + 0.222)\ \text{DIRECT}$$

$$\geq 1.662\ \text{DIRECT} \quad ,$$

where nominal values previously calculated have been included to illustrate the process.

The value of thermal storage expresses the cost of providing exactly the same function as thermal storage by conventional means. The obtainable cost of a thermal storage concept must be lower than the value-derived cost goal; otherwise the users can be expected to employ conventional generation technology in preference to thermal storage. The cost goals are thus absolute criteria for the selection of thermal storage concepts for development in the Thermal Storage Program. Any concept that can not meet one or more costs should not be pursued. CAUTION: the criteria must be applied for fully developed technology or projections for cost when developed. Many concepts are very early in their development stages. With the current state of the art, the predicted cost of any concept may fail to meet cost goals. When such conditions occur, the proper conclusion is to conduct additional research to identify a configuration that can meet the goals. Only if no such configuration can be identified, or if one assesses such to be unobtainable, the research should be terminated.

SECTION 4.0

A METHODOLOGY FOR COMPARING THERMAL STORAGE CONCEPTS

Thermal storage includes three classes: sensible, latent, and thermochemical. In each class, there are several concepts and combinations of concepts (both within and between class). For all elements in the thermal energy storage program plan, several concepts are expected to have obtainable costs that are less than value-derived cost goals. There are insufficient resources to develop all such concepts, and a ranking methodology for identifying the most promising concepts is needed. The methodology must be versatile enough to screen rapidly and to conduct in-depth evaluations. This section presents the derivation of such a methodology.

The ranking methodology is being developed in two forms: "Simplified" and "Computer." The Simplified version is intended to conduct quick screenings of concepts. It includes both cost and performance factors and quantitatively relates the importance of those factors. This version is intended for use both by SERI and others, including contractors. Since the latter group are anticipated to use the methodology to assess the relative merits of their own concepts, the Simplified version has been prepared for hand calculations to expedite its use.

The Computer version is being developed to conduct in-depth evaluations. Because of the basic differences in thermal storage concepts, each one will have different conditions for which it is most effective. Factors that affect the evaluation include the following:

- solar thermal collector area,
- quantity of storage,
- solar thermal plant location (insolation),
- dispatch strategy for thermal storage,
- solar thermal system.

Since the evaluation of each thermal concept over all of these parameters requires extensive calculations, a Computer routine is being developed to conduct those calculations.

This report documents the derivation of the ranking methodology. Both versions employ the same approach and data; the Simplified version is simply a subset of the Computer version. All data necessary to use the Simplified version are included here; the Computer version is under development and will be documented separately.

4.1 APPROACH

The approach taken in the ranking methodology is to compare thermal storage concepts when all are performing the same mission. The approach requires defining a mission by specifying one set for the following parameters:

- one solar thermal receiver collector system,
- one application,

- one collector area,
- one location,
- one dispatch strategy,
- one thermal storage capacity,
- one plant rating,
- one rate of electrical energy and thermal energy delivery (total energy systems),
- one rate of charging thermal storage,
- one rate of discharging thermal storage.

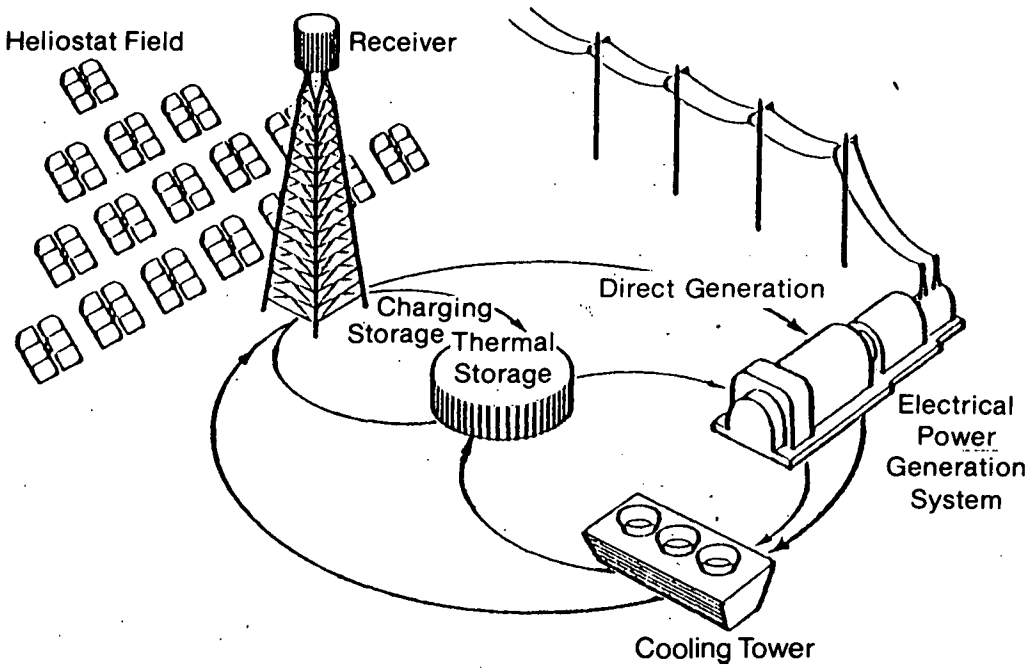
In both versions, the ranking methodology repeatedly does the comparison for different missions. As implied from the above definition of mission, changing only one factor, even slightly, defines another mission.

Figure 4-1 illustrates the basics of the approach. A complete storage-coupled system is defined. This system is designated as the Reference System and includes collector field, receiver, conversion equipment, etc. A thermal storage concept is also defined for the Reference System and is identified as R. Next, a single mission is defined by specifying all of the parameters previously noted. From the set of "alternative" thermal storage concepts, one is selected and is identified as A. (Obviously, A is any of them, and the process will be repeated for each in the set.) Concept R is removed from the Reference System and replaced with concept A. The mission is required to be the same. Thus both A and R must have the same capacity to store heat and the same thermal-storage charging and discharging rates. Because of technical differences in the storage concepts, changes will occur in the solar thermal system. Some examples of items that might be affected are:

- operating temperatures in the system
  - receiver
  - conversion cycle
- plant equipment
  - dual admission/single admission turbines
  - number or types of heat exchangers

Cost differentials associated with the changes are assigned to concept A. Performance factors will also be affected when A is the thermal storage concept. For example, changes in a temperature can affect the receiver efficiency, the conversion cycle efficiency, the heat losses from the system, and other factors. The net effect is to change the delivered electrical (or thermal) energy from thermal storage.

After the alternative concept has been integrated, the cost and performance factors are evaluated for one mission. The delivered energy costs of A are then calculated and compared to R. The process is then repeated over the expected range of parameters for all missions.



All other elements are unchanged (except as may be associated with the new storage, e.g., a dual admission turbine may be replaced with a single admission turbine if the storage allows it).

Figure 4-1. Basics of Approach

4.1.1 Ranking Index

The quantitative evaluation criteria in the ranking methodology is the unit energy cost (bus bar energy cost, BBEC, for electrical energy). The remainder of this discussion employs BBEC for the unit energy costs. For process heat and total energy applications, the equations are the same except for modifications in the term for unit energy costs. BBEC is selected because electrical power is one of the projected uses for all program elements.

BBEC divided by a constant provides exactly the same ranking function as BBEC itself. In the methodology, that division is done and is defined as the Ranking Index (RI), i.e.,

$$RI(A) = \frac{BBEC(A)}{constant} \tag{4-1}$$

The constant is selected to be BBEC(R), the unit energy cost of the reference system; hence the following is true:

$$RI(A) = \frac{BBEC(A)}{BBEC(R)} \quad . \tag{4-2}$$

RI is a dimensionless number and is as quantitative an evaluation criteria as BBEC. RI is the ratio of unit energy cost for A to that of R. Since A is only one of a set of thermal storage concepts, the relationship is true across the whole set.

The Ranking Index is always applied when the concepts are performing the same mission. Within that constraint, BBEC(R) can vary from mission to mission, and in general does so vary. The approach is to vary the mission [thus generating a new number for BBEC(R)] and to compare all alternative concepts to the reference case. Data of the form illustrated in Fig. 4-2 will be generated. The data in this chart are not real and are presented only to illustrate the approach. Clearly, within the range of parameters, the minimum cost of every alternative concept (A, B, C, D, etc.) is somewhere in the data. Precise knowledge of the minimum cost point conditions is not available, but it is also not needed. By inspection of the type of data presented, the best thermal storage concepts are obvious. Also, as indicated in Fig. 4-2, no one "best" thermal storage concept is expected. For example, with short-duration storage, one concept may be preferred (i.e., have the lowest RI); but as storage capacity increases, another concept may be preferred.

The identification of promising storage concepts is dependent upon their intended use (i.e., mission). Solar thermal technologies are being developed for many applications. Early market opportunities are likely to be employed in peaking to intermediate use in electric utilities. However, as the technology develops, baseload use can reasonably be expected. The choice of appropriate storage missions is thus tied to program emphasis. The methodology is general and may be employed regardless of the program emphasis.

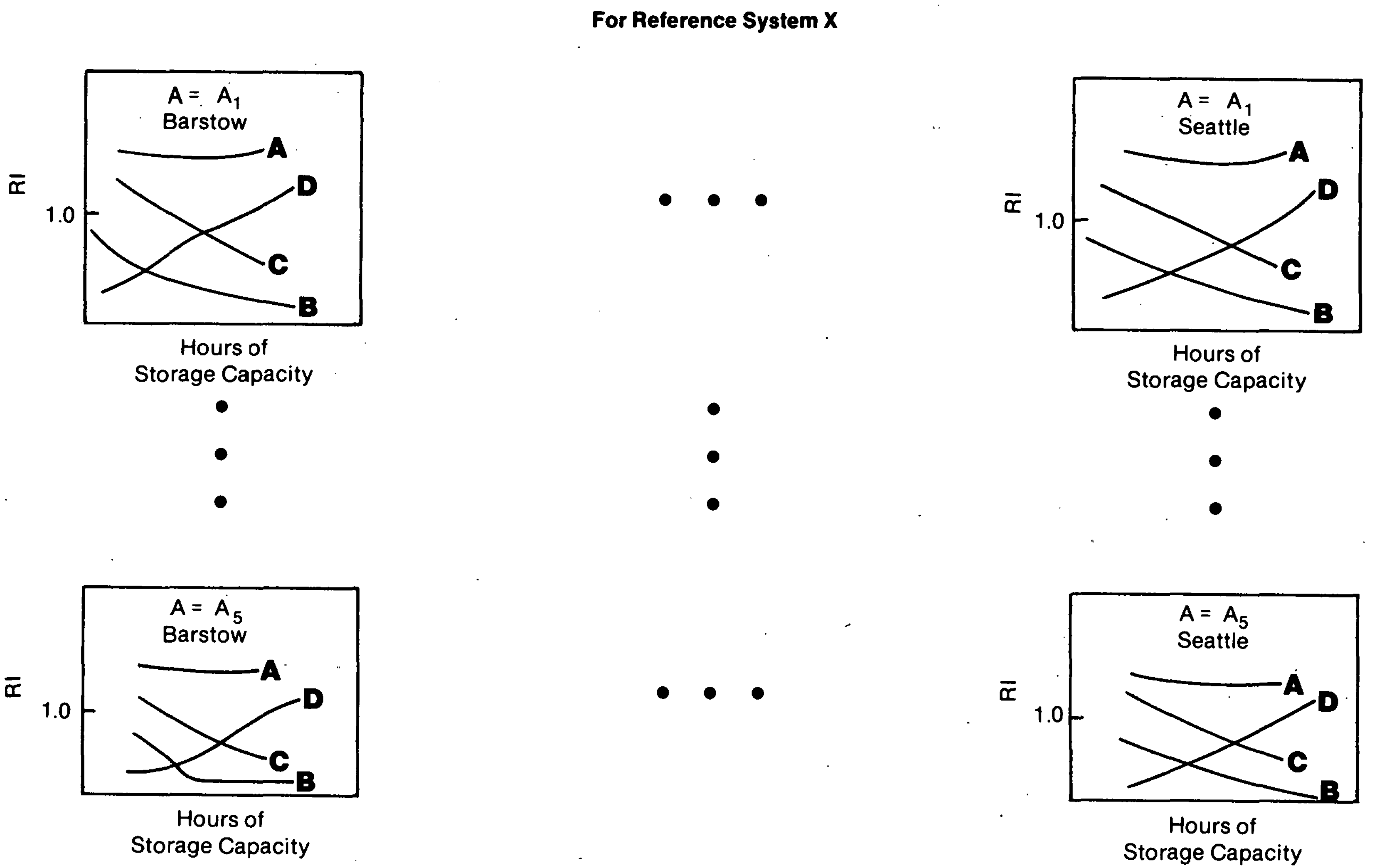


Figure 4-2. Ranking Thermal Storage Concepts with Varying System Parameters (i.e., Missions)

4.1.2 Definition of Storage Capacity, Thermal and Electrical

The Ranking Index requires that concepts be compared when they are performing the same mission. This requirement specifically demands that all have the same ability to accept thermal energy from the receiver (i.e., charging rate) and to supply thermal energy (i.e., discharging rate). Furthermore, the thermal storage must be dispatched in the same manner; thus all concepts must have the same ability to store thermal energy.

These requirements do not demand that for a given mission the same quantity of usable energy be delivered from storage, nor that it be at the same electrical rating. In general, the differences in the concepts' storage efficiencies will cause the delivered energy to be unequal. This fact is included in the calculation of BBEC, and the Ranking Index does account for storage efficiencies.

To avoid confusion in specifying the storage capacity, two terms are defined. The terms are electrical storage capacity (a measure of the energy extracted from storage) and thermal storage capacity (a measure of the energy supplied to storage). The definitions are as follows*:

Electrical: The capacity of one-hour storage, electrical, is the ability to generate the nameplate storage electrical rating of the plant for one hour when operating from storage.

Thermal: The capacity of one-hour storage, thermal, is the ability to store the thermal energy from the receiver that would have operated the plant for one hour at its nameplate direct electrical rating if the thermal energy had not been stored.

Most previous studies comparing thermal storage concepts have conducted the trade-offs with constant electrical storage capacities. This ranking methodology requires constant thermal storage capacities for the comparisons.

4.1.3 Relative Value of Thermal Storage

The users of a solar thermal system are expected to select systems that provide the least cost and greatest value. The cost/benefit (i.e., cost/value) ratio has been used to determine the best conditions. The Ranking Index is based solely on cost. The question arises as to the validity of that approach. The following paragraphs will show that the relative value is near unity, and a cost-only ranking is equivalent to a cost/benefit ranking. A Ranking Index based upon cost/benefit (C/B) is the ratio of the C/B of A to R. The following equation expresses the relationship:

$$\text{C/B RI(A)} = \frac{\text{(C/B)A}}{\text{(C/B)R}} \tag{4-3}$$

*In general, the nameplate storage electrical rating does not equal the nameplate direct electrical rating, e.g., Bastow: 10 MW_e (direct), with 7 MW_e (storage).

Rearranging:

$$C/B\ RI(A) = \frac{\text{COST (A)/COST (R)}}{\text{VALUE (A)/VALUE (R)}} \tag{4-4}$$

$$= \frac{\text{BBEC(A)/BBEC(R)}}{\text{VALUE (A)/VALUE (R)}} \tag{4-5}$$

$$= \frac{\text{RI(A)}}{\text{VALUE (A)/VALUE (R)}}\ , \tag{4-6}$$

where

C/B RI = the Ranking Index based on cost/benefit;

$\frac{\text{VALUE (A)}}{\text{VALUE (R)}}$ = the relative value of the storage-coupled sytems, A to R.

Clearly, if the relative values are equal, the two Ranking Indices are equivalent.

Thermal storage concepts must be compared when all have the same thermal storage capacity. Because of the differences in thermal storage efficiencies, the capacity factor of the solar thermal system will be affected by the choice of concept. Typically, thermal storage efficiencies are in the range of 70% to 98%. This variation can affect the solar plant capacity factor by as much as 20% (depending upon location, collector, and storage capacity). The value of the useable energy from a solar thermal plant is in general a function of capacity factor. The question is whether or not the variation in plant value is significant enough to affect the trade-offs.

An analysis of the relative value of thermal storage-coupled solar thermal systems was conducted to answer this question. Data were generated for EPRI synthetic electric utilities B (Inyokern, California) and E (Fort Worth, Texas). The results indicate that the value of storage is affected by the electrical storage capacity. The magnitude of the variation was small, generally within a range of approximately 2% (e.g., the value of the energy for storage at 5-hours capacity is within 2% of the value of the energy at 7-hours capacity). This fact is relatively true for all storage capacities less than 8 hours, electrical. Significant variations in value occur for large storage capacities (greater than 9 hours), but only when the market penetration is large, on the order of 10% to 20% of the peak utility generation capacity.

No significant difference in value is expected for small utilities and process heat. The reason is that those users in general have the same alternative fuel supply at all times. In total energy systems (cogeneration), the value may or may not be time-of-day dependent. If the user has a fossil fuel alternative, no value difference with storage capacity is expected. If that user buys electricity and also has time-of-day rates available, there may be some value difference at the larger storage capacities.

The ranking methodology makes the assumption that value is not a function of the storage capacity (electrical). For small changes, this assumption is approximately true; the magnitude of the error is on the order of 2% or less. The value difference is not expected to affect the results of the ranking significantly.

4.2. THE RANKING FUNCTION

The Ranking Index is calculated by the ranking function, which is derived in Appendix A and given below:

$$\mathrm{RI}\,(A) = \frac{\dfrac{(CC)\,A}{(CC)\,R} \bullet \left[\dfrac{ESR}{EDR} + 1\right]}{\left[\dfrac{A}{R}_{RT} \bullet \dfrac{A}{R}_{CYCLE} \bullet \dfrac{A}{R}_{COLLECTOR} \bullet \dfrac{ESR}{EDR}\right] + \left[\dfrac{AD}{RD}_{CYCLE} \bullet \dfrac{AD}{RD}_{COLLECTOR}\right]} \tag{4-7}$$

The above function will evaluate RI for one mission. Repeating the calculation for various missions will determine the ranking over as large a range as the user desires.

The function does not require an insolation simulation model of the solar thermal system for every alternative concept to perform the calculation. Insolation is considered in the derivation, but the approach forces the insolation factors to divide out. This property greatly reduces the work required to evaluate the rankings and allows the Simplified version to produce accurate results. The Computer version provides the same data, but over a larger range of missions. Furthermore, since cost data on the reference system is provided, the user does not have to generate that data, reducing his work load and also assuring a consistency in the cost data (in the common elements). ESR/EDR is a function of the reference system only and data are provided for that factor. ESR/EDR contains the performance effects associated with collector area, storage capacity, location, and dispatch strategy. Specific knowledge on those items is not required; simply evaluating the ranking function over the expected range of ESR/EDR will provide the same information as varying all of the factors individually, but with less work. The following sections provide data on the reference systems and an example case illustrating the use of the ranking function.

4.3 EVALUATION OF DATA

This section presents data on the reference systems for use in the ranking function.

4.3.1 Reference Solar Thermal Systems

The original plan of Thermal Energy Storage Technology Development for Solar Thermal Power System (U.S. DOE 1979) called for the development of buffer storage, diurnal storage, and advanced technologies (including long-duration and thermochemical transport). Table 4-1 presents reference systems for the original plan. Element 1 was defined for diurnal storage for Large Power Systems (LPS) with steam power cycles. Three sub-elements (1-A, 1-B, and 1-C) were defined for the major solar thermal systems currently under development. Element 2 was defined for diurnal storage for Brayton Power Cycles. Element 3 was defined for diurnal storage with Small Power Systems (SPS). Three sub-elements were defined for the major SPS systems under development. Element 4 was defined for thermal buffering storage with three sub-elements. Element 5 was defined for long-term thermal storage with LPS in baseload use. Element 6 was defined for SPS with thermochemical energy transport and/or storage. Appendix B presents system schematics and cost data for these reference systems. It also identifies the base references from which additional information may be obtained.

The current version of the program plan calls for the development of thermal storage for the following solar thermal systems.

SOLAR THERMAL SYSTEM (Collector/Receiver)	REFERENCE SYSTEM ELEMENT NUMBER
• Water/Steam	1-A
• Molten Salt	1-C
• Liquid Metal	1-B
• Gas	2
• Organic Fluid	3-B
• Liquid Metal/Salt	4-B

The above indicates the appropriate reference system for each of the current program elements. The remaining reference systems may be used but do not represent current activity in the program.

4.3.2 Reference Thermal Storage Concepts

Table 4-2 presents the reference thermal storage concepts for the current program elements. The first column identifies the program element. The second column identifies the reference thermal storage concepts for all but the 6th (liquid metal/salt collector/receiver). That program element is not sufficiently defined at the current time to specify a reference storage system. The total cost columns present the power-related and capacity-related costs, including nondirect factors. The nondirect factors include contingencies, spares, indirects, and interest during construction; these factors increase the direct costs by 44%. η_{RT} is the 1st Law efficiency of the thermal storage concept. The direct costs columns present the costs for materials, labor, transportation, and installation of the thermal storage concept. The last column presents the 2nd Law efficiency (i.e., the thermodynamic conversion cycle efficiency) when operating through storage.

The data in Table 4-2 are given for electrical power production. To determine the costs for thermal energy only (i.e., process heat) divide these data by η_{CYCLE}. For example, consider a 100 MW_T (from storage) process heat use with a water/steam collector receiver. The steam conditions (from storage) are assumed to be the same as in the electric power case. It is also assumed that the storage capacity is 600 MWh_T. The total storage costs can be calculated as follows:

Table 4-1. REFERENCE SYSTEMS

Element Number	Solar Thermal Application LPS—Large Power Systems SPS—Small Power Systems	Reference Systems
1.	Diurnal Storage (2-12 hours) for 100 MW_e	
	Systems Utilizing Steam Rankine Power Cycle (1000°F) (LPS)	
1-A	Water/Steam Receivers	A. Barstow
1-B	Liquid Metal Receivers	B. Liquid Metal
1-C	Molten Salt Receivers	C. Martin/Molten Salt
2.	Diurnal Storage (2-12 hours) for Systems 100 MW_e Utilizing Brayton Power Cycles (1500°F) (LPS)	Boeing
3.	Diurnal Storage (2-12 hours) for Systems Utilizing Organic Rankine (600°-800°F) or Steam Rankine (800°-1000°F) Power Cycles (SPS)	
3-A	Small Community (10 MW_e)	A. Barstow Technology
3-B	Total Energy 400 kW_e + 1580 lb/h Steam	B. Shenandoah (dishes)
3-C	Remote Power 150 kW_e	C. Troughs + Organic Cycle
4.	Thermal Buffering (1/2 to 2 hours)	
4-A	Steam Rankine Power Cycles (1000°F) (LPS)	A. PSNM Repowering
4-B	Dish-Mounted Power Cycles (SPS) Brayton/Stirling (1300°-1600°F)	B. Dish-Stirling
4-C	Advanced Brayton Power Cycles (2000°F)	C. Black & Veatch
5.	Long-Term ($>$12 hours) Storage (LPS) 100 MW_e	Liquid Metal
6.	Transport and/or Storage (SPS)	Dish + Steam Rankine
6-A	10 MW_e	
6-B	300 kW_e	

Table 4-2. REFERENCE THERMAL STORAGE CONCEPTS COSTS AND EFFICIENCIES[a]

Solar Thermal System (ID. Number)	Type	TOTAL COSTS[c] C_P ($\$/kW_e$)	C_S ($\$/kWh_e$)	DIRECT COSTS[b] C_P ($\$/kW_e$)	C_S ($\$/kWh_e$)	η[d] (%)	ηCycle[e] (%)
1. Water/Steam Receiver (1-A)	Oil/Rock	100	29	69	20	92	25
2. Molten Salt Receiver (1-C)	Dual Tanks, External Insulation	10	27	7	19	98	40
3. Liquid Metal (1-B)	Dual Tanks, External Insulation	43	86	30	60	98	40
4. Hot Gas Receiver (2)	Ceramic Brick, Welded Steel Tank	83	60	58	42	98	30
5. Organic Fluid Receiver (3-B)	Oil/Rock	304	321	211	223	93	15
6. Liquid Metal Salt Collector/Receiver (4-B)	To Be Determined						

[a]All Costs in 1978 \$ and include nondirect capital costs but not O&M (i.e., divided by 1.44 to obtain the direct cost)

[b]Direct costs to obtain total cost; multiply by 1.44 to include the nondirect factors.

[c]C_T ($\$/kW_e$) = $C_P + C_S \cdot H$ (H is hours of storage, electrical)

C_T ($\$/kWh_e$) = $(C_P + C_S \cdot H)/H$

[d] $\eta = \dfrac{\text{Heat out}}{\text{Heat in}}$ (1st Law efficiency)

[e]Cycle efficiency = $\dfrac{\text{Work out}}{\text{Heat into Cycle}}$

Power-Related Costs:

$$C_P\ (\text{ELECTRICAL}) \bullet \begin{matrix}\text{CYCLE}\\\text{EFFICIENCY}\end{matrix} \bullet \begin{matrix}\text{NAME}\\\text{PLATE}\\\text{RATING}\end{matrix}$$

$$= \frac{\$100}{kW_e} \bullet \frac{0.25\ kW_e}{kW_T}\ (100\ MW_T) = \$2.5(10)^6 \quad ;$$

Capacity-Related Costs:

$$C_S \bullet \begin{matrix}\text{CYCLE}\\\text{EFFICIENCY}\end{matrix} \bullet \text{CAPACITY} \bullet \begin{matrix}\text{NAME}\\\text{PLATE}\\\text{RATING}\end{matrix}$$

$$= \frac{\$29}{kWH_e} \bullet \frac{0.25\ kWH_e}{kWH_T}\ (600\ MWH_T) = \$4.39(10)^6 \quad ;$$

Total Costs:

$$\$2.5(10)^6 + \$4.39(10)^6 = \$6.96(10)^6 \quad ;$$

Direct Cost:

$$\frac{\$6.96(10)^6}{1.44} = \$4.78(10)^6 \quad .$$

The same type of process is used to determine the thermal rating of the reference system for electric power. For example, consider a 100-MW_e water/steam reference plant that has a rating of 70 MW_e from storage and a storage capacity of 6 hours at 70 MW_e (420 MWh_e).

Plant Storage Thermal Rating:

$$\frac{\text{STORAGE ELECTRICAL RATING}}{\eta\ \text{CYCLE}} = \frac{70\ MW_e}{0.25\ \frac{kW_e}{kW_T}} = 280\ MW_T$$

Plant Storage, Thermal Capacity (six hours):

$$\frac{\text{STORAGE CAPACITY, ELECTRICAL}}{\eta\,\text{CYCLE}} = \frac{420\ \text{MWh}_e}{0.25\,\dfrac{\text{kW}_e}{\text{kW}_T}} = 1680\ \text{MWh}_T$$

The reader is reminded that the thermal ratings, power and capacity, must be equal when comparing alternative thermal storage concepts.

4.3.3 Conversion Cycle Efficiencies

The conversion cycle efficiencies have been calculated for all reference systems except Stirling cycles. These efficiencies have been calculated in a self-consistent manner by Stearns Roger. Appendix C presents that data and a report on the analyses. That data also presents the cycle efficiency of each system. The Stearns-Roger data is slightly different from the efficiencies in Table 4-3; that is not an unexpected condition, since the data were generated by different people. The data are sufficiently close that no correction is necessary as long as the data is used in a consistent manner.

Figure 4-3 presents the ratio of conversion cycle efficiencies for Program Element 1, water/steam collector/receiver. These data were calculated for a Barstow technology reference system. The power plant is a nonreheat steam Rankine cycle. A dual admission turbine is employed for operation from storage. The data are presented as a function of steam conditions, temperature, and pressure for both direct and storage operations.

Figure 4-4 presents the ratio of conversion cycle efficiencies for Program Elements 2 and 3, molten salt collector/receivers and liquid metal collector/receivers. The data were generated based on the G.E. design with a liquid metal system. The molten salt and other liquid metal designs have slightly different reference conditions. The data may be adjusted by a simple correction factor as illustrated below:

$$\frac{\eta\,\text{CYCLE(A)}}{\eta\,\text{CYCLE (NEW)}} = \frac{\eta\,\text{CYCLE(A)}}{\eta\,\text{CYCLE (G.E.)}}\ \frac{1}{\dfrac{\eta\,\text{CYCLE (NEW)}}{\eta\,\text{CYCLE (G.E.)}}}\ ,$$

where both ratios are determined from Fig. 4-4. The data were based on a reheat steam Rankine cycle with reheat temperature equal to the high-pressure turbine throttle temperature.

Figure 4-5 presents the ratio of conversion cycle efficiencies for Program Element 4, gas collector/receiver. The data were generated for a closed Brayton cycle with oil as the working fluid. The cycle includes regeneration and one intercooler between two compressors.

Figures 4-6 and 4-7 present conversion cycle efficiency data for a total energy system, Program Element 5. The data were generated for a steam Rankine cycle with steam extraction for process heat. Steam condensing occurred at a high temperature (230°F) to provide heat for building heating, absorption, air-conditioning, and hot water. Figure 4-6 presents the ratio of conversion cycle efficiencies with a fixed process heat use rate.

Table 4-3. ASSUME PARAMETERS FROM THE EXAMPLE

Item	Oil/Rock (Reference System)	Alternative	$\frac{A}{R}$	
Storage				
- 1st Law Efficiency	95%	95%	$\frac{A}{R}_{RT}$	= 1.0
- Cycle Efficiency	27.7%	36.2%	$\frac{A}{R}_{Cycle}$	= 1.307
- Receiver Temperature	Assumed Equal		$\frac{A}{R}_{Collector}$	= 1.0
- Cost (Direct) Power Related	$165/kW$_e$	$15/kW$_e$		
Energy Related	$15/kWh$_e$	$85/kWh$_e$		
- O&M Cost	Assumed Equal			
Direct				
- Cycle Efficiency	36.2%	36.2%	$\frac{AD}{RD}_{Cycle}$	= 1.0
- Receiver Temperature	Assumed Equal		$\frac{AD}{RD}_{Collector}$	= 1.0
Other				
- Electric Plant	Dual-Admission Turbine	Single-Admission Turbine (Costs assumed equal)		

*Note: These data were assumed to illustrate the ranking methodology and do not necessarily represent data for real systems.

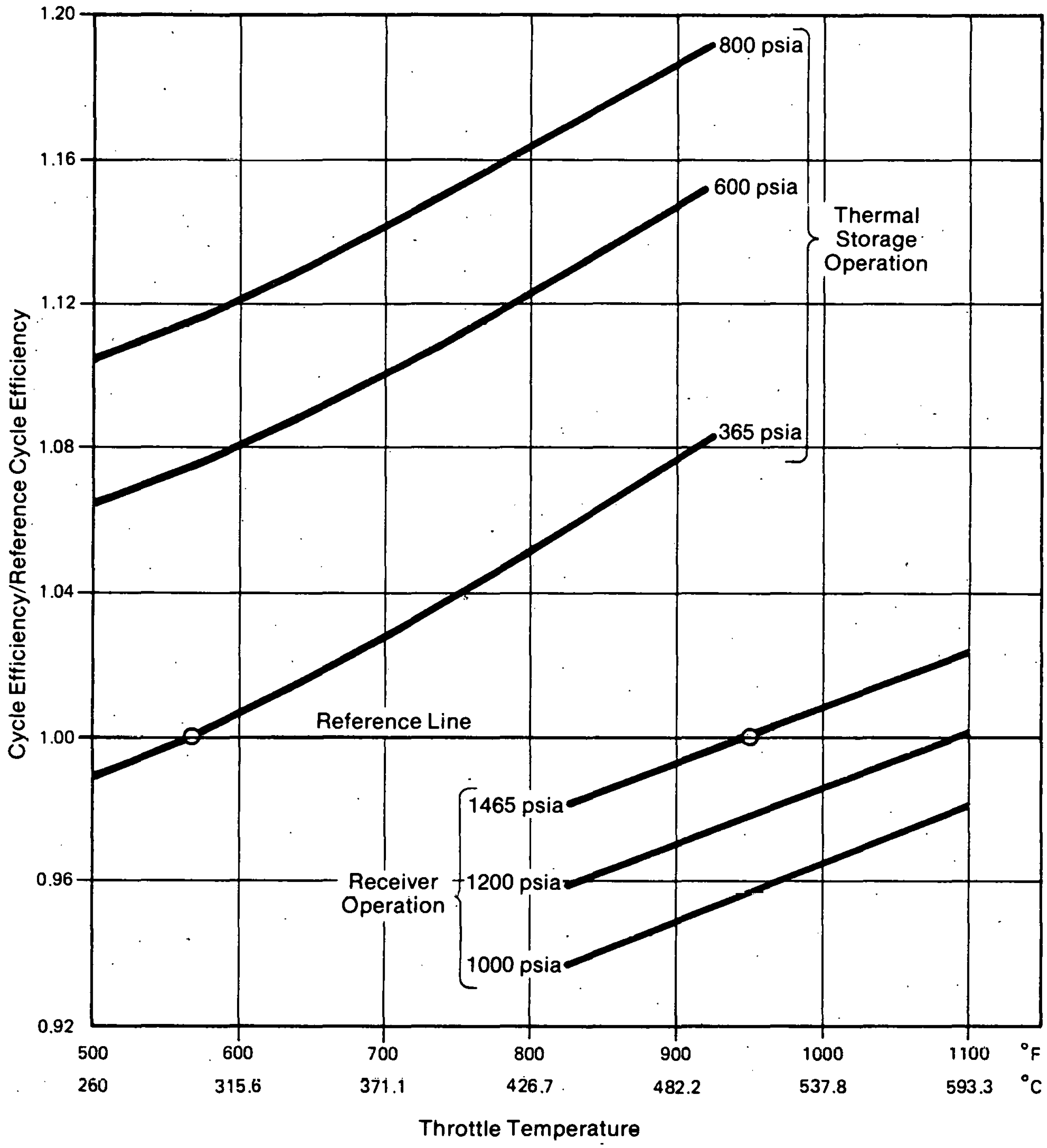


Figure 4-3. Program Element 1, Water/Steam Collector/Receiver Steam Rankine Nonreheat Cycle, Relative Cycle Efficiency vs. Throttle Temperature

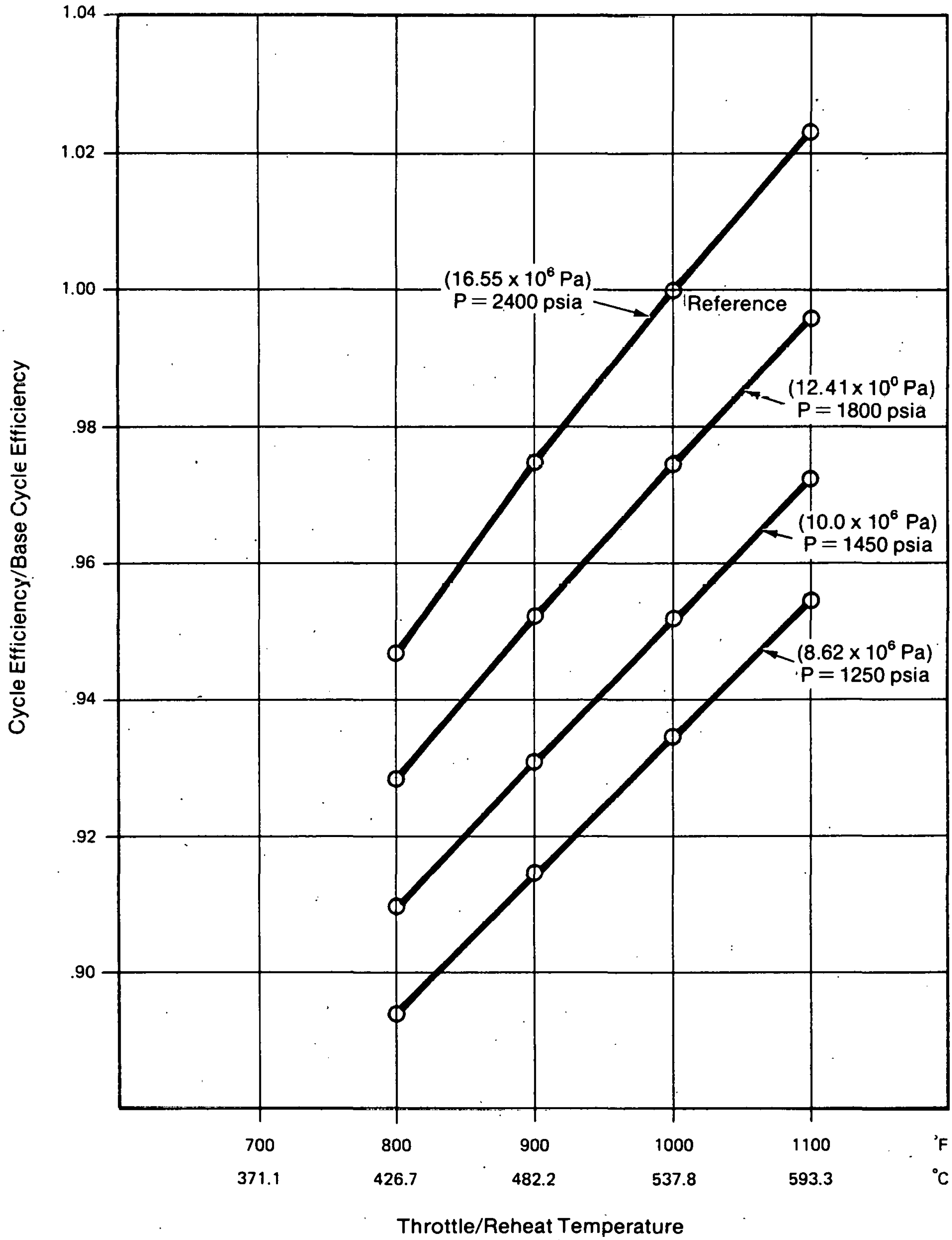


Figure 4-4. Program Elements 2 and 3 - Reheat Steam Rankine Cycle, Relative Cycle Efficiency vs. Throttle Temperature

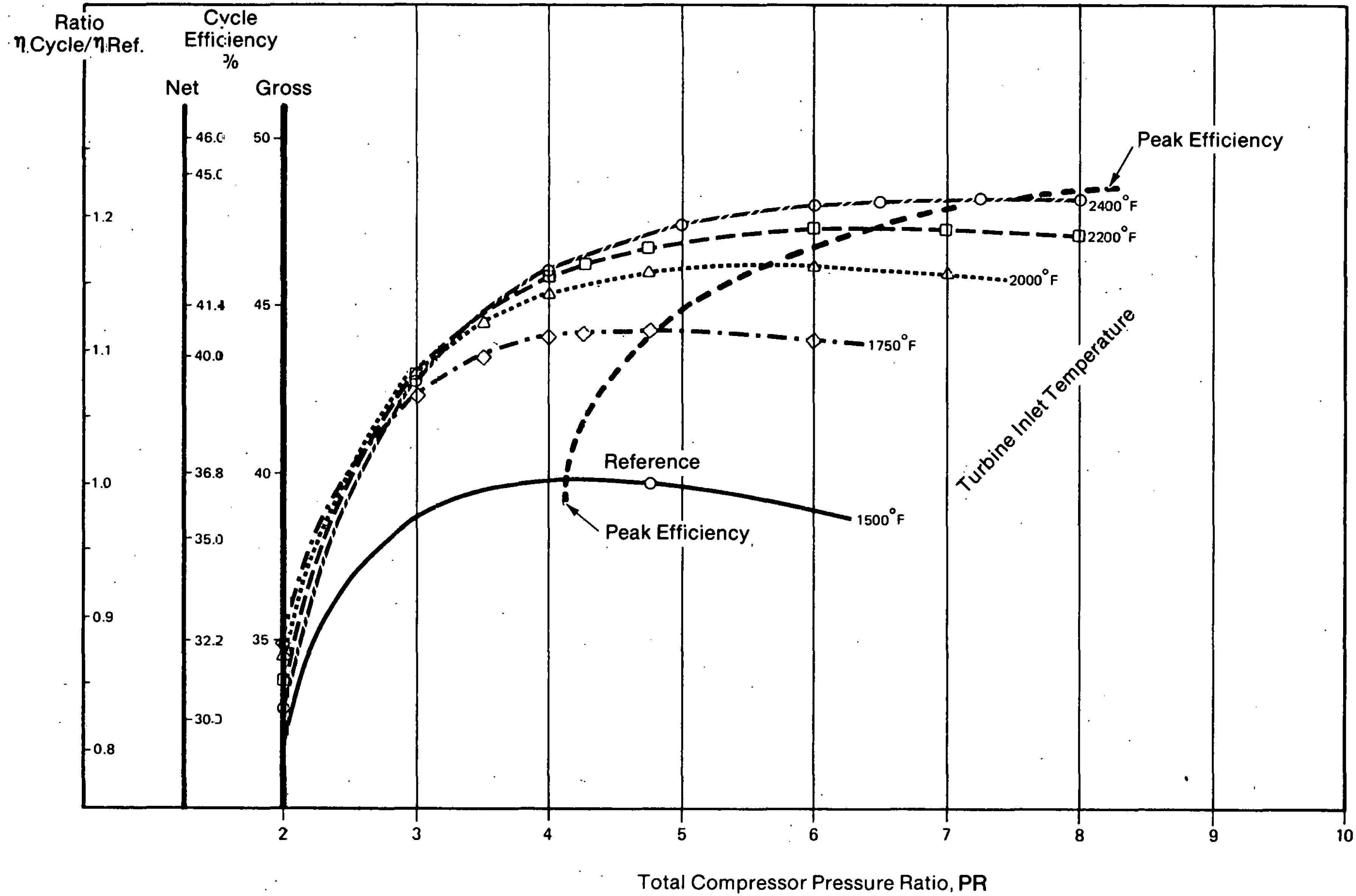


Figure 4-5. Program Element 4 — Closed Advanced Air Regenerative Cycle, Cycle Efficiency vs. Compressor Pressure Ratio & Turbine Inlet Temperature

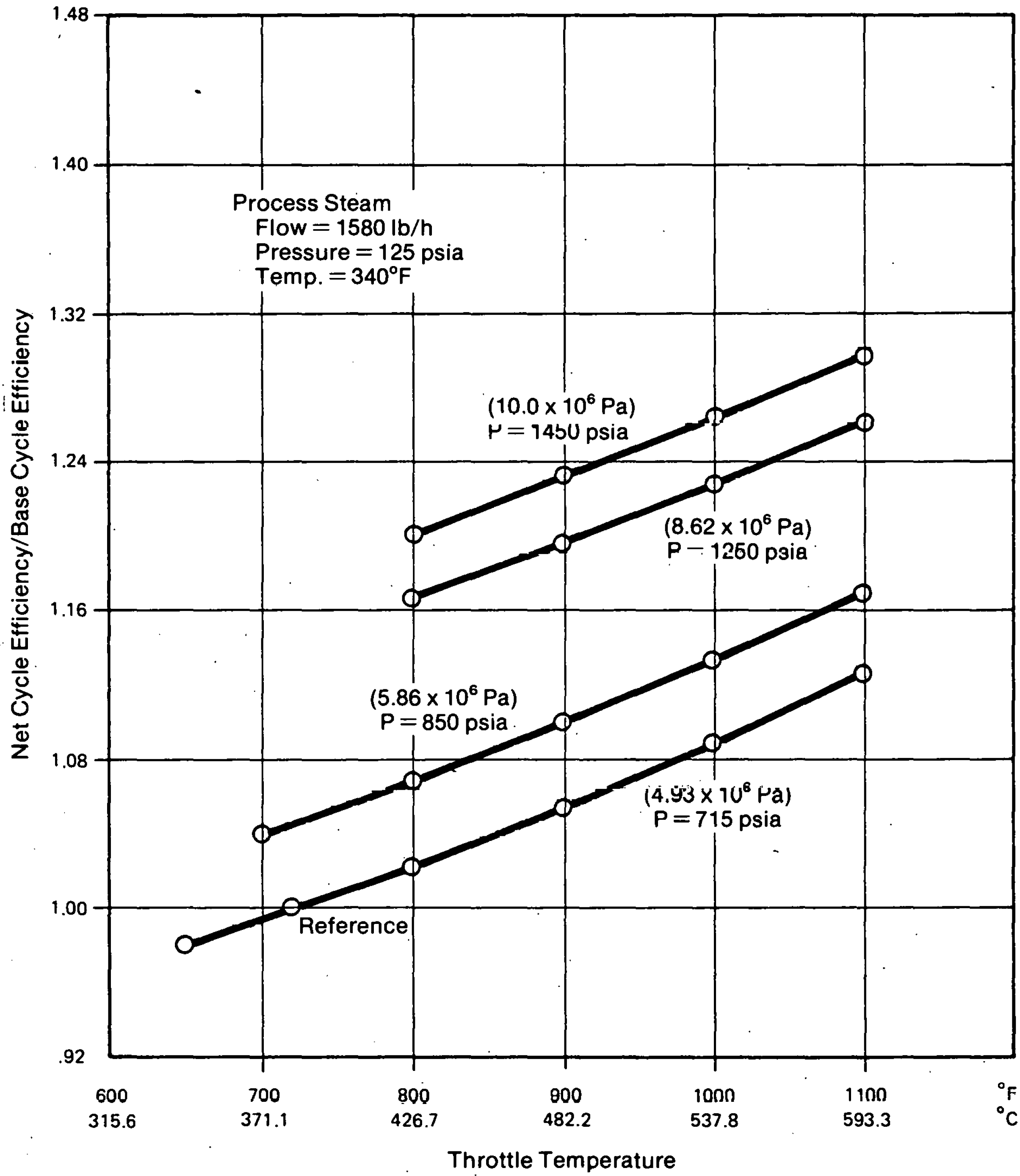


Figure 4-6. Program Element 5 — Total Energy — Steam Rankine Cycle, Relative Cycle Efficiency vs. Throttle Temperature

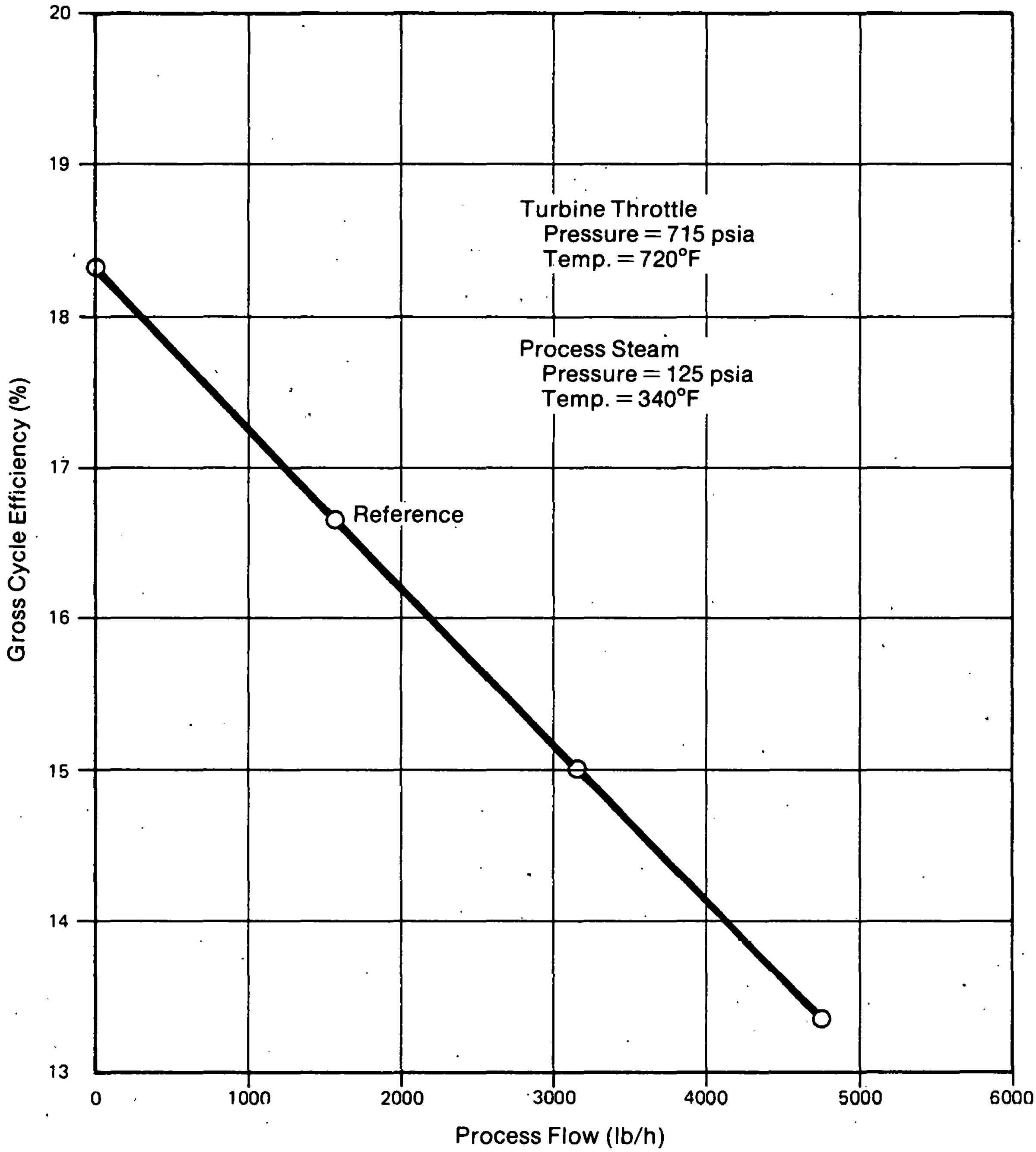


Figure 4-7. Program Element 5 — Total Energy — Steam Rankine Cycle, Cycle Efficiency vs. Process Flow

Figure 4-7 presents the gross cycle efficiency with reference system conditions but varying process heat flows.

The Stirling cycle will be employed for Program Element 6, liquid metal/salt collector/receiver. Data on the conversion cycle efficiencies have not been generated at this time.

4.3.4 Collector Efficiency

Collector efficiency is a function of the receiver temperature and other collector parameters, including concentration ratio, receiver design, location, etc. In the ranking methodology, the thermal storage concepts are evaluated when all factors not affected by storage are maintained. Thus the receiver temperature, which can be affected by storage, is the only variable of interest.

Figure 4-8 presents the ratio of receiver efficiencies as a function of average receiver temperature. The data are the annual average efficiency and were calculated employing the Small Power Systems study computer routine with Barstow insolation. The systems are identified below:

Program Element	Identification
• Water/Steam Collector/Receiver (Point focus central receiver: steam)	PFCR-ST
• Molten Salt Collector/Receiver (Point focus central receiver: salt)*	PFCR-NA
• Liquid Metal Collector/Receiver (Point focus central receiver: sodium)	PRCR-NA
• Gas Collector/Receiver (Point focus central receiver: air Brayton)	PFCR-B
• Organic Fluid Collector/Receiver (Point focus distributed receiver: oil/Rankine)	PFDR-R
• Liquid Metal/Salt Collector/Receiver (Point focus distributed receiver: Stirling)	PRDR-S

For each of the systems, the collector efficiency is plotted as a function of the average temperature of the working fluid. For the sensible heat transport fluids, the average temperature is simply the inlet plus the outlet temperature divided by two. For steam, which has a phase change, the average temperature is calculated as follows:

$$T_{av} = \frac{(1/2)(T_S + T_{in})(h_f - h_{in}) + h_{fg} T_S + (1/2)(T_S + T_o)(h_o - h_g)}{(h_o - h_{in})}$$

*The same data are to be employed for both the molten salt and liquid metal systems since these collectors are very similar.

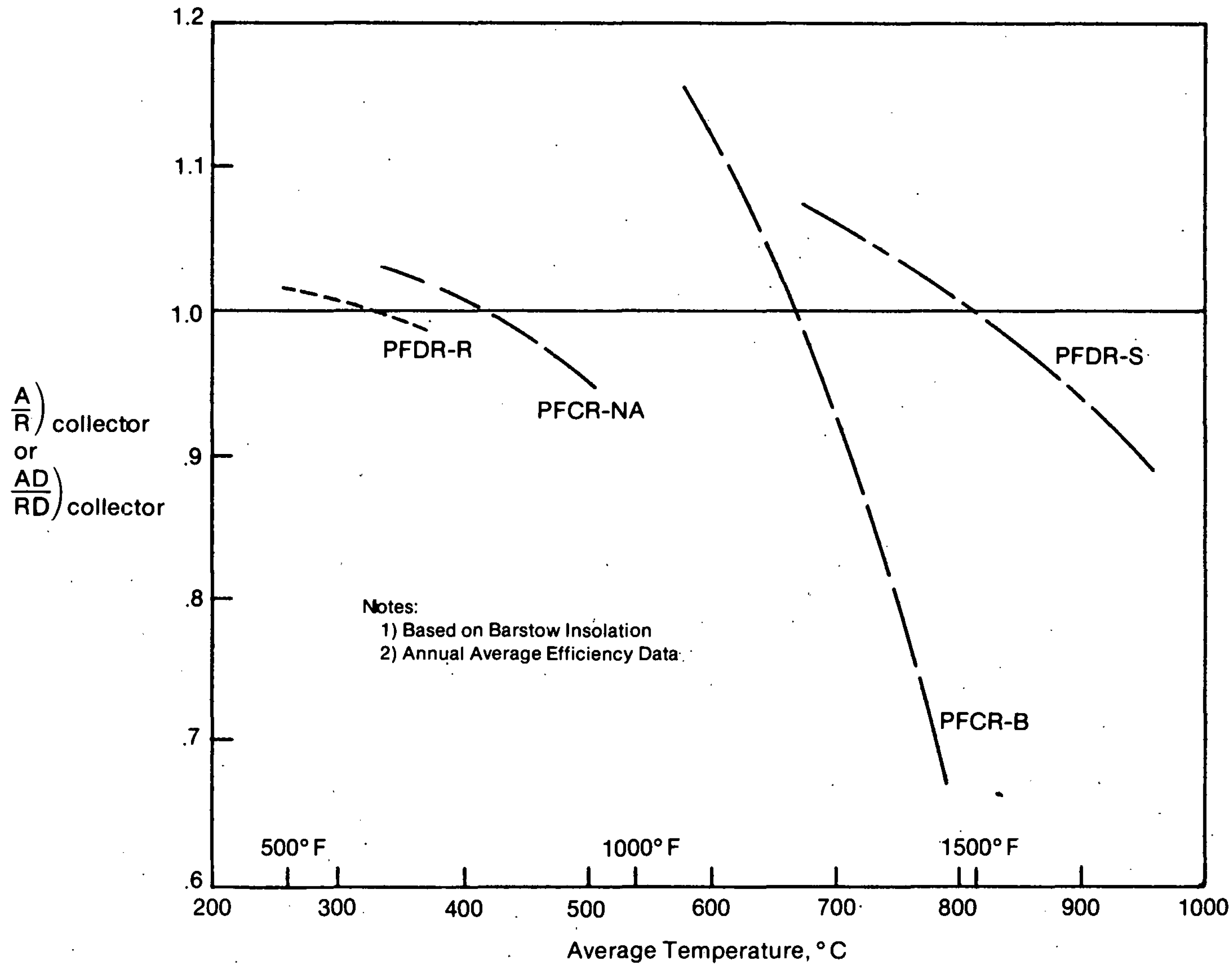


Figure 4-8. Relative Receiver Efficiency

where:

T_{av} = average temperature,
T_{in} = inlet temperature of the subcooled water,
T_S = temperature of the water at saturation for the pressure in the receiver,
T_o = outlet temperature of the superheated steam,
h_{in} = enthalpy of the inlet water,
h_f = the enthalpy of saturated liquid water,
h_{fg} = the enthalpy change of evaporation,
h_g = the enthalpy of the saturated steam,
h_o = the enthalpy of the outlet superheated steam.

The reference conditions for each system presented in Fig. 4-8 are presented below:

TEMPERATURE

System	Inlet °F	Inlet °C	Outlet °F	Outlet °C	Average °F	Average °C
PFCR-B	1000	538	1500	816	1250	675
PFCR-NA	500	260	1100	593	800	426
PFCR-ST[a]	350	177	950	510	622	328
PFDR-S		(Isothermal)	1500	816	1500	816
PFDR-R	406	257	752	400	623	328

[a]1500 psia, T_S = 596.2°F.

To determine the ratio of collector efficiencies, (A/R)COLLECTOR or (AD/RD)COLLECTOR, one first determines the average temperature of the working fluid in the receiver. Then the data are read from Fig. 4-8.

4.3.5 ESR/EDR

ESR/EDR is the ratio of the usable energy delivered from storage to the usable energy delivered direct in the reference system. For a given reference system, this factor is a function of the following mission parameters:

- collector area;
- location;
- quantity of storage (i.e., capacity hours); and
- dispatch strategy.

However, in the use of the ranking methodology, specific knowledge is required only on the collector area and the quantity of storage (because both affect costs). The effects of the location and dispatch strategy are determined by performing the rankings over the expected range of ESR/EDR.

ESR/EDR data are presented in Fig. 4-9. These data were derived from capacity factor data given in Melton (1978). The data were generated for Barstow technology (water/steam collector/receiver) with a 100-MW_e power plant and 1.0 $(10)^6$ m^2 collector area. The dispatch strategy was to meet an evening peak load but otherwise to generate power as thermal energy was received. That strategy is not an extreme; it does not maximize or minimize the use of storage.

The data are plotted as a function of storage capacity in the range of zero to nine hours of storage (electrical). Long-duration storage sufficient to give a unity availability factor is also given. The data for six locations are given with only diurnal storage. Somewhat surprisingly, ESR/EDR is lowest for very sunny locations (Inyokern) and highest for an average area of insolation. The poor locations are between the extremes. This effect is probably due to the dispatch strategy; Ft. Worth has a pronounced evening peak load, and the others have relatively smaller evening peak by comparison. For long-duration storage, the lowest ESR/EDR occurs with the best insolation (Inyokern), and the highest ESR/EDR occurs with the lowest insolation (Seattle). That effect is due to the inherently low capacity factors with low insolation; and thus storage must be used more frequently to achieve a baseload capability.

Work is in progress to generate ESR/EDR data for other reference systems as a function of the mission parameters. The data are not available at this time. Until better data are available, Fig. 4-9 may be used for all systems, since the data are expected to be similar.

4.4 AN EXAMPLE USE OF THE RANKING METHODOLOGY

An example case of the ranking methodology is presented in this section. The example compares two thermal storage concepts in a water/steam collector/receiver in an electric power application. The reference system is Barstow technology. The reference storage concept is an oil/rock thermal storage, requiring a dual-admission steam turbine. The alternative thermal storage concept is a focalized thermal storage concept, allowing a single-admission turbine. Costs for the alternative system were arbitrarily assigned (costs were NOT calculated). Costs were chosen to illustrate the methodology; the data presented are not real and no conclusions on the attractiveness of the alternative thermal storage concepts are possible.

4.4.1 Reference System for the Example

Figure 4-10 presents the schematic for the reference 100-MW_e power plant. Thermal storage subsystem components are shown cross-hatched. The throttle conditions into the turbine are as follows:

- direct: 950°F, 1465 psia,
- from storage: 570°F, 350 psia.

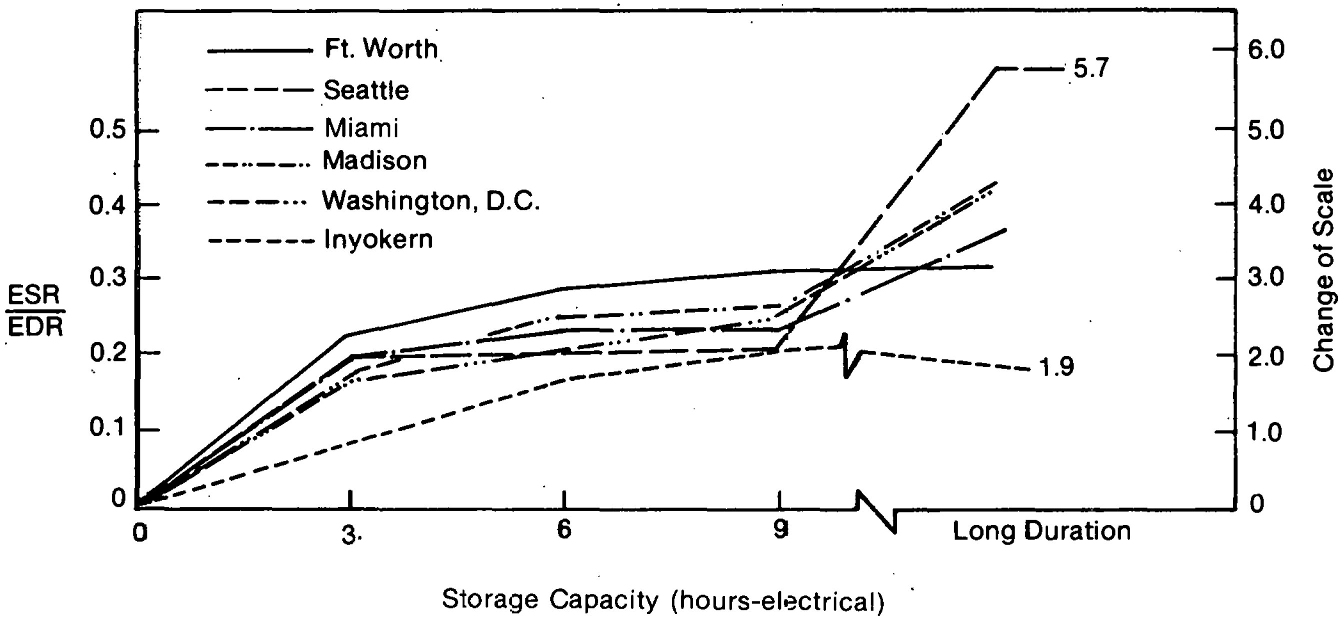


Figure 4-9. ESR/EDR Data Derived for Barstow Technology at 100-MW_e Plant Rating, 1.0 km^2 Collector Area. Data Illustrates Only the Effect of Variable Location

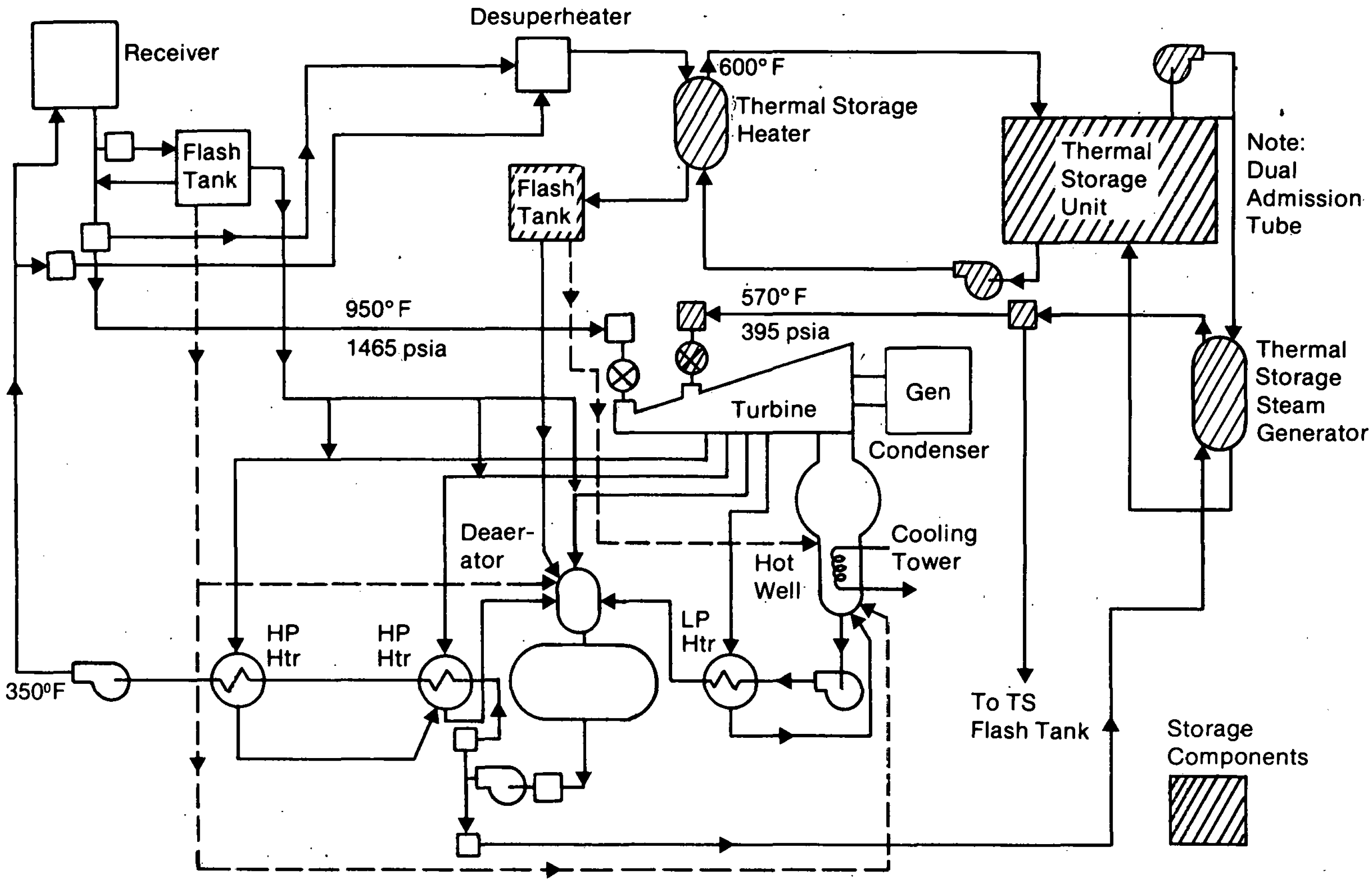


Figure 4-10. Reference System Schematic for the Example

Steam from the receiver at direct throttle conditions is employed to charge storage; it is desuperheated to 600°F to prevent exposure of the oil to high temperatures that would cause decomposition of the oil. Two sets of heat exchanges are required in this system, one to charge storage and one to discharge storage. Because of the low-quality steam that can be generated from storage, a dual-admission turbine is required.

4.4.2 ALTERNATIVE SYSTEM FOR THE EXAMPLE

Figure 4-11 presents the schematic for an alternative 100-MW_e power plant. Thermal storage components are shown cross-hatched. The thermal storage concepts* are assumed to employ a direct contact between the steam and storage media. The media itself becomes the heat exchanger. Furthermore, the media are assumed to have a very large surface area. The steam conditions from storage are thus approximately equal to the steam conditions charging storage. The throttle conditions into the turbine are thus the same (950°F, 1465 psia) direct and from storage. Only a single-admission turbine is required. The steam conditions employed to charge storage are 950°F, 1500 psia; a high-pressure steam is required to provide the temperature difference for heat transfer. Table 4-3 presents the assumed performance and cost parameters for this example. The 1st Law efficiencies were assumed to be equal. The 2nd Law efficiciencies were taken from Melton (1978). Receiver temperatures are slightly different, but the collector efficiencies were assumed to be equal. Costs for the reference systems were taken from Melton (1978) since the standard data were not completely assembled at the time of the analysis. The alternative storage system costs were assigned arbitrarily. The solar thermal system costs were assumed to be equal. There should be a cost differential for a dual-admission turbine versus a single admission, but turbine costs were assumed as equal.

4.4.3 <u>Calculating the Ranking Index</u>

The calculation of the Ranking Index requires both cost and performance data. Table 4-4 presents capital and O&M costs for one mission. The total capitalization is the capital cost plus the capitalized equivalent of the levelized O&M costs. Employing cost data from Westinghouse (1978) and Table 4-3, total capitalization costs were determined as a function of the collector area, storage capacity, and storage concept. The data are presented in Table 4-5.

The Ranking Index is given by the following equation:

$$RI = \frac{\dfrac{(CC)\,A}{(CC)\,R} \bullet \left[1 + \dfrac{ESR}{EDR}\right]}{\left[\dfrac{A}{R}_{RT} \bullet \dfrac{A}{R}_{CYCLE} \bullet \dfrac{A}{T}_{COLLECTOR} \bullet \dfrac{ESR}{EDR}\right] + \left[\dfrac{AD}{RD}_{CYCLE} \bullet \dfrac{AD}{RT}_{COLLECTOR}\right]}.$$

*This concept is highly idealized and may not be practiced to build in a commercial system.

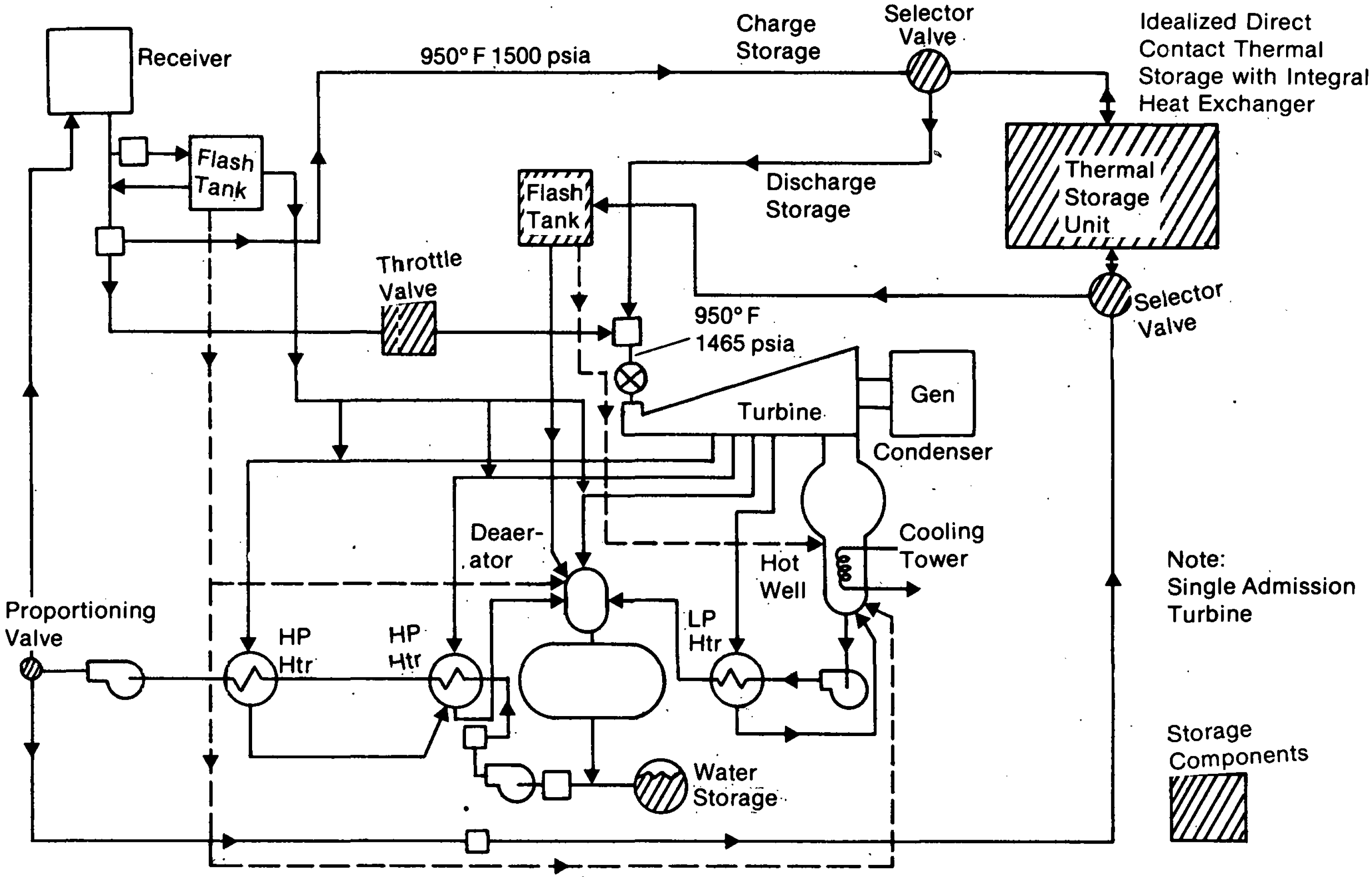


Figure 4-11. Alternative System Schematic for the Example

Table 4-4. REFERENCE SOLAR THERMAL STAND-ALONE PLANT COST[a] FOR THE EXAMPLE 1976 M\$.

(100-MW_e Plant; 700,000 m^2 Collector, 3-h storage, electrical; Barstow Technology)

Land	0.6
Structures and Facilities	4.7
Heliostats	56.0
Receiver and Tower	20.0
Steam Boiler Equipment	
Storage	21.0
Turbine	15.0
Electric Plant Equipment	2.3
Miscellaneous	1.0
Total Direct	120.6
Contingency and Spares	18.1
Indirect Costs (10%)	12.1
Subtotal	150.8
Interest During Construction 15% (IDC)	22.6
Total Capital	173.4
O&M -3%/year of Capital -17% FCR -1.8 Levelizing Factor	55.1
CC (R), Total Capitalization	228.5

[a]From Westinghouse, July 25, 1978.

Table 4-5. COST DATA FOR THE EXAMPLE

Storage Capacity Hours; Thermal (R and A)	Total Capitalized Cost (1976 \$, M)[a]					
	A = 500,000 m^2		A = 700,000 m^2		A = 1,000,000 m^2	
	(CC) R	(CC) A	(CC) R	(CC) A	(CC) R	(CC) A
1.38	196.8	189.1	225.2	217.9	263.0	255.3
4.14	199.7	221.1	228.5	249.9	265.9	287.3
8.27	207.6	269.2	236.4	298.0	273.8	335.4
12.41	214.1	317.2	242.9	346.0	280.3	383.4

[a]All capitalized O&M assumed at \$55.1M regardless of area, storage capacity, or storage material.

Substituting the values from Table 4-3 the following equation is obtained for this example:

$$RI = \frac{\frac{(CC)\,A}{(CC)\,R} \bullet \left[1 + \frac{ESR}{EDR}\right]}{\left[1.307 \bullet \frac{ESR}{EDR}\right] + 1} .$$

This equation requires ESR/EDR data. From Fig. 4-9, the maximum (i.e., high) and minimum (low) magnitudes of ESR/EDR as a function of storage capacities were employed regardless of location.

Figure 4-12 presents the results of the calculations. The graph plots the ranking index as a function of storage capacity for three collector areas and the maximum and minimum ESR/EDR for each storage capacity (solid and dashed lines respectively). For RI greater than one, the busbar energy cost of the alternative is higher than the reference system; BBEC(A) is lower when RI is less than one. The line noting equal capital costs is the storage capacity at which the total capitalized costs are equal. RI is less than one at this storage capacity since the alternative is more efficient and delivers more energy. Although the costs are higher for more than 2.1 hours of storage, the alternative still has a lower BBEC until about three hours. For large storage capacities the higher costs outweigh the performance advantage.

By inspection of Fig. 4-12, there are conditions at which A is the preferred concept and others at which the reference is best. The choice of the best system to be developed is not obvious. The choice may well be that both concepts should be developed, one for small-capacity storage and the other for high-capacity storage. The choice depends upon the characteristics desired by the users, the costs and value of the storage-coupled solar thermal system, program emphasis, resources, and other factors.

4.5 VALIDATION OF THE RANKING METHODOLOGY

The ranking methodology compares thermal storage concepts when the only variables are the concepts. An alternative approach is to compare the concepts when each delivers exactly the same quantity of energy at the same nameplate rating. The latter approach has been employed in previous trade-offs of thermal storage concepts (Sandia 1977). The results for the two approaches are compared in this section. Section 4.5.1 describes the basic differences in the two approaches. Section 4.5.2 presents the concepts analyzed in this comparison. Section 4.5.3 presents the results of the comparison.

4.5.1 Two Approaches for Comparing Thermal Storage Concepts

The two approaches are herein called (1) constant thermal and (2) constant electrical:

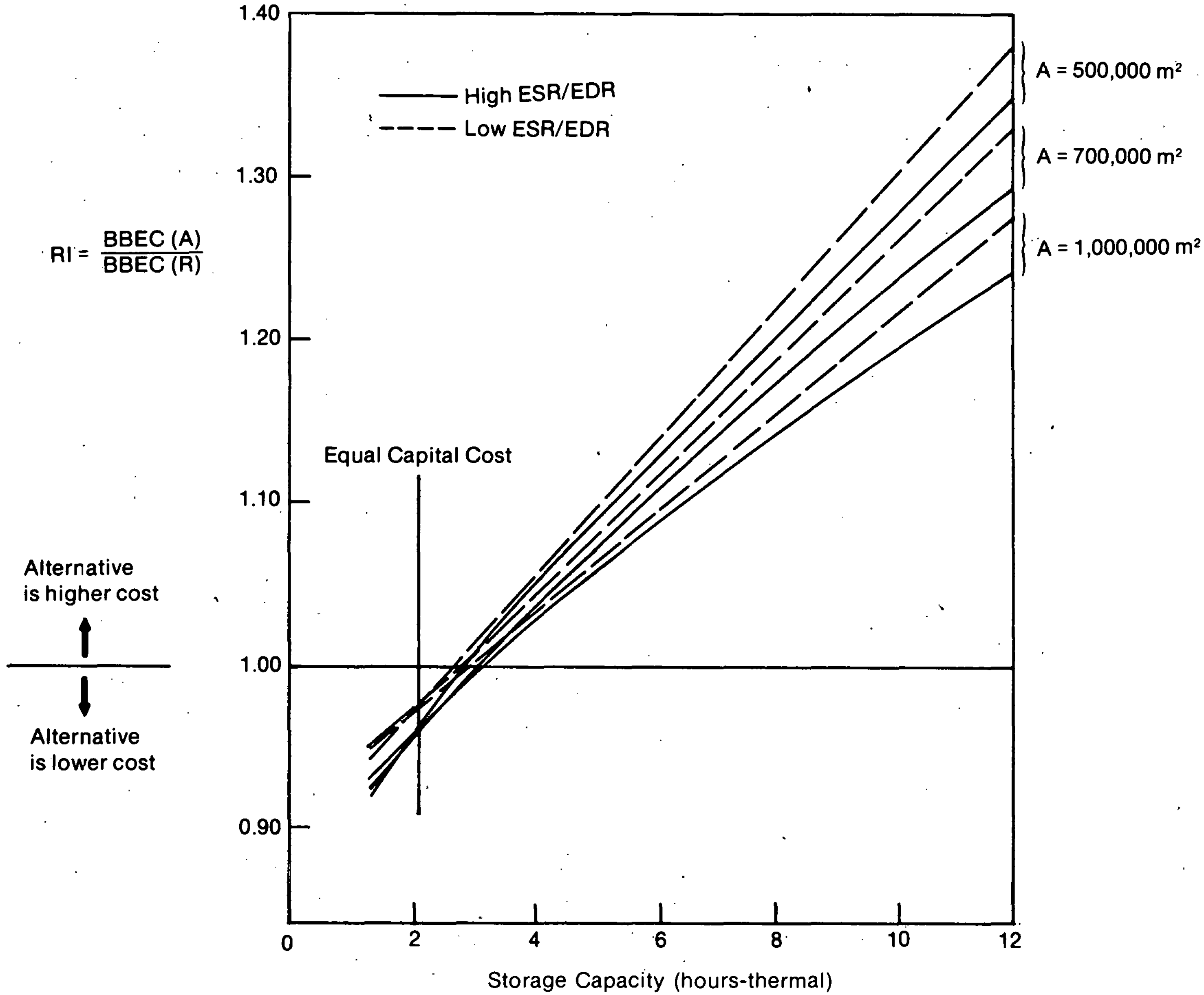


Figure 4-12. RI Data for the Example

1. Constant Thermal is the ranking methodology previously described. Key feature: the concepts are compared when all have the same thermal rating and capacity;

2. Constant Electrical is the approach that was employed by Sandia (1977). Key feature: the concepts are compared when all have the same electrical rating and capacity;

where the definitions of thermal and electrical storage capacities are as previously described.

The constant thermal approach requires a reference solar thermal system and thermal storage concept. The reference may be any one of the alternatives without loss of generality. The concepts are compared on a bus bar energy cost [BBEC] basis. The cost of each alternative is calculated as a ratio to the BBEC of the reference system; the ratio is designated as the Ranking Index RI. In this approach, the collector field area is constant; the heat transfer rates of charging and delivery are equal; and the quantities of thermal energy in storage are all equal. The nameplate electrical rating and storage capacity are not necessarily equal, due to efficiency differences. The quantity of electrical energy delivery is therefore not necessarily equal, but this effect is included in the BBEC calculation.

The constant electrical approach compares thermal storage concepts, when each delivers the same quantity of electric energy, at the same nameplate rating (electrical) from storage. Because of efficiency differences between concepts, the collector field, receiver, heat exchangers, etc., are in general not equal; for each concept, the whole system must be individually optimized for each location and dispatch strategy. This procedure requires relationships for all major items in the solar thermal system as a function of size, and hour-by-hour simulation data for the insolation at each site. The absolute value of the BBEC is calculated for each concept considered. Therefore, the constant electrical approach does not require a reference system; the approach does allow selection of one of the alternatives as a reference. BBEC ratio can thus be calculated in precisely the same manner as in the other approach.

An example illustrates the differences in the two approaches. Consider a comparison of three concepts (designated as A, B, and C) for a condition of three hours of storage and a given location and dispatch strategy. Assigning a different efficiency to each concept (high, medium, and low, respectively), B is designated as the reference system. With B the collector area is 1.0 X; and the two concepts A and C have 10% better and 10% less overall efficiency of storage. The conditions for this example are summarized in Table 4-6; these data are representative (but not precise). Concept B is the same in both approaches, since it is the reference system. For the other two concepts, the collector area, and quantity of storage (thermal and electrical), are all different. The conditions are very close for a given concept in both approaches, and the resulting BBECs will also be close, but obviously not precisely equal.

The primary difference in the two approaches is a slight variation in the conditions of comparison. Very close to equal BBECs will be calculated, and no difference in conclusions will occur. These facts will be demonstrated with real data in the following section.

Table 4-6. EXAMPLE COMPARISON OF CONDITIONS

Concept	Constant Thermal Approach	Constant Electrical Approach
Concept A (High Efficiency)		
Collector Field	1.0 X	0.97 X
Hours, Thermal	3.0	2.7
Hours, Electrical	3.3	3.0
Concept B (Reference)		
Collector Field	1.0 X	1.0 X
Hours, Thermal	3.0	3.0
Hours, Electrical	3.0	3.0
Concept C (Low Efficiency)		
Collector Field	1.0 X	1.03 X
Hours, Thermal	3.0	3.3
Hours, Electrical	2.7	3.0

4.5.2 Thermal Storage Concepts for the Validation

A validation of the ranking methodology requires a comparison of results when the only variables are those associated with the approach. Data for comparing concepts with constant electrical capacity exist (Sandia 1977). Three concepts are considered; each is briefly described below:

1. Reference System: MDAC

   A single-stage dual-media thermocline system, employing oil (Caloria) and rocks (granite and sand) for storage.

2. Martin

   A two-stage system: Hitec in a two-tank system forms the high-temperature stage; oil (Caloria) in a two-tank system forms the low-temperature stage.

3. Honeywell

   A two-stage system: Hitec in a two-tank system forms the high-temperature stage; a dual-media oil (Caloria) and rock (crushed granite) in a thermocline tank forms the low-temperature stage.

All three concepts are candidates for a 100-MW_e (operating direct) central receiver system employing a water/steam system. Each of the thermal storage concepts requires a dual-admission turbine that is assumed to have the same cost regardless of steam conditions.

The ratio of the overall efficiency of each of the concepts is as follows:

	(NA)/(NR)
MDAC	1.0 (Reference)
Martin	1.14
Honeywell	1.12

when (NA)/(NR) is the product of the 1st Law, cycle, and receiver efficiencies. Because of the performance difference, the collector areas are also different in the constant electrical approach; also, the nameplate ratings from storage are different in a constant thermal approach.

These data, and the cost of thermal storage data, are presented in Table 4-7 for seven hours of storage with both approaches. Note that the MDAC data are identical in both approaches, and thus the BBEC of the MDAC must be the same. This fact allows the calculation of BBEC of any concept with the ranking methodology (simply by multiplying the Ranking Index by the BBEC of the reference, MDAC, system).

Table 4-7. COST AND PERFORMANCE DATA FOR THE COMPARISON OF THE TWO APPROACHES: THERMAL STORAGE COSTS FOR SEVEN HOURS OF STORAGE

	Cost[a]	Discharge Rate	Thermal Storage Capacity	Collector Area
Sandia (1977) Data Seven Hours Electrical at 70 MW_e				
Honeywell	\$38.54 M	251 MW_T	1757 MWh_T	830,000 m^2
Martin	\$79.46 M	251 MW_T	1757 MWh_T	830,000 m^2
MDAC	\$26.34 M	286 MW_T	2002 MWh_T	900,000 m^2
SERI Data Seven Hours Thermal				
Honeywell (80 MW_e)	\$43.92 M	286 MW_T	2002 MWh_T	900,000 m^2
Martin (80 MW_c)	\$90.55 M	286 MW_T	2002 MWh_T	900,000 m^2
MDAC (70 MW_e)	\$26.34 M	286 MW_T	2002 MWh_T	900,000 m^2

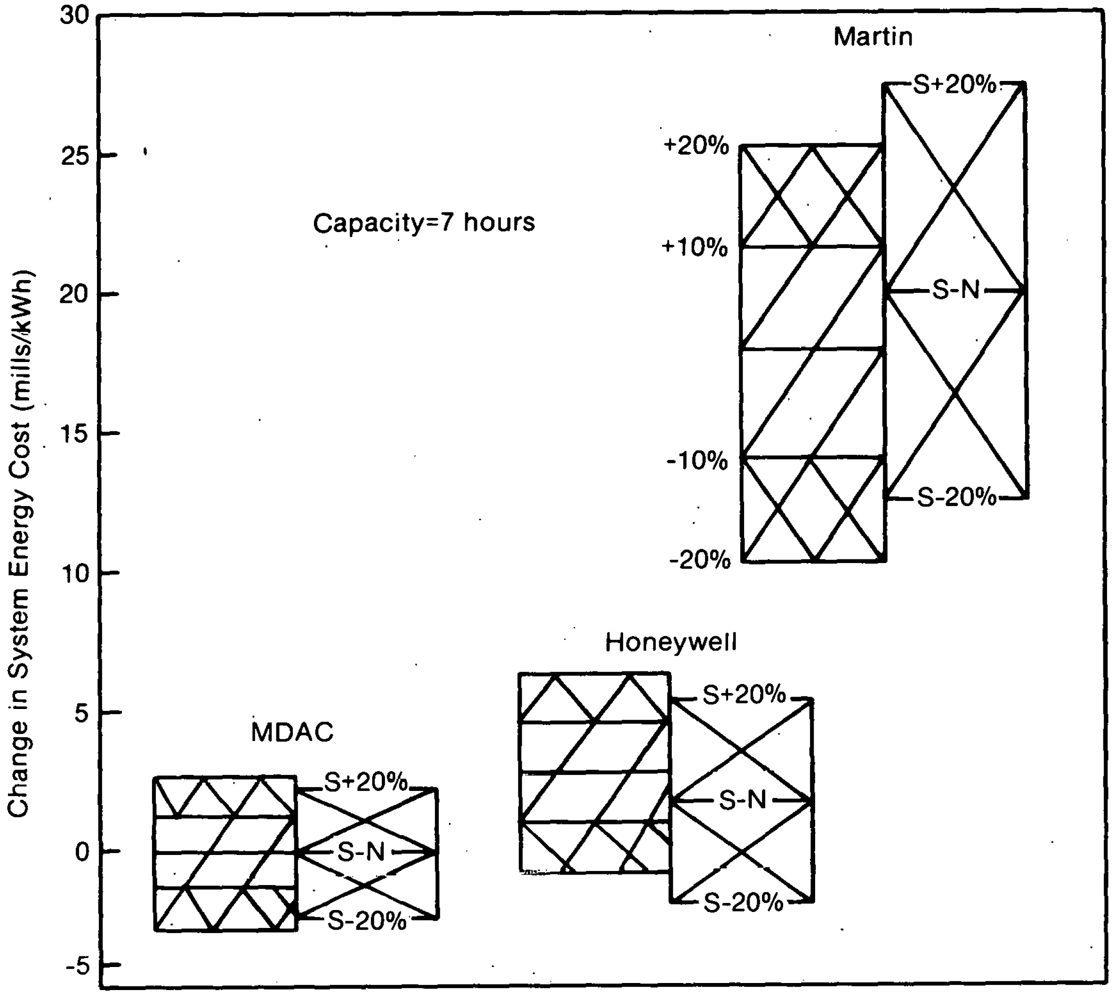
[a]O&M excluded: it was assumed to be a percentage of direct and nondirect costs.

4.5.3 Results of the Validation

Figure 4-13 presents the change in BBEC for the three concepts by both approaches. The change in BBEC is the difference between the cost for the alternative less the cost of the MDAC reference system. The results for constant electrical storage are on the left; and the results for constant thermal storage are on the right for each concept. The maximum and minimum values for each concept are the cost differentials due to a ±20% uncertainty in the cost of the thermal storage subsystem.

The results for the two approaches are very close. Some differences in the data exist; these differences are 0.5% to 1.5% of the reference system BBEC (0.5 to 2.5 mills/kWh$_e$). The conclusions based on each approach are identical, even when considering the effect of the cost uncertainties of each thermal storage concept.

This case demonstrates no difference in the conclusions for either approach. Based on this fact, and the basic similarities of the two approaches, either one is adequate, and the two are essentially equivalent for the purpose of identifying promising thermal storage concepts.



1. 1st Commercial Plant Cost = 168 mills/kWh$_e$, MDAC Storage
2. S Denotes SERI Data; the Other Data Are from Sandia (1977)

Figure 4-13. Results of the Validation Analysis

SECTION 5.0

CONCLUSIONS AND RECOMMENDATIONS

5.1 CONCLUSIONS

Value data for solar thermal systems have been analyzed and the value of thermal storage has been calculated. Thermal storage value is calculated as a function of collector area, location, and storage capacity. The collector area has a very strong influence on thermal storage value. The collector area that provides the most favorable cost/benefit for the storage-coupled solar thermal system was deemed to be the most appropriate.

Thermal storage value data have been determined for buffer, diurnal, and long-duration storage in electric utility applications of solar thermal. The buffer storage value is very high; establishing buffering thermal cost goals by obtainable cost is more appropriate, since the value is likely to be higher than the cost of all reasonable thermal storage candidates. Diurnal thermal storage values in electric utility applications are determined by a remix analysis method employing Westinghouse and Aerospace data. The results are strongly dependent upon the fuel-price scenario and location. Values data are presented for high-, medium-, and low-insolation locations. A very preliminary analysis is performed for the value of long-duration storage. The analysis employs a next-plant decision method for a new baseload plant. Long-duration storage value is calculated by comparing two cases, one of them with long duration. Value of long-duration storage is shown to be a strong function of efficiency, with a relatively small effect of location.

A method of ranking thermal storage is derived. The methodology is capable of evaluating thermal storage concepts over a wide range of mission parameters, including:

- collector area,
- storage capacity,
- plant locations,
- dispatch strategy, and
- solar thermal system.

The ranking is based upon unit energy costs; no significant change in rankings is expected if the rankings are based upon cost/benefit ratio. The methodology requires that thermal storage concepts be compared when all have the same ability to transfer and store thermal energy; and also each concept is employed in the same solar thermal systems with identical mission parameters. These mission parameters are evaluated over the expected range to identify the most attractive thermal storage concepts. An example case is evaluated to illustrate the use of the ranking methodology. As indicated in the example, no one thermal storage concept is anticipated to be best in all cases. The choice of storage concepts to develop is dependent upon the intended use of the storage-coupled solar thermal system and the thermal storage program emphasis. The ranking methodology was validated by comparing it with another approach.

5.2 RECOMMENDATIONS

Table 5-1 presents recommended cost goals for thermal storage. These data were determined based on the value data presented in Section 3.0 of this report. The diurnal data are neither the value of thermal storage based upon the Westinghouse data nor the Aerospace data. Rather the recommended cost goals are approximately the average of the values from those two sources and rounded off to a multiple of five.

The cost goals are based upon value that is determined by the cost of alternative fuels and generating equipment. The cost goals are thus dependent upon the fuel price estimates. A low-price fuel scenario of oil, gas, coal, etc. would demand lower cost goals for thermal storage; a high-price fuel scenario would allow higher cost goals. The data on Table 5-1 reflect a relatively low price fuel scenario. The data were derived for Barstow technology and calculated for plant start-up in the late 80s/early 90s. Cost goals are presented for three levels of insolation, i.e., plant locations. The cost goals are based on about a 1990 plant start-up with small market penetration. For later start-up dates, these goals are thus very conservative, since fossil fuel prices in the long term will be much higher than those assumed in the calculations. An annual real escalation rate of 2% is recommended for plant start-up at later dates.

The long duration cost goals are calculated for a stand-alone baseload electric power plant. The analysis employed a next-plant analysis method; a remix analysis is expected to show lower cost goals. There is a very strong dependency of the cost goal with storage efficiency. The effect is due to the value of the energy which is lost with low storage efficiencies. The storage capacity required to achieve a baseload capability is not known. The range is expected to be between 40 and 300 hours. Additional study to identify a narrower range is recommended.

Table 5-1. RECOMMENDED COST GOALS FOR THERMAL STORAGE IN SOLAR THERMAL ELECTRIC PLANTS, 1976 $\$/kW_e$[a]

Quantity of Storage	High Insolation	Medium Insolation	Low Insolation
Buffer		established by obtainable cost	
Diurnal			
3 Hours	255	120	60
6 Hours	300	180	90
9 Hours	—	225	110
Long Duration			
62% Efficiency	230	210	180
100% Efficiency	695	620	515

[a]Costs include the overall costs due to (1) storage power related cost, (2) storage energy related costs, and (3) operations and maintenance (+ replacement) costs. The costs also include nondirect factors for (A) contingency and spares (15%), (B) indirects (10%), and (C) interest during construction (15%) for an overall factor (O.F.) of 1.44 = [(1 + 0.15 + 0.10) x (1.15)]; i.e., divide by 1.44 to determine allowable direct installed cost for the system.

SECTION 6.0

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APPENDIX A

DERIVATION OF THE RANKING FUNCTION

The derivation of the quantitative expression of the ranking index is described in this appendix. The ranking function contains both cost factors and performance factors and provides a quantitative relationship of their importance. The methodology allows the consideration of several missions in the ranking and does so with a minimum of work by the user.

A.1 ASSUMPTIONS

The ranking methodology requires several assumptions for its derivation. These assumptions effectively are constraints on its use. The key items are presented in the following paragraphs.

A.1.1 Fixed Plant Parameters (Mission)

The approach is to define a mission, including a reference solar thermal system and storage concept, R. All alternative concepts, A, are evaluated by replacing the reference storage concept, keeping all other parameters constant. Note, the process is repeated for every mission the user desires so that the ranking may be conducted over a wide range of collector area, storage capacity, locations, dispatch strategy, etc.

A.1.2 Same Storage Capacity, Thermal

The performance and cost data must be analyzed when all concepts have the same capacity to receive, deliver, and store thermal energy. Thus, all concepts can be dispatched in precisely the same manner. The "Btu bucket" must be the same size in all cases, as well as the rates of charging and discharging.

A.1.3 Same Dispatch Strategy

The manner in which the solar thermal plant is dispatched must be the same for all concepts. In particular, the quantity of insolation delivered to storage is equal. Furthermore, since the collector area is constant, the total insolation available is constant. These terms, Q_S and Q_{SS}, are defined below:

$$Q_S = \text{TOTAL INSOLATION}\big|_A = \text{TOTAL INSOLATION}\big|_R \, ,$$

and

$$Q_{SS} = \text{INSOLATION FOR STORAGE}\big|_A = \text{INSOLATION FOR STORAGE}\big|_R \, ,$$

and

$$Q_S - Q_{SS} = \text{INSOLATION FOR DIRECT USE}\big|_A = \text{INSOLATION FOR DIRECT USE}\big|_R \, ,$$

where the last equation expresses the quantity of insolation that is used directly to deliver usable energy (i.e., not through storage).

A.2 MATHEMATICAL DERIVATION

The ranking index can be expressed mathematically. In this section, the equations are derived. The Ranking Index is the ratio of unit energy costs and is given below:

$$RI(A) = \frac{\text{BBEC (A)}}{\text{BBEC (R)}}$$

Rearranging:

$$RI(A) = \frac{\dfrac{\text{(COST) A}}{\text{(ENERGY) A}}}{\dfrac{\text{(COST) R}}{\text{(ENERGY) R}}} = \frac{\dfrac{\text{(COST) A}}{\text{(COST) R}}}{\dfrac{\text{(ENERGY) A}}{\text{(ENERGY) R}}} \qquad \text{(A-1)}$$

and

$$RI(A) = \frac{\dfrac{\text{(CC) A}}{\text{(CC) R}}}{\dfrac{\text{EA}}{\text{ER}}} \qquad \text{(A-2)}$$

where

(COST) R	=	annual cost of operating the reference system [$/year];
(COST) A	=	(COST) R adjusted for new storage element [$/year];
(ENERGY) A or EA	=	total usable energy delivered from the solar thermal plant with A as storage [kWh_e/year];
(ENERGY) R or ER	=	total usable energy delivered from the solar thermal plant with R, the reference storage [kWh_e/year];
(CC) R	=	the capitalized cost of the reference system ($/$kW_e$), the equivalent capital cost for equipment, operations and maintenance (O&M), and fuel costs; and
(CC) A	=	the capitalized cost of the alternative system ($/$kW_e$).

The numerator of Eqs. A-1 and A-2 contains only cost factors, which are discussed in Section A.2.1. The denominator contains only performance factors, which are discussed in Section A.2.2.

A.2.1 Evaluation of Costs

The calculation of costs must be for the whole storage-coupled solar thermal system. These costs include the equipment capital cost, operations and maintenance (O&M), and fuel costs. Following the EPRI methodology (Day 1978), the annualized costs are calculated as follows:

$$\text{COST (\$/YEAR)} = \begin{array}{c}\text{LEVELIZED}\\ \text{CAPITAL}\\ \text{COST}\end{array} + \begin{array}{c}\text{ANNUAL}\\ \text{LEVELIZED}\\ \text{O\&M}\\ \text{COST}\end{array} + \begin{array}{c}\text{ANNUAL}\\ \text{LEVELIZED}\\ \text{FUEL}\\ \text{COST}\end{array} \tag{A-3}$$

$$= \text{FCR} \times \begin{bmatrix}\text{CAPITAL}\\ \text{COST}\end{bmatrix} + \text{O\&M} + \text{FUEL}$$

where

$$\text{FCR} = \text{FIXED CHARGE RATE } \frac{\$}{\$\text{ - YEAR}}$$

$$\begin{array}{c}\text{CAPITAL}\\ \text{COST}\end{array} = \begin{array}{c}\text{INITIAL COST OF ALL PLANT}\\ \text{EQUIPMENT .}\end{array}$$

The levelizing procedure is presented in EPRI (1978).

An equivalent capitalized cost can be calculated for the annualized cost. The capitalized cost is calculated by dividing the annualized cost by the fixed charge rate as follows:

$$\text{CC (\$)} = \frac{\text{COST (\$/YEAR)}}{\text{FCR (\$/\$-YEAR)}} \tag{A-4}$$

$$= \begin{array}{c}\text{CAPITAL}\\ \text{COST}\end{array} + \frac{\text{O\&M}}{\text{FCR}} + \frac{\text{FUEL}}{\text{FCR}}$$

$$= \begin{array}{c}\text{CAPITAL}\\ \text{COST}\end{array} + \begin{array}{c}\text{CAPITALIZED}\\ \text{O\&M}\end{array} + \begin{array}{c}\text{CAPITALIZED}\\ \text{FUEL}\end{array}$$

Clearly, the ratio of costs is equivalent regardless of which costing approach is employed, since the following equation is obviously valid:

$$\frac{(\text{COST})\,A}{(\text{COST})\,R} = \frac{\dfrac{(\text{COST})\,A}{\text{FCR}}}{\dfrac{(\text{COST})\,R}{\text{FCR}}} = \frac{(\text{CC})\,A}{(\text{CC})\,A} \,. \tag{A-5}$$

In this analysis the capitalized cost approach is employed.

Table A-1 presents the approach to calculation of costs. Note that nondirect factors are included in the calculation of total capital costs. For each reference system (which are defined in following paragraphs), a complete set of cost data are supplied. For items that are significant parameters (i.e., collector area), the parametric relationship is also supplied. For all items not affected by thermal storage, the user simply employs the reference system data. For items affected by the choice of alternative storage concepts, the user supplies the appropriate data. Those latter items include the thermal storage costs and other items (e.g., turbine generator, single admission versus dual admission).

A.2.2 Evaluation of EA/ER

The quantity of usable energy delivered by a storage-coupled solar thermal system is the sum of energy from storage and the energy direct. The ratio of energy deliveries with the alternative storage concept (EA) to that of the reference concept (ER) is calculated as follows:

$$\frac{(\text{ENERGY})\,A}{(\text{ENERGY})\,R} = \frac{(\text{ENERGY FROM STORAGE})\,A + (\text{ENERGY DIRECT})\,A}{(\text{ENERGY FROM STORAGE})\,R + (\text{ENERGY DIRECT})\,R}$$

$$= \frac{EA}{ER} = \frac{ESA + EDA}{ESR + EDR} \,, \tag{A-6}$$

where

ESA = energy from storage, with A,

EDA = energy direct, with A,

ESR = energy from storage, with R,

EDR = energy direct, with R.

Table A-1. COST CALCULATIONS

Item	Reference System, R, Costs ($)	Alternative System, A, Costs ($)
1. Solar Collector	Supplied for Reference System (See Appendix B)	Costs Change Only as Affected by the New Thermal Storage System
2. Transport		
3. Thermal Storage $(MBtu)_T$		
4. Turbine/Generator		
5. Controls		
6. BOP		
7. Subtotal (Directs)		
8. Nondirects[a] at ___% of 7		
9. Total Capital		
10. Capitalized O & M		
11. Capitalized Fuel		
12. Capitalized Total Cost (CC)		

[a]Nondirects include contingency and spares; indirects; and interest during construction.

Rearranging:

$$\frac{EA}{ER} = \frac{\left(\frac{ESA}{ESR} \bullet \frac{ESR}{EDR}\right) + \frac{EDA}{EDR}}{\frac{ESR}{EDR} + 1} \quad . \qquad (A-7)$$

The above equation requires three items: ESA/ESR, EDA/EDR, and ESR/EDR. The first two are functions of the alternative storage concept. The third, ESR/EDR, is a function only of the reference system; data on that factor are supplied for the user.

A.2.2.1 ESA/ESR

ESA/ESR is the ratio of the energy delivered from storage A to R. The quantity of usable energy can be calculated as follows:

ENERGY FROM STORAGE

= STORAGE EFFICIENCY (i.e., 1-LOSSES) • POWER CYCLE EFFICIENCY OPERATING FROM STORAGE • COLLECTOR EFFICIENCY

• INSOLATION USED FOR STORAGE • UTILIZATION FACTOR FOR STORAGE .

That is

$$\text{ENERGY FROM STORAGE} = (\eta RT)(\eta CYCLE)(\eta COLLECTOR)(Q_{SS})(UF_S) \quad , \qquad (A-8)$$

where the utilization factor accounts for energy that can not be stored because the storage is full or the insolation level is too low to merit its collection.

Applying the above, the following equation is obtained:

$$\frac{ESA}{ESR} = \frac{[(\eta RT) \bullet (\eta CYCLE) \bullet (\eta COLLECTOR) \bullet Q_{SS} \bullet UF_S]\ (A)}{[(\eta RT) \bullet (\eta CYCLE) \bullet (\eta COLLECTOR) \bullet Q_{SS} \bullet UF_S]\ (R)} \quad . \qquad (A-9)$$

Since the plants are being dispatched in the same manner and the collector areas are equal, the utilization factors and insolations are equal; i.e.,

$$UF_S\,(A) = UF_S\,(R)$$

and

$$Q_{SS}\,(A) = Q_{SS}\,(R)$$

These factors divide out and the equation simplifies to the following:

$$\frac{ESA}{ESR} = \frac{(\eta RT) \bullet (\eta CYCLE) \bullet (\eta COLLECTOR)A}{(\eta RT) \bullet (\eta CYCLE) \bullet (\eta COLLECTOR)R} = \frac{A}{R}_{RT} \bullet \frac{A}{R}_{CYCLE} \bullet \frac{A}{R}_{COLLECTOR} \tag{A-10}$$

where

$\frac{A}{R}_{RT}$ = the ratio of the 1st law efficiencies of the two thermal storage concepts,

$\frac{A}{R}_{CYCLE}$ = the ratio of the conversion cycle efficiencies for the two thermal storage concepts when operating through storage,

$\frac{A}{R}_{COLLECTOR}$ = the ratio of the solar collector efficiencies for the two thermal storage concepts when charging storage.

The 1st Law efficiency accounts for losses of storage. Included in this item are heat leakages (through insulation tanks, lines, etc.), heat rejections (nonrecoverable losses such as periodic stabilizing thermoclines) and the thermal equivalent of work inputs (compressor or pump work to charge and/or discharge storage).

(A/R)(CYCLE) accounts for the usable energy that can be delivered through the thermodynamic cycle of the system when operating through storage. If the cycle working-fluid temperatures and pressures are the same as the reference system, this item will be unity. However, the various thermal storage concepts do have different temperature limits, which will affect the operating conditions of the cycle. This item quantitatively expresses the importance of that effect.

(A/R)(COLLECTOR) accounts for the difference in collected thermal energy from the receiver. Since receiver temperatures may be affected by the choice of thermal storage, the receiver efficiency will also be affected.

A.2.2.2 EDA/EDR

EDA/EDR is the ratio of the the energy delivered direct (not through storage) A to R. The quantity of usable energy can be calculated as follows:

$$\text{ENERGY DIRECT (without use of storage)} = \text{POWER CYCLE EFFICIENCY OPERATING DIRECT} \bullet \text{COLLECTOR EFFICIENCY} \bullet \text{INSOLATION USED DIRECT} \bullet \text{UTILIZATION FACTOR FOR DIRECT.}$$

Thus

$$\frac{EDA}{EDR} = \frac{[(\eta CYCLE) \bullet (\eta COLLECTOR) \bullet (Q_S - Q_{SS}) \bullet UF_D]\ (A)}{[(\eta CYCLE) \bullet (\eta COLLECTOR) \bullet (Q_S - Q_{SS}) \bullet UF_D]\ (R)} . \quad \text{(A-11)}$$

Since the plants are being dispatched in the same manner and the collector areas are equal, the utilization factors and insolations are equal, i.e.:

$$UF_D\ (A) = UF_D\ (R)$$

and

$$(Q_S - Q_{SS})\ (A) = (Q_S - Q_{SS})\ (R).$$

The equation simplifies to the following:

$$\frac{EDA}{EDR} = \frac{[(\eta CYCLE) \bullet (\eta COLLECTOR)]\ (A)}{[(\eta CYCLE) \bullet (\eta COLLECTOR)]\ (R)} = \frac{AD}{RD}_{CYCLE} \bullet \frac{AD}{RD}_{COLLECTOR} \quad \text{(A-12)}$$

where

$\frac{AD}{RD}_{CYCLE}$ = the ratio of the conversion cycle efficiencies for the two concepts when operating direct,

$\frac{AD}{RD}_{COLLECTOR}$ = the ratio of the solar collector efficiencies for the two thermal storage concepts when operating direct.

A.2.2.3 ESR/EDR

ESR/EDR is the ratio of the usable energy delivered from storage in the reference system to the usable energy delivered direct in the reference system. The factor is only a function of the reference system. The factor is affected by the collector area, location, quantity of storage, dispatch strategy, and reference system performance factors (i.e., efficiencies). Data for this factor are supplied to the user of the ranking methodology.

A.2.3 The Ranking Function

The Ranking Index is calculated by the ranking function, which is given by Eq. A-2:

$$\mathrm{RI\,(A)} = \frac{\dfrac{\mathrm{(CC)\,A}}{\mathrm{(CC)\,R}}}{\dfrac{\mathrm{EA}}{\mathrm{ER}}} \tag{A-2}$$

Substituting expressions from Eqs. A-7, A-10, and A-12, the following is obtained:

$$\mathrm{RI\,(A)} = \frac{\dfrac{\mathrm{(CC)\,A}}{\mathrm{(CC)\,R}} \bullet \left[\dfrac{\mathrm{ESR}}{\mathrm{EDR}} + 1\right]}{\left[\dfrac{\mathrm{A}}{\mathrm{R}}_{\mathrm{RT}} \bullet \dfrac{\mathrm{A}}{\mathrm{R}}_{\mathrm{CYCLE}} \bullet \dfrac{\mathrm{A}}{\mathrm{R}}_{\mathrm{COLLECTOR}} \bullet \dfrac{\mathrm{ESR}}{\mathrm{EDR}}\right] + \left[\dfrac{\mathrm{AD}}{\mathrm{RD}}_{\mathrm{CYCLE}} \bullet \dfrac{\mathrm{AD}}{\mathrm{RD}}_{\mathrm{COLLECTOR}}\right]} \tag{A-13}$$

The above function will evaluate RI for one mission. Repeating the calculation for various missions will determine the ranking over as large a range as the user desires.

The function does not require an insolation simulation model of the solar thermal system for every alternative concept to perform the calculation. Insolation is considered in the derivation, but the approach forces the insolation factors to divide out. This property greatly reduced the work required to evaluate the rankings.

SERI

APPENDIX B

REFERENCE SYSTEMS

Prepared by M. E. Karpuk

B.1 SUMMARY

Cost estimates were made for the solar thermal power systems used with the thermal storage ranking methodology being developed at SERI. The source of the cost information primarily is advocate publications. Algorithms were developed to estimate system cost with changes in collector area. The data are shown in Section B.4, Tables B-1 through B-14.

B.2 INTRODUCTION

A methodology to rank thermal energy storage technologies for solar power applications is being developed at SERI. Part of the data required to use this methodology is the cost of the solar system that contains the storage element. The purpose of this study is to provide the cost data for the solar systems used with the methodology.

The solar power systems used with the methodology include large and small central receiver and distributed focus concepts. These systems were defined by Karpuk (1979). Many of the systems are being developed by DOE contractors. For these systems, cost data were obtained from design reports.

It is envisioned that for various storage concepts and capacities, the collector field area will be allowed to vary. Algorithms have therefore been developed to estimate system cost changes with collector area. Changes of collector area will not affect the electric rating of the plant.

Estimating the cost of the storage systems is beyond the scope of this study.

B.3 DISCUSSION

The major results of this study are contained in Tables B-1 through B-14. The system costs are broken down to the extent possible into the standard subsystem accounts. The costs shown are for a commercial plant, i.e., mature technology, and in 1978 dollars. Each of the systems is described below with an explanation of the cost derivation.

Element 1A is a 100-MW_e central power station employing a steam water central receiver and oil-rock storage. A schematic of the system is shown in Fig. B-1. Because the steam generated from storage is at a significantly lower temperature and pressure than the steam generated in the receiver, the turbine has a separate admission port for steam generated from storage.

The cost data shown in Table B-1 are from an EPRI study done by Westinghouse (1978). The cost data were given in 1976 dollars so an inflation rate of 9% was used to bring the cost to 1978 dollars. The cost of the heliostats was assumed to be $\$80/m^2$.

The algorithm to estimate the increase in plant cost with collector area is shown in Table B-14. The algorithm assumes:

1. Heliostat costs increase linearly with area at $\$80/m^2$.
2. Tower costs increase as the square of tower height.
3. Tower height increases as the square root of collector area.
4. Receiver and piping increase as the 0.8 power with collector area (thermal power).

The first assumption is based on DOE goals and mass production of heliostats. The second and third assumptions are based on Figs. B-2 and B-3. Figure B-2 shows tower cost for many proposed solar plants as a function of height. Most of the towers shown are concrete. The wide variation in cost is due to different wind loading, seismic requirements, and receiver weight. A square curve was drawn through the data and represents the data fairly well. Figure B-3 shows tower height variation with heliostat area for capital receiver solar plants. For a particular plant design, i.e., Martin Marietta's molten salt concept, the data correlate well with a square-root curve. Physically this would occur to keep the ratio of the heliostat field radius to tower height constant. Assumptions 2 and 3 mean that the tower costs increase directly with collector area.

The algorithm shown in Table B-14 for Element 1A assumes the cost of the tower is 40% of the 4500 account cost. This number was derived from a cost study by McDonnell Douglas (1977). This number is not critical, however, since the tower represents only a small portion of the total of accounts 4400 and 4500.

Element 1B is an advanced central receiver with a 100-MW_e rating. Two concepts were considered for this element; a molten salt system and a liquid metal system. A molten salt system proposed by Martin Marietta is shown in Fig. B-4 and a liquid metal system proposed by General Electric is shown in Fig. B-5. Table B-2 shows a cost estimate for the molten salt system from Martin Marietta (1978). Table B-3 shows the cost estimate for a liquid metal receiver concept from Rockwell International (1978).

The cost algorithms shown in Table B-14 are based on the assumptions used for Element 1A and the tower costs from Martin Marietta (1978) and Rockwell International (1978).

Element 2 is an open air Brayton cycle with a 100-MW_e output. A central receiver is used along with an oil combustor to bring the turbine inlet temperature to 2000°F. The plant configuration is shown in Fig. B-6. Black and Veatch Consulting Engineers have done a conceptual design of a 60-MW_e plant (Grosskreutz et al. 1977). The cost estimates for that plant were scaled up and converted to 1978 dollars. The cost information is shown in Table B-4.

The cost algorithms shown in Table B-14 are based on the assumptions used for Element 1A and a tower cost of 40% of the 4500 account.

Element 3A is a 10-MW_e plant with a steam water central receiver. The plant configuration is similar to the pilot plant to be built at Barstow, Calif. and is shown in Fig. B-7. Table B-5 shows a cost estimate from Aerospace Corp. (1978). The cost algorithm shown in Table B-14 is based on the assumptions used for Element 1A and a tower cost of $630,000 from the same source.

Element 3B is a 150-kW_e organic Rankine power cycle with distributed tracking concentrating collectors. A schematic of the plant is shown in Fig. B-8. This system could be used as a power source for irrigation or to generate electricity in remote areas. The cost data shown in Table B-6 are from Barber (1978) and a discussion with J. Finegold of SERI. Since this system uses distributed tracking collectors, accounts 4400 and 4500 were grouped together. The cost algorithm in Table B-14 is based on $90/$m^2$ cost of tracking trough collectors.

Element 3C is a total energy system. The system provides 400 kW of electric power from a steam Rankine cycle as well as 1580 lb/h process steam at 340°F and 105 psig. The system is identical to the Shenandoah total energy system that will be built at a textile mill in Georgia. A system schematic is shown in Fig. B-9. Most of the cost estimates shown in Table B-7 are from data General Electric (1978) has generated for the Shenandoah plant. The cost of the parabolic dishes were estimated from DOE cost goals. The cost algorithm is shown in Table B-14.

Element 4A is an existing fossil-fired plant repowered with a solar steam supply. A heliostat field with a central steam water receiver is installed next to an existing 100-MW_e nonreheat power plant. Figure B-10 is a schematic of the plant with parallel buffering storage. Table B-8 shows the cost of the solar steam supply from Christmas et al. (1979). Since the design of the heliostat field and receiver is identical to Element 1A, the cost algorithm shown in Table B-14 is also identical.

Element 4B is a 15-MW_e dish Stirling plant. It uses 1456 dishes with a 10-m diameter. At the focal point of each dish is a Stirling engine with a 10.3-kW_e output. A typical plant of this type is shown in Fig. B-11. The cost data from A. Herlevich (discussion) is shown in Table B-9. The cost algorithm is shown in Table B-14. Since all of the cost accounts are affected by increases in collector area, the algorithm represents total direct plant cost.

Element 4C is a closed Brayton cycle with a capacity of 100 MW_e. The schematic of a plant designed by Boeing is shown in Fig. B-12. The cost for this plant, which has a capacity of 150 MW_e, was scaled down to 100 MW_e and is shown in Table B-10. The cost data from W. D. Beverly of Boeing Engineering and Construction Company (discussion, Feb. 1979) are preliminary. The cost algorithm in Table B-14 is based on the Element 1A assumptions and the tower costs from Beverly.

Element 5 (Table B-11) is identical to the liquid metal receiver plant, Element 1B, except for the addition of a thermochemical storage element. The thermochemical storage element requires low-pressure steam during discharge. This complicates turbine design, and 50% was added to the turbine account 4200. A schematic of the system is shown in Fig. B-13. The cost algorithm shown in Table B-14 is identical to Element 1B since the thermochemical storage system does not affect the solar part of the plant.

Element 6A and 6B (Fig. B-14) are small power systems with the collector field connected to the steam turbine via a thermochemical heat transport pipeline. Element 6A has a 10-MW_e capacity and Element 6B has a 300-kW_e capacity. The cost estimates in Tables B-12 and B-13 are only for the collector field and power conversion equipment. The cost algorithm for changes in collector field size is shown in Table B-14.

B.4 ELEMENT TABLES AND FIGURES

Table B-1. Element 1A, 100-MW_e Steam Water Central Receiver[a]

	1978\$, $X10^{-3}$	1978\$/kW_e
4100 Land Structures & Facilities	6,300	63
4200 Turbine Plant	17,800	178
4300 Electric Plant Equipment	2,700	27
4400 Collector Equipment Area — 700,000 m^2 \$80/m^2	56,000	560
4500 Receiver Equipment	23,700	237
Total Direct	106,500	1,065
Contingency & Spares — 15%	15,975	160
Indirect Costs — 10%	10,650	107
Total Capital	133,125	1,331
Interest During Construction — 15%	19,969	200
Total Capitalization	153,094	1,531

[a]Westinghouse (1978).

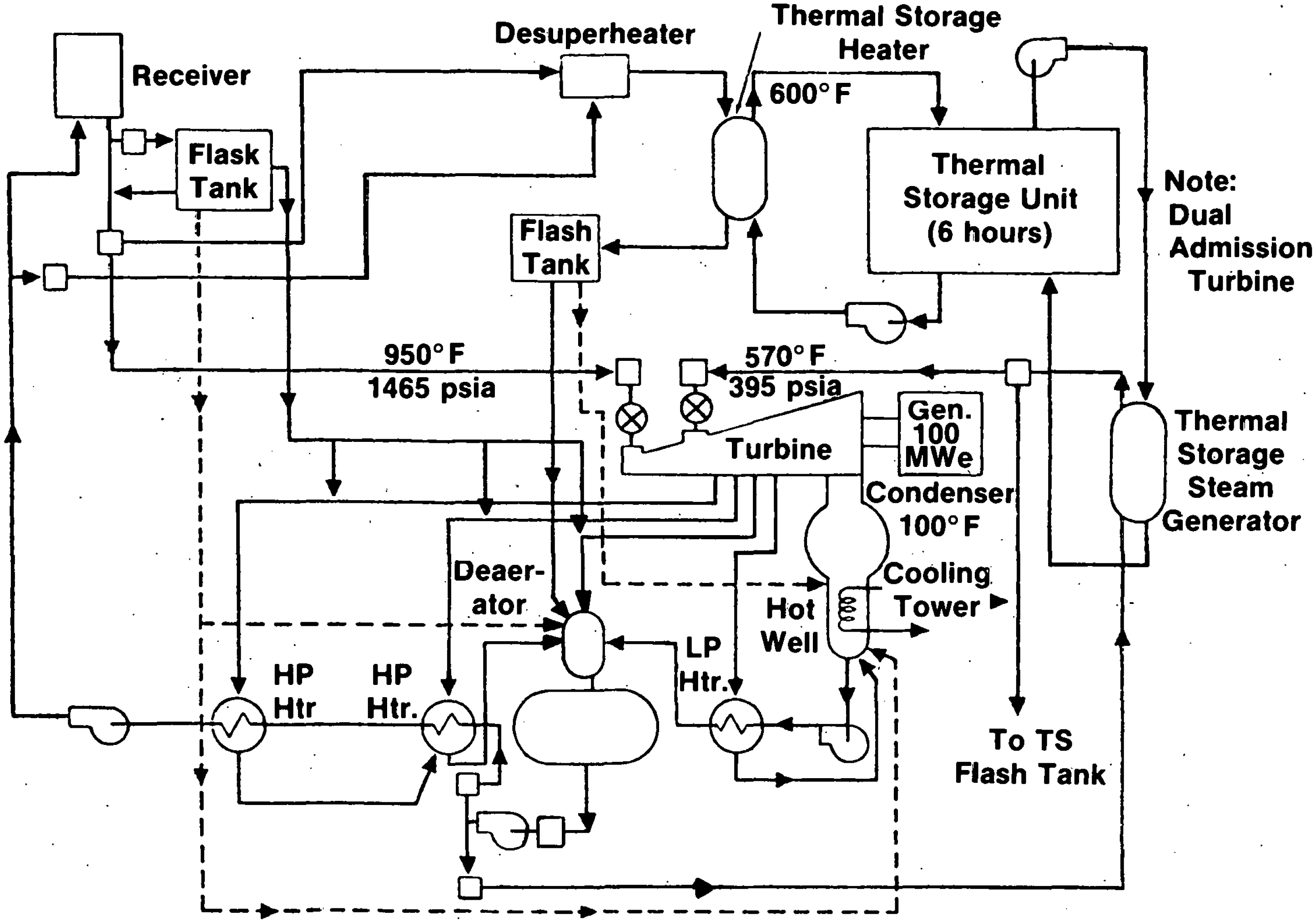

Receiver
Flask Tank
Desuperheater
Thermal Storage Heater
600° F
Thermal Storage Unit (6 hours)
Flash Tank
Note: Dual Admission Turbine
950° F
1465 psia
570° F
395 psia
Gen. 100 MWe
Turbine
Condenser
100° F
Thermal Storage Steam Generator
Cooling Tower
Deaer-ator
Hot Well
LP Htr.
HP Htr
HP Htr.
To TS Flash Tank

Figure B-1.

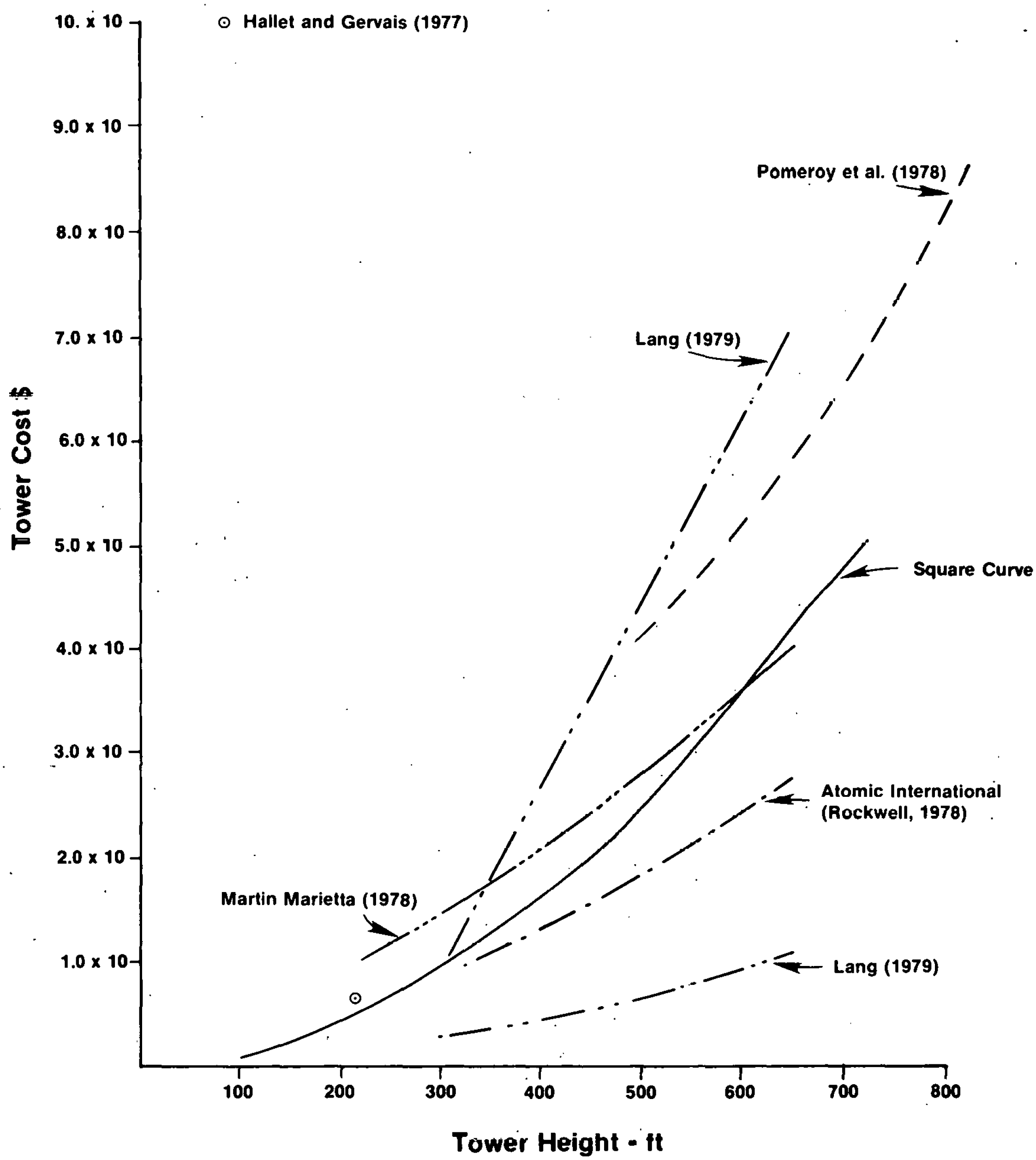

Hallet and Gervais (1977)
10. x 10
9.0 x 10
8.0 x 10
7.0 x 10
6.0 x 10
5.0 x 10
4.0 x 10
3.0 x 10
2.0 x 10
1.0 x 10
Tower Cost $
Pomeroy et al. (1978)
Lang (1979)
Square Curve
Atomic International
(Rockwell, 1978)
Martin Marietta (1978)
Lang (1979)
100
200
300
400
500
600
700
800
Tower Height - ft

Figure B-2.

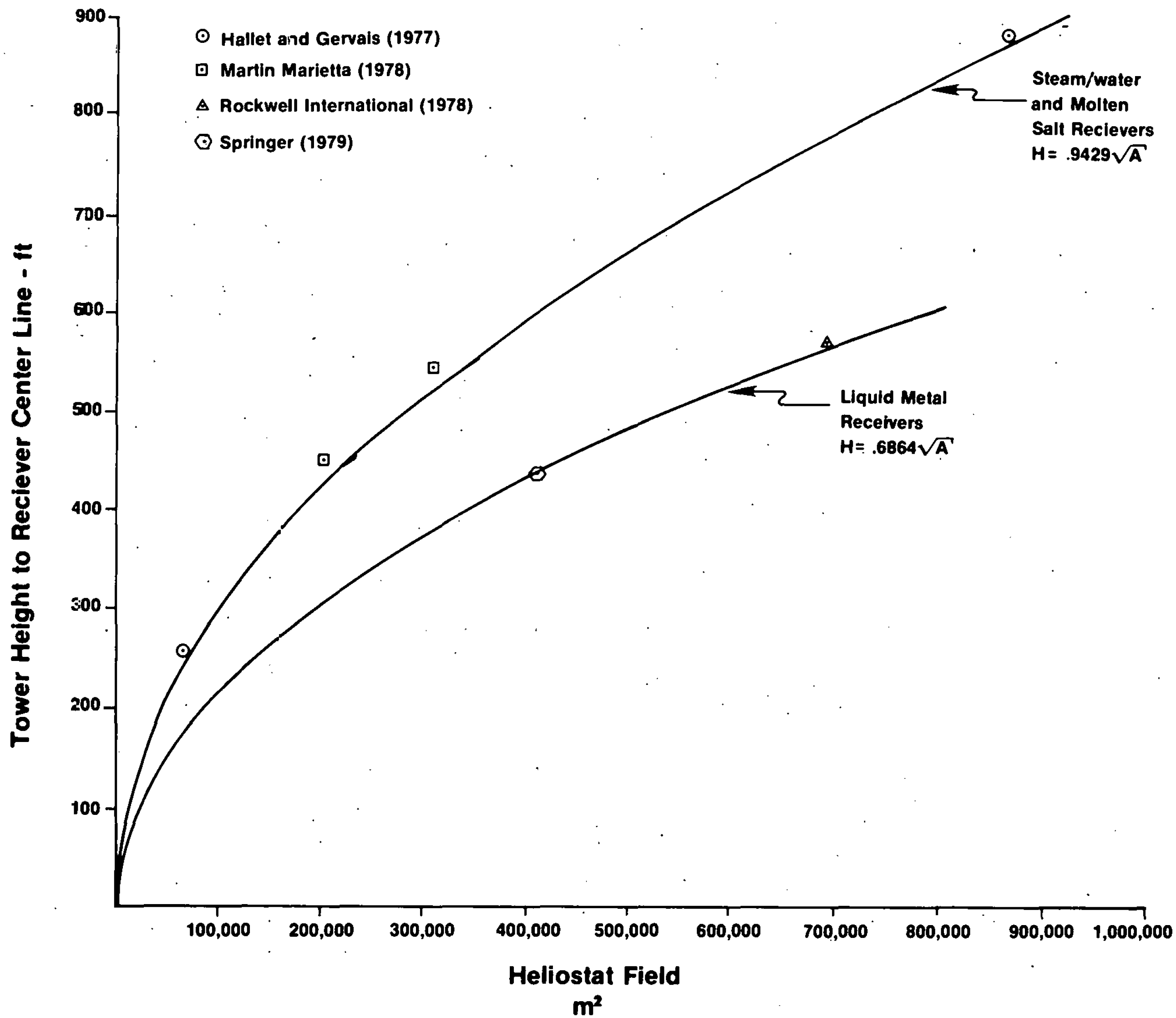

Hallet and Gervais (1977)
Martin Marietta (1978)
Rockwell International (1978)
Springer (1979)
Tower Height to Reciever Center Line - ft
900
800
700
600
500
400
300
200
100
100,000
200,000
300,000
400,000
500,000
600,000
700,000
800,000
900,000
1,000,000
Heliostat Field
m^2
Steam/water and Molten Salt Recievers
$H = .9429\sqrt{A}$
Liquid Metal Receivers
$H = .6864\sqrt{A}$

Figure B-3.

Table B-2. Element 1B, 100-MW_e Molten Salt Central Receiver[a]

	1978\$, $X10^{-3}$	1978\$/kW_e
4100 Land Structures & Facilities	1,000	10
4200 Turbine Plant	30,200[b]	302
4300 Electric Plant Equipment		
4400 Collector Equipment Area — 607,080 m^2 \$80/m^2	48,570	485
4500 Receiver Equipment	10,700	107
Total Direct	90,470	905
Contingency & Spares — 15%	13,570	136
Indirect Costs — 10%	9,047	90
Total Capital	113,087	1,131
Interest During Construction — 15%	16,963	170
Total Capitalization	130,050	1,301

[a]Martin Marietta (1978).

[b]Master Control, EPGS, Salt Steam HX.

Table B-3. Element 1B, 100-MW_e Liquid Metal Central Receiver[a]

	1978\$, $X10^{-3}$	1978\$/kW_e
4100 Land Structures & Facilities	4,841	43
4200 Turbine Plant	19,424	194
4300 Electric Plant Equipment	3,920	40
4400 Collector Equipment	56,400	564
Area — 705,000 m^2 \$80/m^2		
4500 Receiver Equipment	19,040	190
Total Direct	103,625	1,036
Contingency & Spares — 15%	15,544	155
Indirect Costs — 10%	10,363	104
Total Capital	129,531	1,295
Interest During Construction — 15%	19,430	194
Total Capitalization	148,960	1,490

[a]Rockwell International (1978).

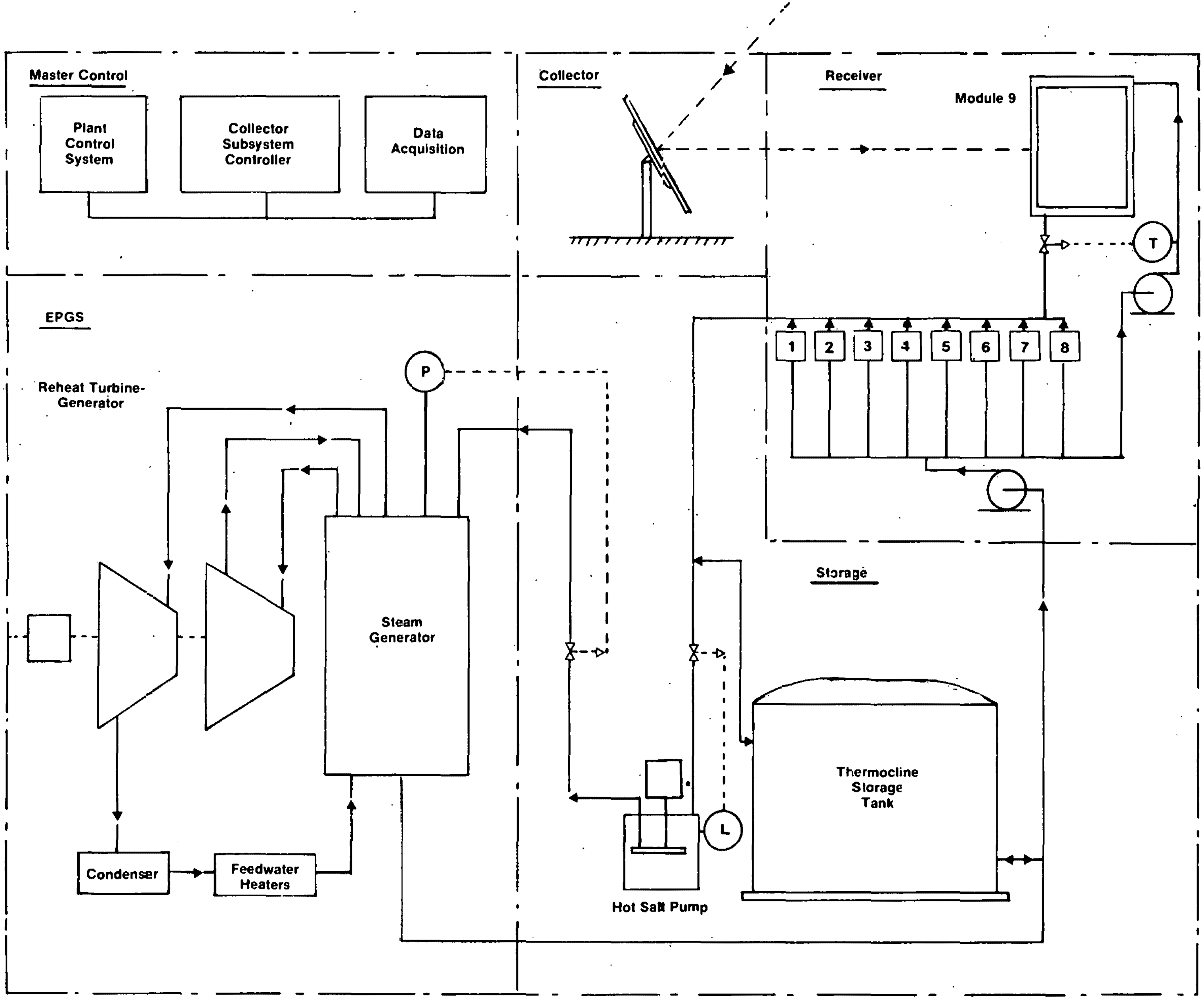

Master Control
Plant Control System
Collector Subsystem Controller
Data Acquisition
Collector
Receiver
Module 9
T
EPGS
Reheat Turbine-Generator
P
1
2
3
4
5
6
7
8
Steam Generator
Storage
Thermocline Storage Tank
L
Condenser
Feedwater Heaters
Hot Salt Pump

Figure B-4.

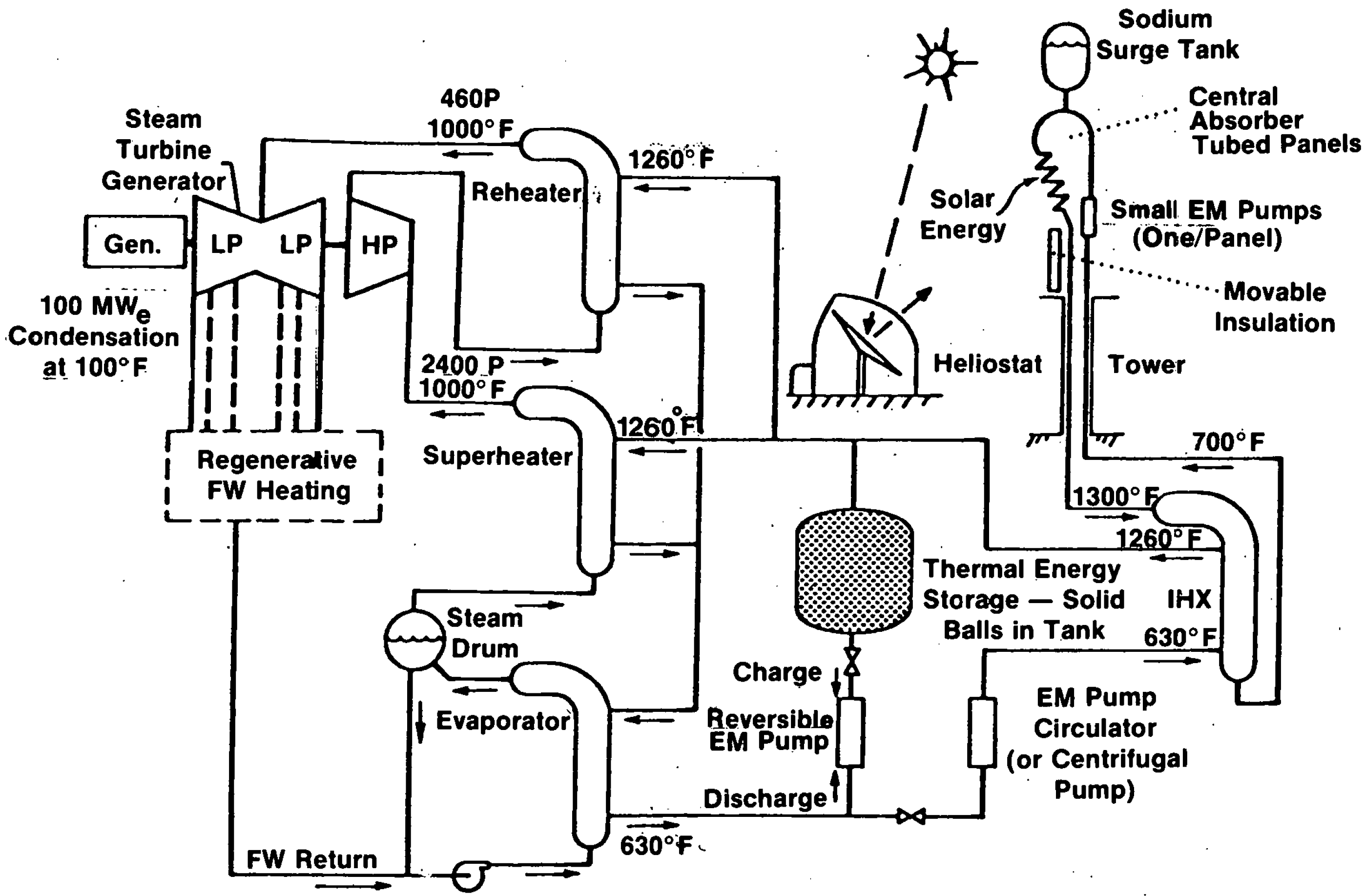
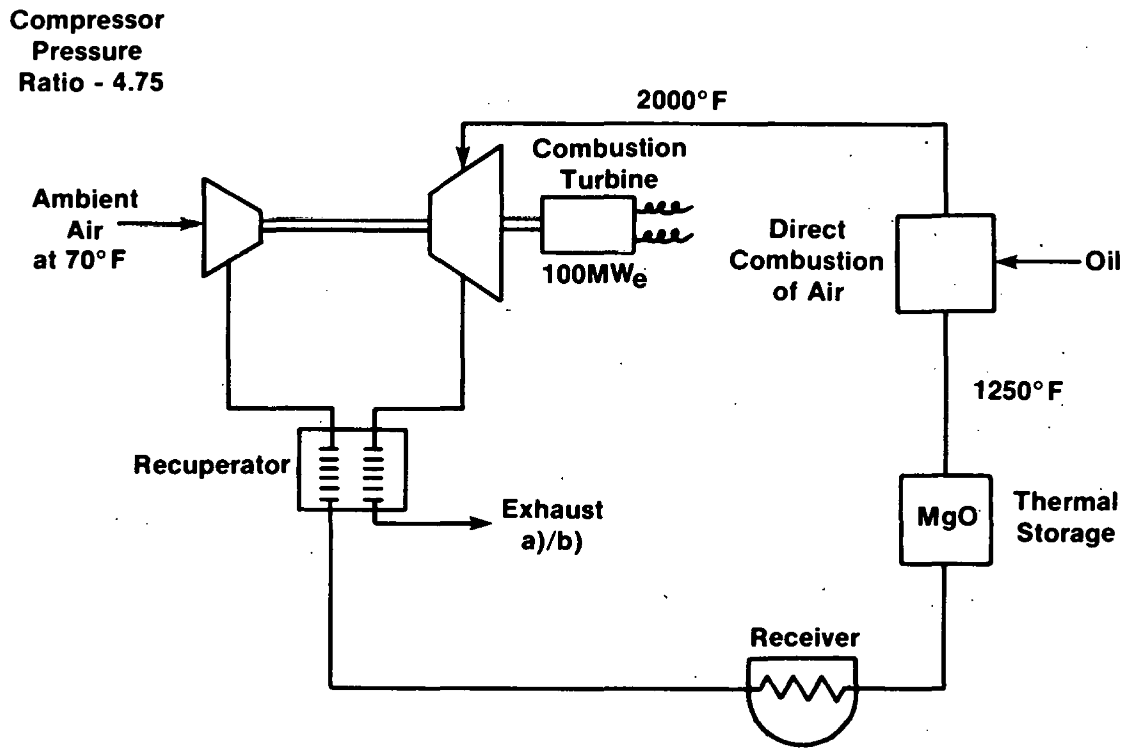

Steam
Turbine
Generator
Gen.
LP
LP
HP
100 MWe
Condensation
at 100°F
460P
1000°F
Reheater
1260°F
2400 P
1000°F
Superheater
1260°F
Regenerative
FW Heating
Steam
Drum
Evaporator
FW Return
630°F
Solar
Energy
Heliostat
Sodium
Surge Tank
Central
Absorber
Tubed Panels
Small EM Pumps
(One/Panel)
Movable
Insulation
Tower
700°F
1300°F
1260°F
Thermal Energy
Storage — Solid
Balls in Tank
IHX
630°F
Charge
Reversible
EM Pump
Discharge
EM Pump
Circulator
(or Centrifugal
Pump)

Figure B-5.

Table B-4. Element 2, 100-MW$_e$ Open Air Brayton Cycle[a]

	1978\$, X10^{-3}	1978\$/kW$_e$
4100 Land Structures & Facilities	5,100	51
4200 Turbine Plant	24,700	247
4300 Electric Plant Equipment	3,100	31
4400 Collector Equipment Area — 733,333 m^2 \$80/m^2	58,667	587
4500 Receiver Equipment	41,200	412
Total Direct	132,800	1,328
Contingency & Spares — 15%	19,920	199
Indirect Costs — 10%	13,280	133
Total Capital	166,000	1,660
Interest During Construction — 15%	24,900	249
Total Capitalization	190,900	1,909

[a]Grosskreutz et al. (1977).



a) Exhaust air is directly released

b) Exhaust supplies heat to a steam Rankine bottoming cycle (i.e., a combined cycle system)

Figure B-6.

Table B-5. Element 3A, 10-MW_e Steam Water Central Receiver[a]

	1978\$, $X10^{-3}$	1978\$/kW_e
4100 Land Structures & Facilities	2,550	255
4200 Turbine Plant	3,820	382
4300 Electric Plant Equipment	3,380	338
4400 Collector Equipment Area — 66,800 m^2 \$80/m^2	5,344	534
4500 Receiver Equipment	5,640	564
Total Direct	20,734	2,073
Contingency & Spares — 15%	3,110	311
Indirect Costs — 10%	2,073	207
Total Capital	25,917	2,591
Interest During Construction — 15%	3,887	389
Total Capitalization	29,804	2,980

[a]Aerospace Corp. (1978).

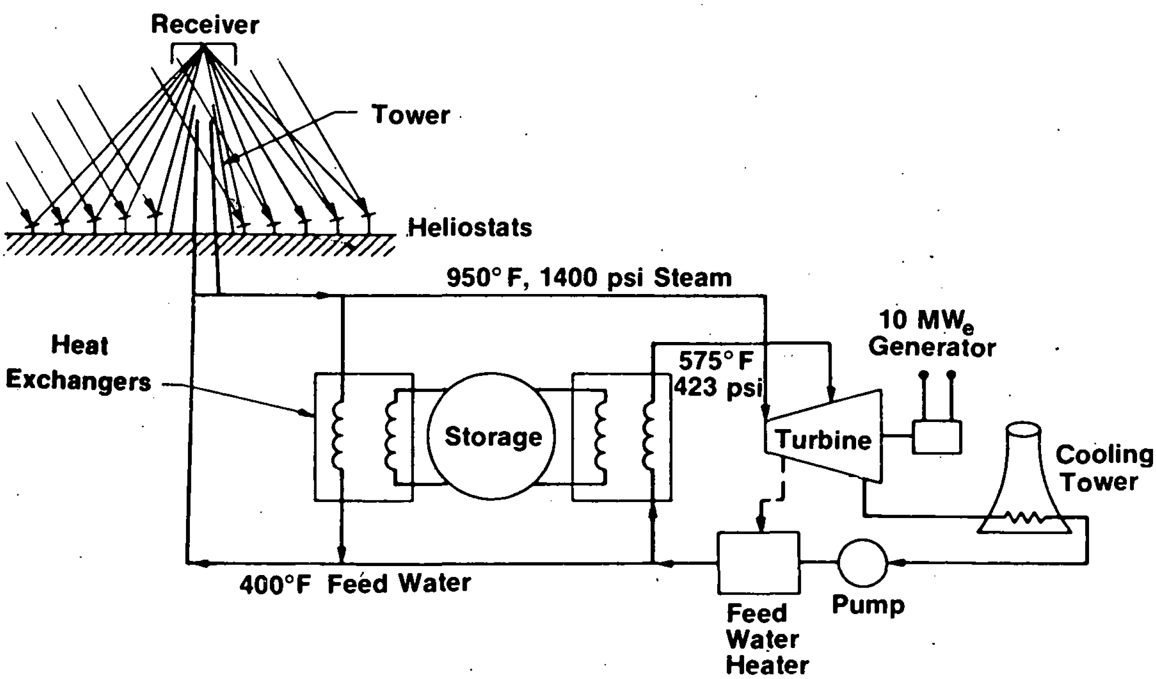


Figure B-7. Central Receiver Solar Thermal-Electric Power Plant

Table B-6. Element 3B, 150-kW_e Organic Rankine Cycle with Distributed Collectors[a,b]

	1978\$, X10^{-3}	1978\$/kW_e
4100 Land Structures & Facilities	10	67
4200 Turbine Plant	98[a]	653
4300 Electric Plant Equipment		
4400 Collector Equipment Area — $18 m^2/kW$ 2,700 m^2 \$90/m^2	243[b,c]	560
4500 Receiver Equipment	10[d]	68
Total Direct	361	2,412
Contingency & Spares — 15%	54	362
Indirect Costs — 10%	36	241
Total Capital	451	3,015
Interest During Construction — 15%	45	302
Total Capitalization	496	3,317

[a]Barber (1978).

[b]Discussion with J. Finegold, SERI Small Power System Study Group.

[c]Collector includes receiver.

[d]Field piping costs, J. Finegold.

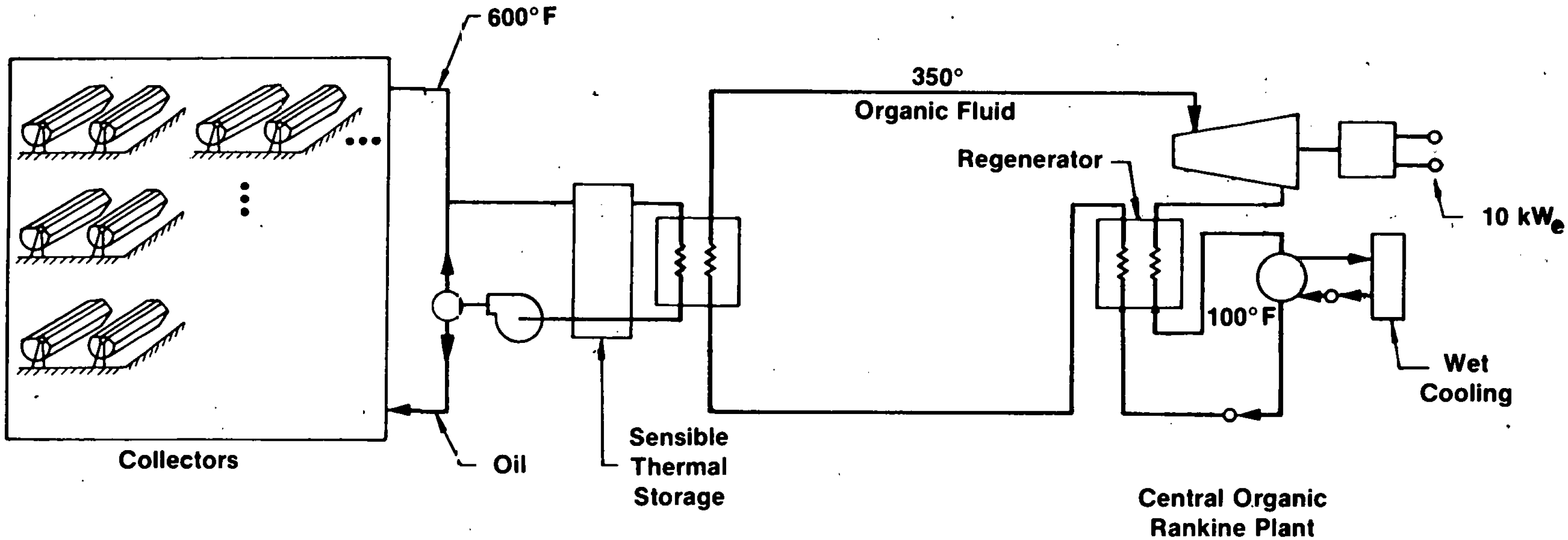

600° F
350°
Organic Fluid
Regenerator
10 kWe
100° F
Wet Cooling
Collectors
Oil
Sensible Thermal Storage
Central Organic Rankine Plant

Figure B-8.

Table B-7. Element 3C, Solar Total Energy System with 400-kW_e and 1580-lb/h Steam Capacity[a]

	1978\$, $X10^{-3}$	1978\$/kW_e
4100 Land Structures & Facilities	50	125
4200 Turbine Plant Generator	500	1,250
4300 Electric Plant Equipment, Controls	440	1,100
4400 Collector Equipment[b] Area — 7,386,000 m^2 \$100/m^2	738[d]	1,846
4500 Receiver Equipment	510[e]	1,290
Other Direct Costs	1,850	4,625
Total Direct	4,088	10,236
Contingency & Spares — 15%	541	1,535
Indirect Costs — 10%	409	1,024
Total Capital	5,110	1,331
Interest During Construction — 15%	511	1,279
Total Capitalization	5,621	14,075

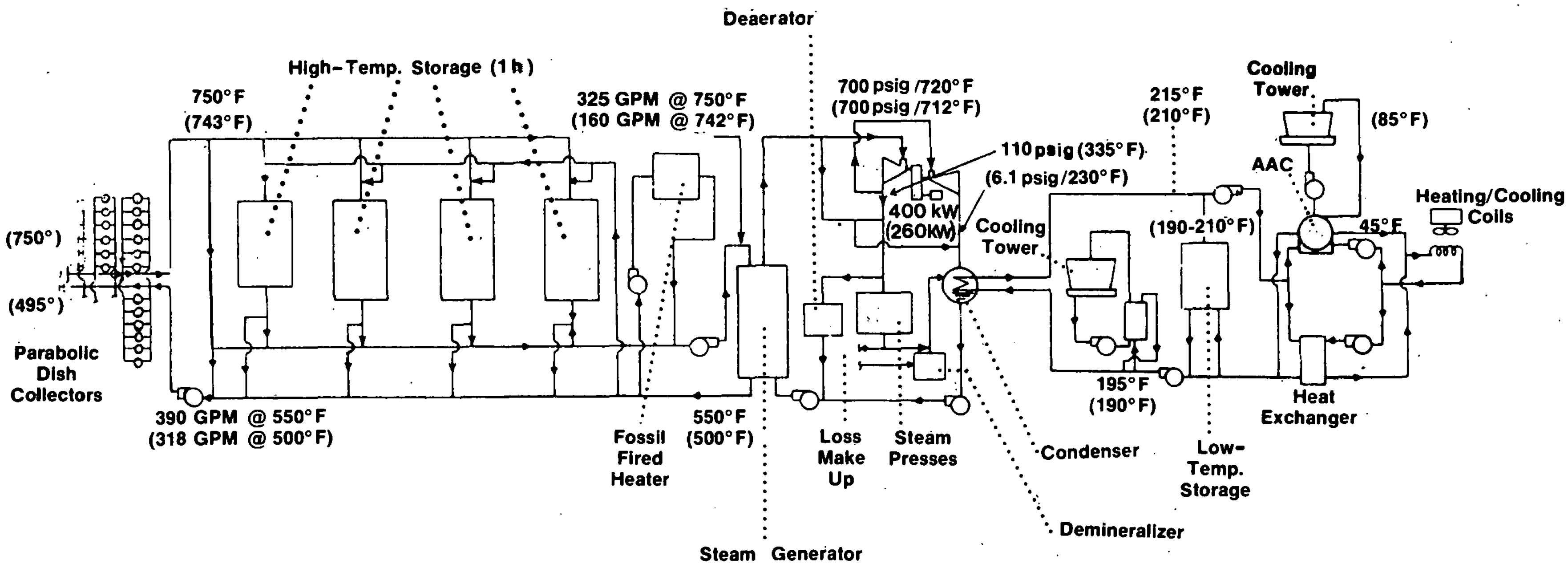
[a]General Electric (1978).

[b]This cost based on DOE goal.

[c]Energy plant costs from General Electric (1978).

[d]Collector includes receiver.

[e]Collector Field piping (Iannucci and Eicker, 1980).



Note:

Design Maximum : I_{dn} — 300 Btu/h-ft²

Q_{elec} — 400 kW

Nominal : I_{dn} — 270 Btu/h-ft²

Q_{elec} — 260 kW

Figure B-9. Shenandoah LSE System Schematic

Table B-8. Element 4A, 100-MW_e Solar Repower, Central Steam-Water Receiver[a]

	1978$, X$10^{-3}$	1978$/$kW_e$
4100 Land Structures & Facilities	800	8
4200 Turbine Plant	15,500	155
4300 Electric Plant Equipment		
4400 Collector Equipment	40,000	400
Area — 500,000 m^2 $80/$m^2$		
4500 Receiver Equipment	17,600[b]	176
Total Direct	73,900	739
Contingency & Spares — 15%	11,085	111
Indirect Costs — 10%	7,390	74
Total Capital	92,375	924
Interest During Construction — 15%	13,856	138
Total Capitalization	106,231	1,062

[a]Christmas et al. (1979).

[b]Estimated from Table 14.

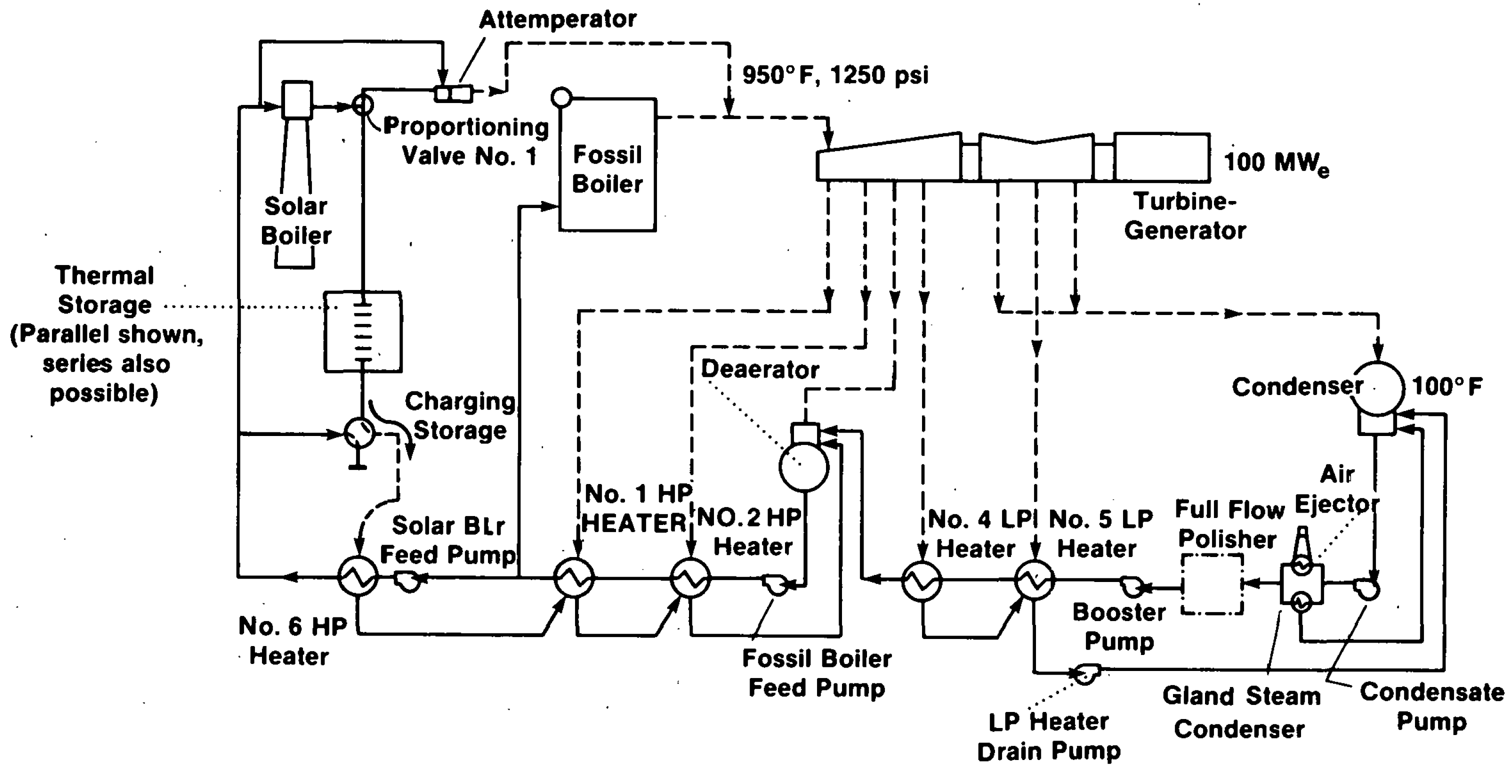

Attemperator
950° F, 1250 psi
Proportioning Valve No. 1
Fossil Boiler
100 MWe
Turbine-Generator
Solar Boiler
Thermal Storage (Parallel shown, series also possible)
Charging Storage
Deaerator
Condenser
100° F
Air Ejector
Full Flow Polisher
No. 1 HP HEATER
NO. 2 HP Heater
No. 4 LP Heater
No. 5 LP Heater
Solar BLr Feed Pump
No. 6 HP Heater
Fossil Boiler Feed Pump
Booster Pump
LP Heater Drain Pump
Gland Steam Condenser
Condensate Pump

Figure B-10.

Table B-9. Element 4B, 15-MW_e Dish Stirling Plant[a]

	1978\$, X10^{-3}	1978\$/kW_e
4100 Land Structures & Facilities	15	1
4200 Turbine Plant	1,875	125
4300 Electric Plant Equipment	6,110	407
4400 Collector Equipment Area — 1,456; 10-m diameter 114,353 m^2 \$80/m^2	11,435	762
4500 Receiver Equipment	1,485	99
Total Direct	20,920	1,394
Contingency & Spares — 15%	3,138	209
Indirect Costs — 10%	1,046	70
Total Capital	25,104	1,673
Interest During Construction — 15%	2,510	167
Total Capitalization	27,614	1,840

[a]Discussion with A. Herlevich, SERI Small Power Systems Group.

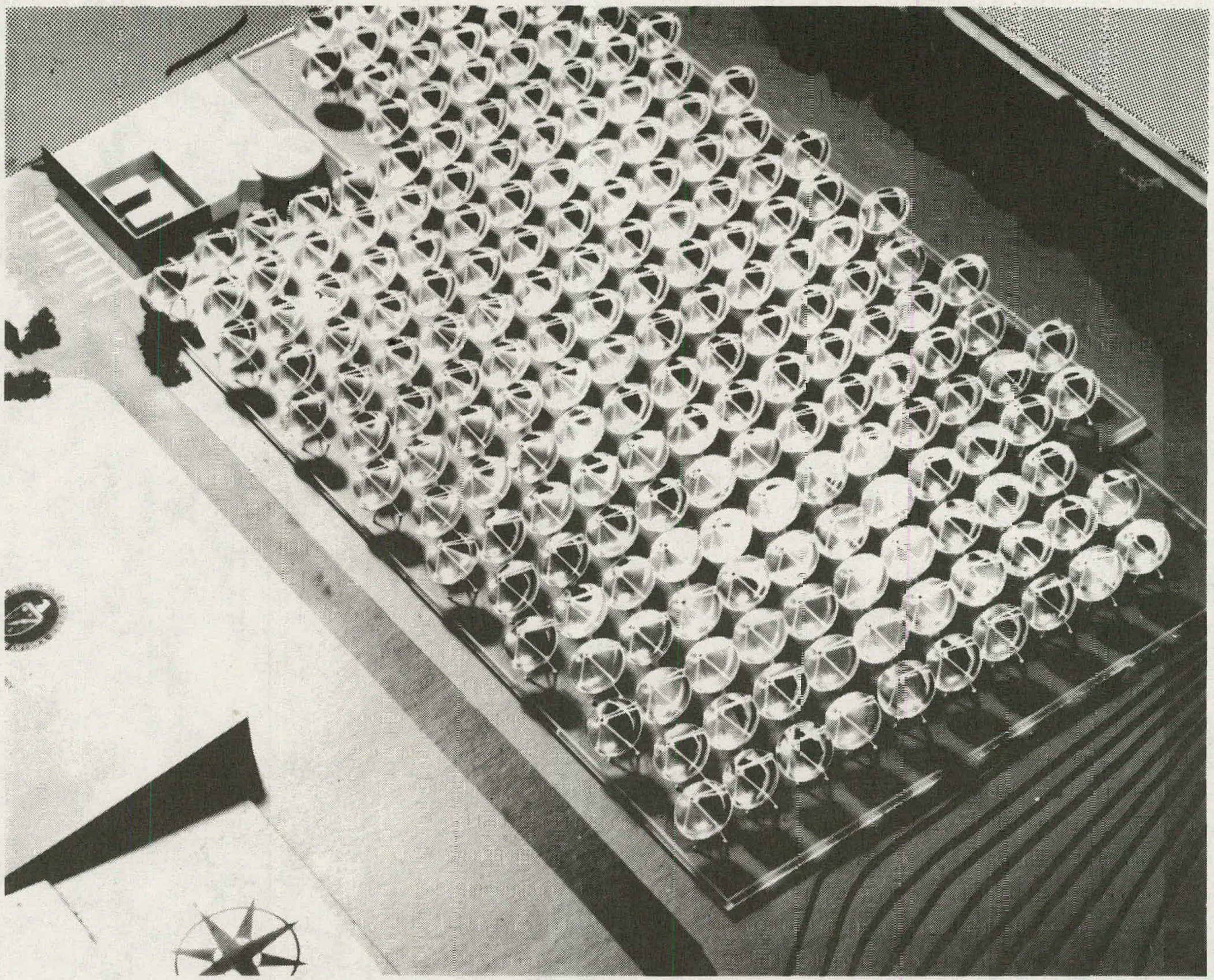
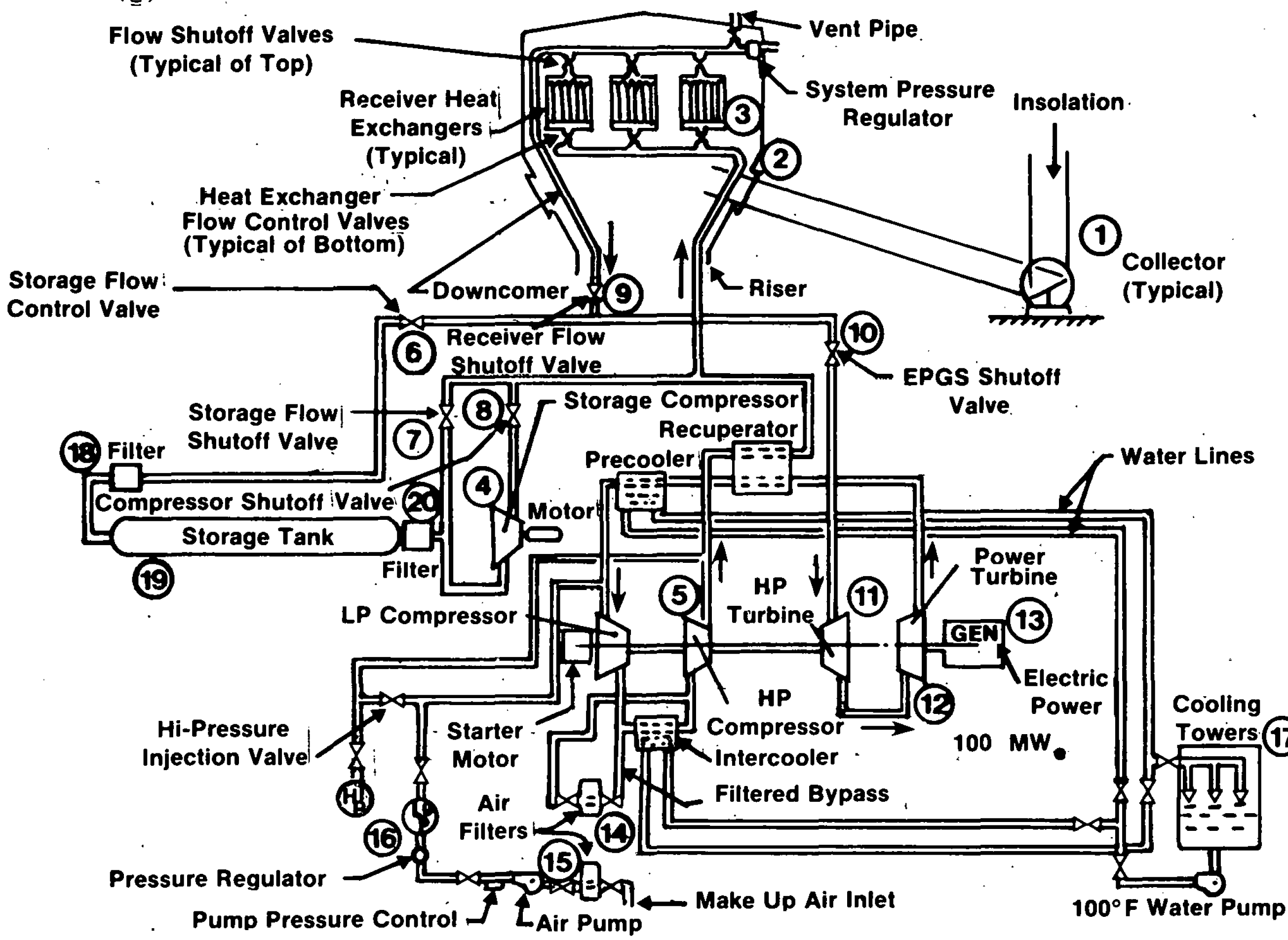


Figure B-11

Table B-10. Element 4C, 100-MW_e Closed Brayton Cycle[a]

	1978\$, $X10^{-3}$	1978\$/kW_e
4100 Land Structures & Facilities	13,400	134
4200 Turbine Plant	25,900	259
4300 Electric Plant Equipment	15,400	154
4400 Collector Equipment	73,440	734
Area — 918,000 m^2 \$80/m^2		
4500 Receiver Equipment	20,800	208
Total Direct	148,940	1,489
Contingency & Spares — 15%	22,340	223
Indirect Costs — 10%	14,894	149
Total Capital	186,175	1,861
Interest During Construction — 15%	27,926	279
Total Capitalization	214,101	2,140

[a]Discussion with W. D. Beverly of Boeing Engineering and Construction Co., 7 Feb. 1979.



Preferred EPGS Design Point Requirements (Closed Cycle Air)

Turbine Inlet Temperature	815.6°C (1500°F)
Turbine Inlet Pressure	34.5 bar (500 psia)
Compressor Pressure Ratio	4.75
Compressor Inlet Temperature	37.8°C (100°F)
Recuperator Effectiveness	0.902
Component Pressure Losses (P/P)	
Precooler	1.0%
Intercooler	1.0%
Recuperator	4.0%
Receiver	3.5%
EPGS Ducting (P/P)	2.2%
Cooling Water Inlet Temperature	29.4° C (85° F)
Cooling Water Temperature Rise	
Precooler	12° C (26° F)
Intercooler	17° C (30° F)

Figure B-12.

Table B-11. Element 5, Liquid Metal Central Receiver with Thermochemical Storage

	1978\$, $X10^{-3}$	1978\$/kW_e
4100 Land Structures & Facilities	4,841	48
4200 Turbine Plant	29,136	291
4300 Electric Plant Equipment	3,920	40
4400 Collector Equipment	56,400	564
Area — 705,000 m^2 \$80/m^2		
4500 Receiver Equipment	19,040	190
Total Direct	113,364	1,133
Contingency & Spares — 15%	17,004	170
Indirect Costs — 10%	11,336	113
Total Capital	141,705	1,417
Interest During Construction — 15%	21,256	213
Total Capitalization	162,961	1,630

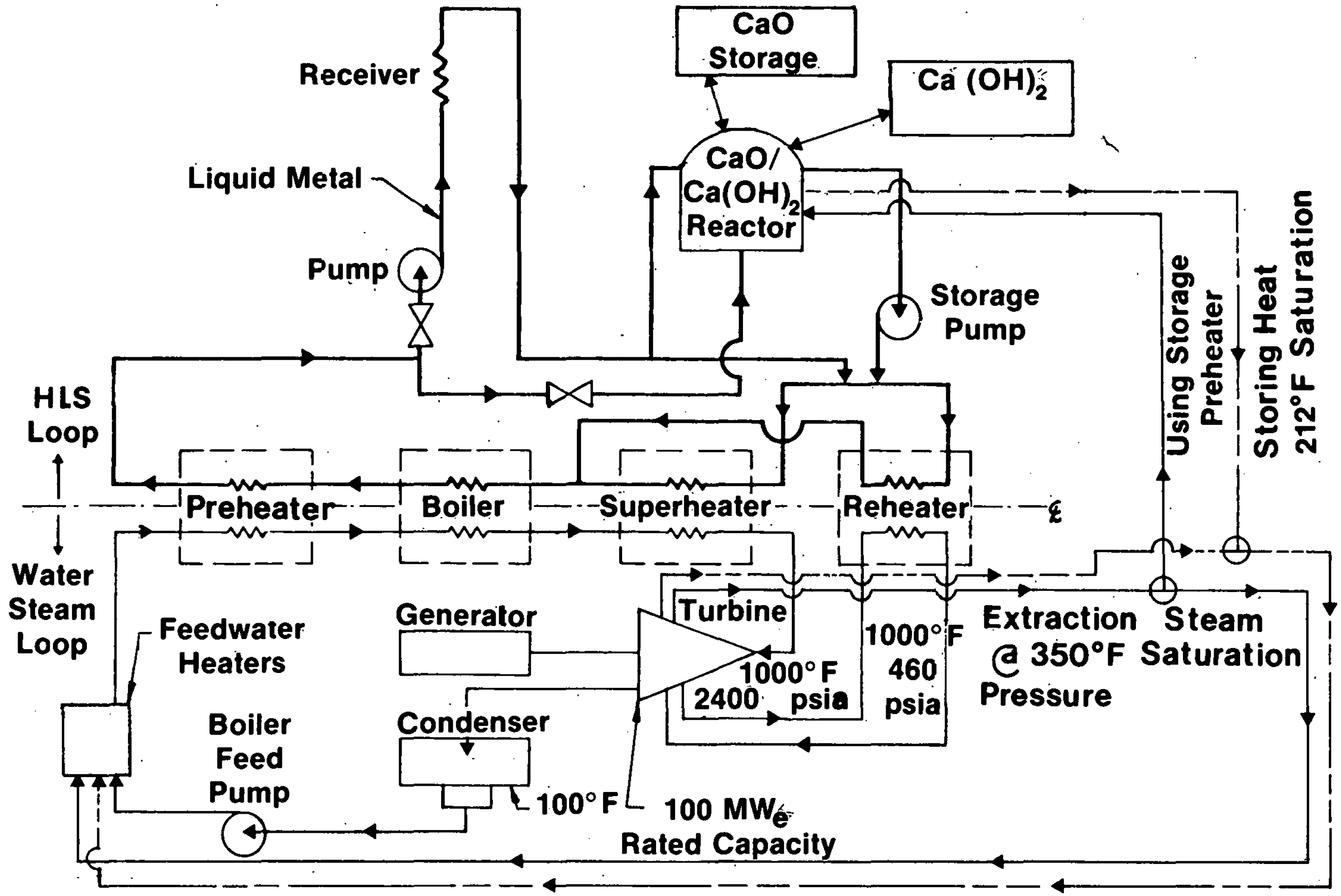

Receiver
Liquid Metal
Pump
HLS Loop
Water Steam Loop
CaO Storage
Ca (OH)$_2$
CaO/ Ca(OH)$_2$ Reactor
Storage Pump
Using Storage Preheater
Storing Heat 212°F Saturation
Preheater
Boiler
Superheater
Reheater
℄
Feedwater Heaters
Generator
Turbine
1000°F 2400 psia
1000°F 460 psia
Extraction @ 350°F Pressure
Steam Saturation
Condenser
Boiler Feed Pump
100°F
100 MW$_e$ Rated Capacity

Figure B-13.

Table B-12. Element 6A, 10-MW_e Nonreheat Steam Power Cycle with Thermochemical Transport[a]

	1978\$, X10^{-3}	1978\$/kW_e
4100 Land Structures & Facilities	2,550	255
4200 Turbine Plant	3,820	380
4300 Electric Plant Equipment	3,380	338
4400 Collector Equipment	15,000	1,500
Area — 700,000 m^2 \$80/m^2		
4500 Receiver Equipment		
Total Direct	24,750	2,475
Contingency & Spares — 15%	3,712	371
Indirect Costs — 10%	2,475	248
Total Capital	30,937	3,094
Interest During Construction — 15%	3,094	309
Total Capitalization	34,031	3,403

[a]Estimate does not include cost of thermochemical transport or steam generation equipment.

Table B-13. Element 6B, 300-kW_e Nonreheat Steam Power Cycle with Thermochemical Transport[a]

	1978\$, $X10^{-3}$	1978\$/kW_e
4100 Land Structures & Facilities	77	255
4200 Turbine Plant	482[b]	1,608
4300 Electric Plant Equipment		
4400 Collector Equipment Area — 15 m^2/kW_e 4,500 m^2 \$100/m^2	450	1,500
4500 Receiver Equipment		
Total Direct	1,009	3,363
Contingency & Spares — 15%	151	504
Indirect Costs — 10%	101	336
Total Capital	1,261	4,203
Interest During Construction — 15%	126	420
Total Capitalization	1,387	4,623

[a]Estimate does not include cost of thermochemical transport or steam generator equipment.

[b]Stanley Consultants (1979).

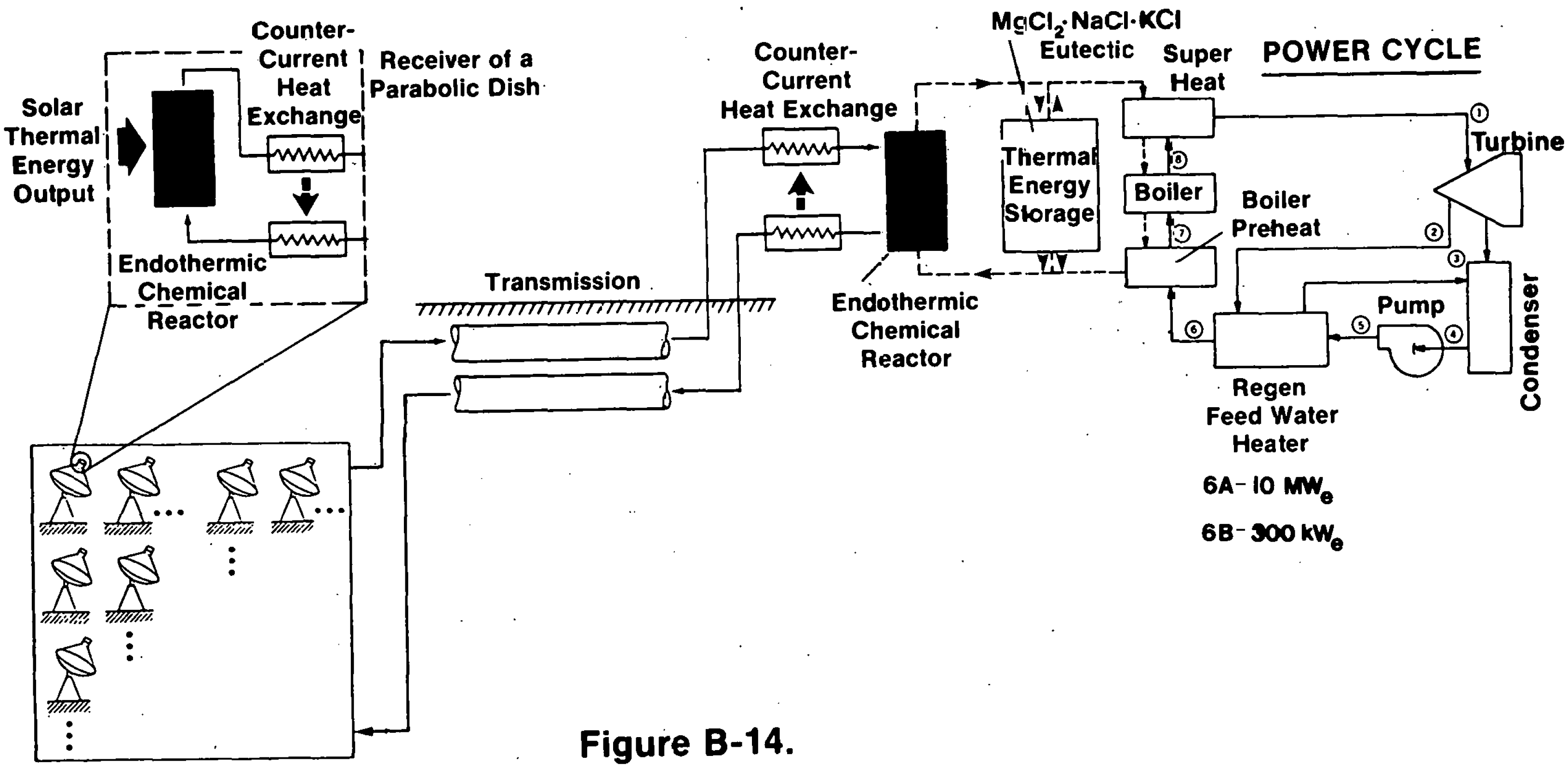

Solar Thermal Energy Output
Counter-Current Heat Exchange
Receiver of a Parabolic Dish
Endothermic Chemical Reactor
Transmission
Counter-Current Heat Exchange
Endothermic Chemical Reactor
MgCl2·NaCl·KCl Eutectic
Thermal Energy Storage
Super Heat
Boiler
Boiler Preheat
POWER CYCLE
Turbine
Condenser
Pump
Regen Feed Water Heater
6A-10 MWe
6B-300 kWe

Figure B-14.

Table B-14. Cost Algorithms[a]

<table>
<tr><th>Element</th><th>Account # 4400</th><th>Account # 4500</th></tr>
<tr><td>1A</td><td>80A</td><td>$135A + 300A^{0.8}$</td></tr>
<tr><td>1B Molten Salt</td><td>80A</td><td>$6.25A + 163A^{0.8}$</td></tr>
<tr><td>1B Liquid Metal</td><td>80A</td><td>$3.34A + 3497A^{0.8}$</td></tr>
<tr><td>2</td><td>80A</td><td>$22.5A + 502A^{0.8}$</td></tr>
<tr><td>3A</td><td>80A</td><td>$9.43A + 691A^{0.8}$</td></tr>
<tr><td>3B</td><td colspan="2">90A[c]</td></tr>
<tr><td>3C</td><td colspan="2">100A[c]</td></tr>
<tr><td>4A</td><td>80A</td><td>$13.5A + 300A^{0.8}$</td></tr>
<tr><td>4B[b]</td><td colspan="2">$141.98A + 753.3A^{0.75}$[c]</td></tr>
<tr><td>4C</td><td></td><td>$5.34A + 269A^{0.8}$</td></tr>
<tr><td>5</td><td>80A</td><td>$3.34A + 349A^{0.8}$</td></tr>
<tr><td>6A</td><td colspan="2" rowspan="2">100A[c]</td></tr>
<tr><td>6B</td></tr>
</table>

A = Collector Area in square meters.

[a]Cost algorithms are for Direct Cost only.

[b]Algorithms for Element 4B is for all of the cost accounts.

[c]Algorithm represents the sum of both accounts (4400 and 4500).

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SERI

APPENDIX C

CONVERSION EFFICIENCIES OF LARGE & SMALL SOLAR THERMAL POWER SYSTEMS

Stearns-Roger
Report

CONVERSION EFFICIENCIES OF LARGE AND SMALL SOLAR THERMAL POWER SYSTEMS

FINAL REPORT

PREPARED FOR THE

SOLAR ENERGY RESEARCH INSTITUTE

1536 COLE BOULEVARD

GOLDEN, COLORADO 80401

CONTRACT NO. AP-9-8035-1

BY

STEARNS-ROGER SERVICES CO.

DENVER, COLORADO

JUNE 1979

INTRODUCTION

Stearns-Roger Service Co. has performed a study of a number of different thermodynamic electric power generating cycles under funding by the Solar Energy Research Institute (SERI) of Golden, Colorado.

The purpose of the study was to provide energy conversion system efficiency data as a function of maximum system temperature for each of the thermodynamic cycles as defined in the contract Statement of Work. The energy conversion system was assumed to be that portion of a solar power system containing the prime mover.

The data generated by this study will be used by SERI in evaluating the effects of various solar thermal transport and storage systems on the power generating system.

The approach used in the analysis and presented in this report was to:

1. Calculate the performance of the reference cycle as provided by SERI in the work statement.

2. Maintain a constant cycle heat input and rejection temperature for each non-reference condition as determined from the reference cycle.

3. Analyze each non-reference system assuming that it is operating at its design point. No "off-design" conditions were analyzed.

4. Assume component efficiencies based on current design practice derived from the literature or from Stearns-Roger power plant experience.

5. Develop a schematic of the cycle components together with the cycle state points for each of the reference cycles.

6. Calculate cycle performance for variable maximum cycle temperatures for each of the reference cycles. These data are presented as curves of cycle efficiency and ratios of cycle efficiency to reference cycle efficiency vs. maximum temperature.

7. Describe each cycle together with the assumptions used and cycle limitations.

A total of eleven cycles were evaluated for this study including steam Rankine (reheat and non-reheat), open and closed Brayton, organic Rankine and a total energy system using a steam Rankine cycle.

Existing or specially developed digital computer programs were used to perform the individual cycle calculations.

The report is divided into eleven Sections. Each Section contains a brief description of the cycle analyzed with the assumptions used, and the cycle schematic and efficiency curves.

No conclusions or comparisons between cycles are drawn from this study since the sole purpose of the study is to present cycle performance data to be used by SERI in a further study of solar power generating storage systems.

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FIG. NO. PG. NO.

LIST OF FIGURES (CONT'D)

FIG. NO. PG. NO.

LIST OF FIGURES (CONT'D)

FIG. NO. PG. NO.

SECTION 1
ELEMENT 1A - STEAM RANKINE NON-REHEAT CYCLE

1.1 INTRODUCTION

Cycle heat balances were performed for a 100 MWe (nominal) steam Rankine, non-reheat cycle incorporating a five heater feedwater heating system. The cycle was duplicated from that of the Barstow 100 MWe Commercial Plant using steam from a solar receiver.

1.2 CYCLE DESCRIPTION

For operation on receiver steam the throttle conditions are 1465 psia (10.10×10^6 Pa) and 950°F (510°C) for the reference cycle. At these throttle conditions and constant final feedwater conditions of 2600 psia (17.93×10^6 Pa) and 425.5°F (218.6°C), a heat input to the cycle was determined. This heat input was held constant for the varying throttle temperatures and pressures studied. The throttle temperatures were varied from the reference cycle conditions down to a temperature 825°F (441°C) which yielded approximately 84 percent minimum quality steam leaving the last stage of the turbine, and up to 1100°F (593°C) (the upper limit for existing steam turbine technology). In actual operation, it is expected that the last stage quality will not be permitted to drop significantly below 88 percent. Generator output was allowed to vary with throttle conditions.

A cycle schematic showing all component efficiencies is presented in Figure 1—1, with a plot of cycle efficiency versus throttle temperature at different pressures shown in Figure 1—2. The turbine and pump efficiencies were obtained from data given in Reference 1. The cycle efficiencies are presented as gross (total energy output divided by total energy input to the cycle) and net (assuming 8 percent of total energy output including pumping power goes to auxiliary demand). A plot of the ratio of non-reference cycle efficiency to reference cycle efficiency versus throttle temperature and pressure is shown in Figure 1—3, and generator output (gross and net) versus throttle temperature, in Figure 1—4.

The turbine used in this model is a standard-frame General Electric utility, non-reheat steam turbine exhausting at an assumed 2.5 in. HgA (0.0984 mm HgA) to a tube and shell condenser. The turbine's five extractions are connected to three closed high-pressure heaters, an open deaerating heater and one closed low-pressure heater operating at various pressures. The heater operating characteristics were duplicated from the Barstow Commercial Solar Plant as documented in Reference 1, and held constant for the variable throttle conditions. The calculation procedure used to analyze this system is as described in the appendix and Reference 2.

For the case of steam supplied entirely from thermal storage (admission steam), the steam was admitted to the turbine downstream of normal throttle steam (receiver steam). Because of the point of admission and the low thermodynamic properties of the steam (365 psia (2.52×10^6 Pa) and 565°F (292°C)), the three top heaters are taken out of service for this operating mode. A flow of 3 to 5 percent of admission steam is required for cooling the high-pressure turbine stages bypassed by the admission steam. This steam does no work in the high-pressure stages, however, it does perform work as it recombines with the admission steam.

Again, the heat input to the reference cycle was determined and held constant for the non-reference cases. Also, the admission steam temperature was varied from the reference cycle admission temperature down to a temperature of 500°F (260°C) yielding a minimum 84 percent steam quality leaving the turbine, up to 925°F (496°C). Again, the last stage quality will actually be a minimum of approximately 88 percent. At the constant heat input, several throttle pressures were studied to illustrate the effects on the cycle if the admission point were varied up to the normal (receiver operation) pressure (1465 psia).

A reference cycle diagram for the admission steam is shown in Figure 1—5. Plots of the gross and net cycle efficiencies versus throttle temperature at different pressures are shown in Figure 1—2, assuming 6 percent auxiliary power usage. A plot of the ratio of non-reference cycle efficiency to reference cycle efficiency versus throttle temperature and pressure is shown in Figure 1—3, and a plot of generator output (gross and net) versus throttle temperature and pressure, in Figure 1—4.

REFERENCES

1. McDonnell Douglas Astronautics Company, "Central Receiver Solar Thermal Power System, Phase 1," (Volume 6, EPGS, MDC-G-6776), October 1977.

2. General Electric Company, "A Method for Predicting the Performance of Steam Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

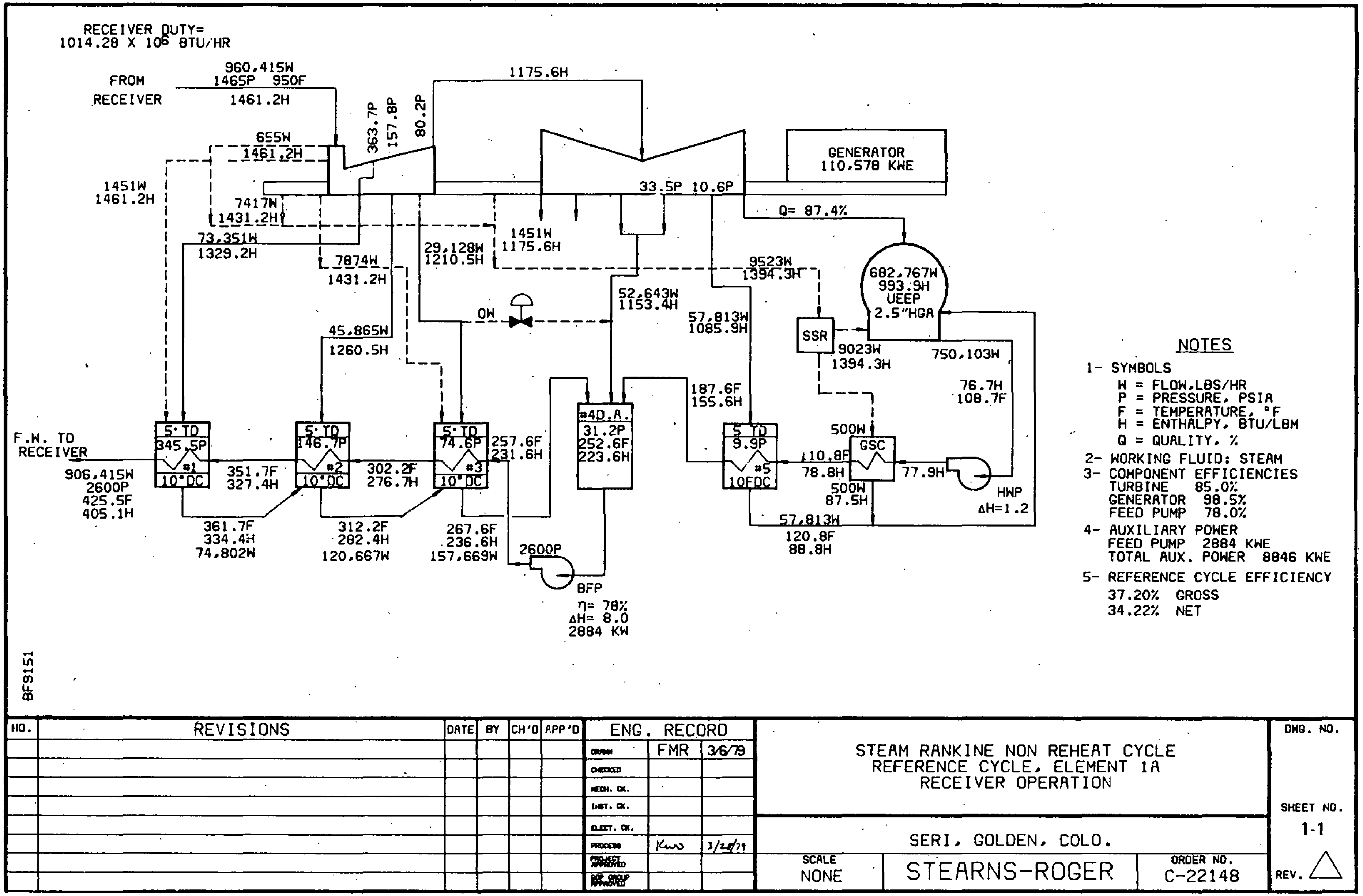


Figure 1—1. ELEMENT 1A - STEAM RANKINE NON-REHEAT CYCLE, RECEIVER OPERATION

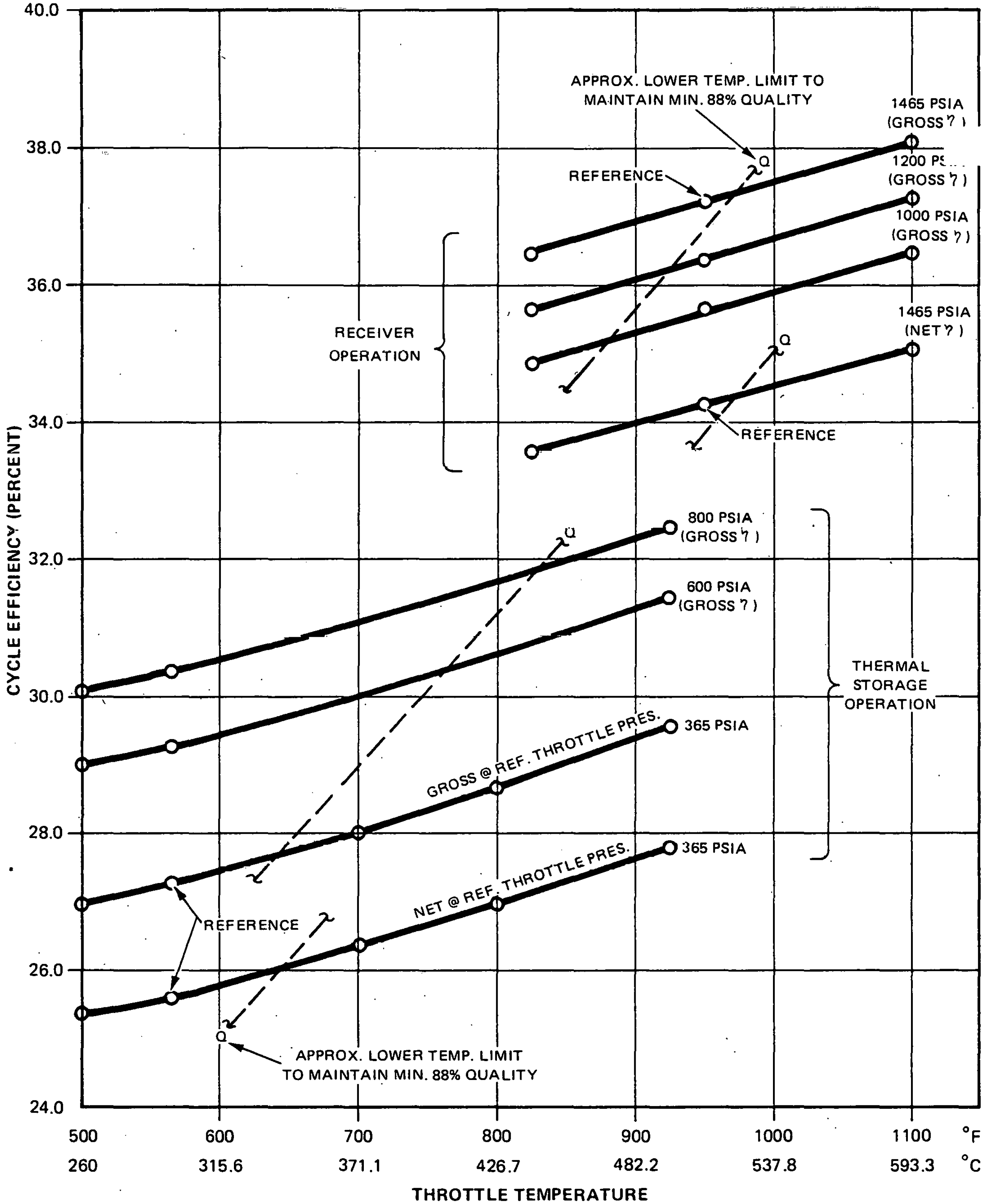


Figure 1–2. ELEMENT 1A - STEAM RANKINE NON-REHEAT CYCLE, CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

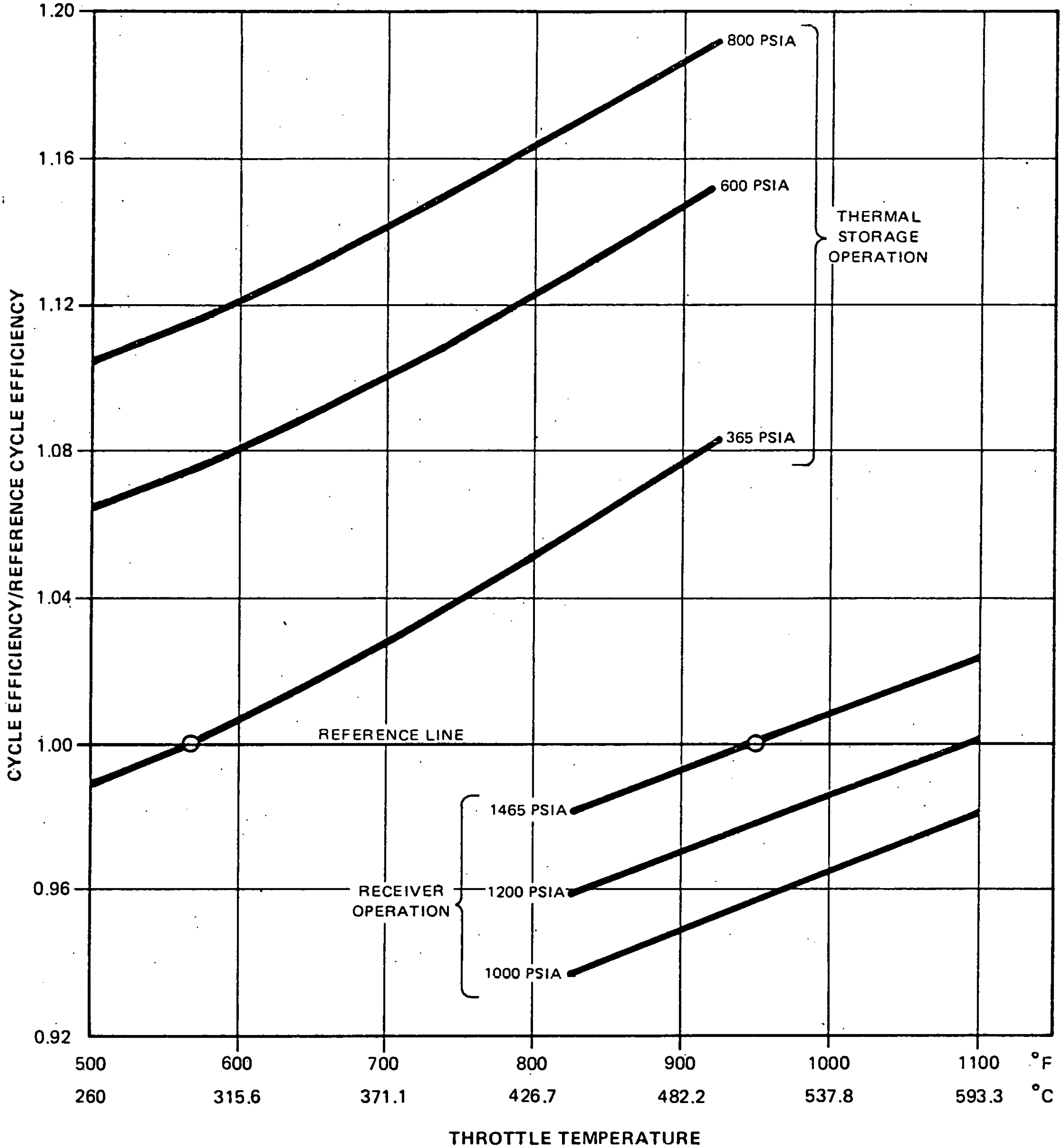


Figure 1–3. ELEMENT 1A - STEAM RANKINE NON-REHEAT CYCLE, REFERENCE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

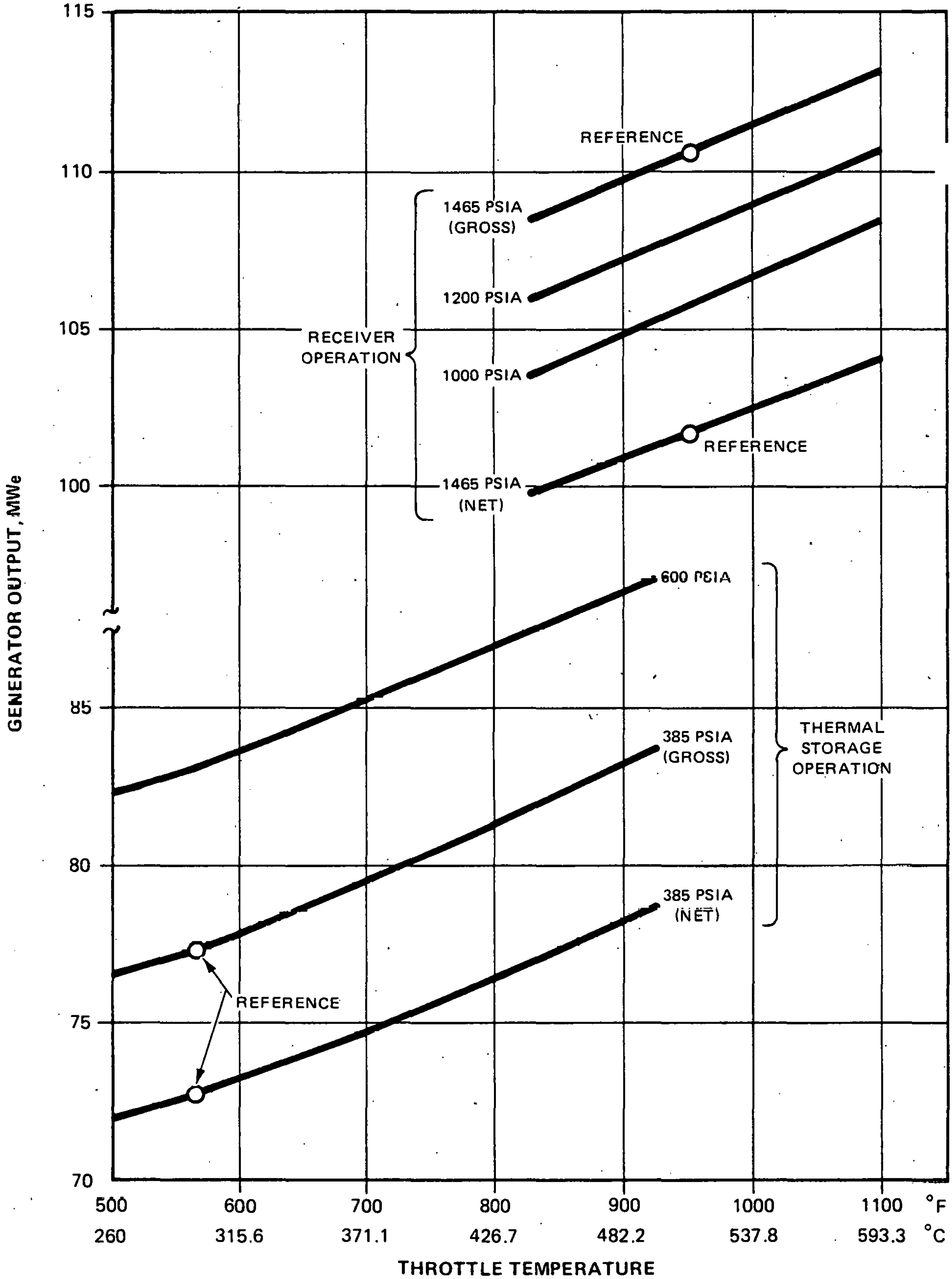


Figure 1—4. ELEMENT 1A - STEAM RANKINE NON-REHEAT CYCLE, GENERATOR OUTPUT VS. THROTTLE TEMPERATURE

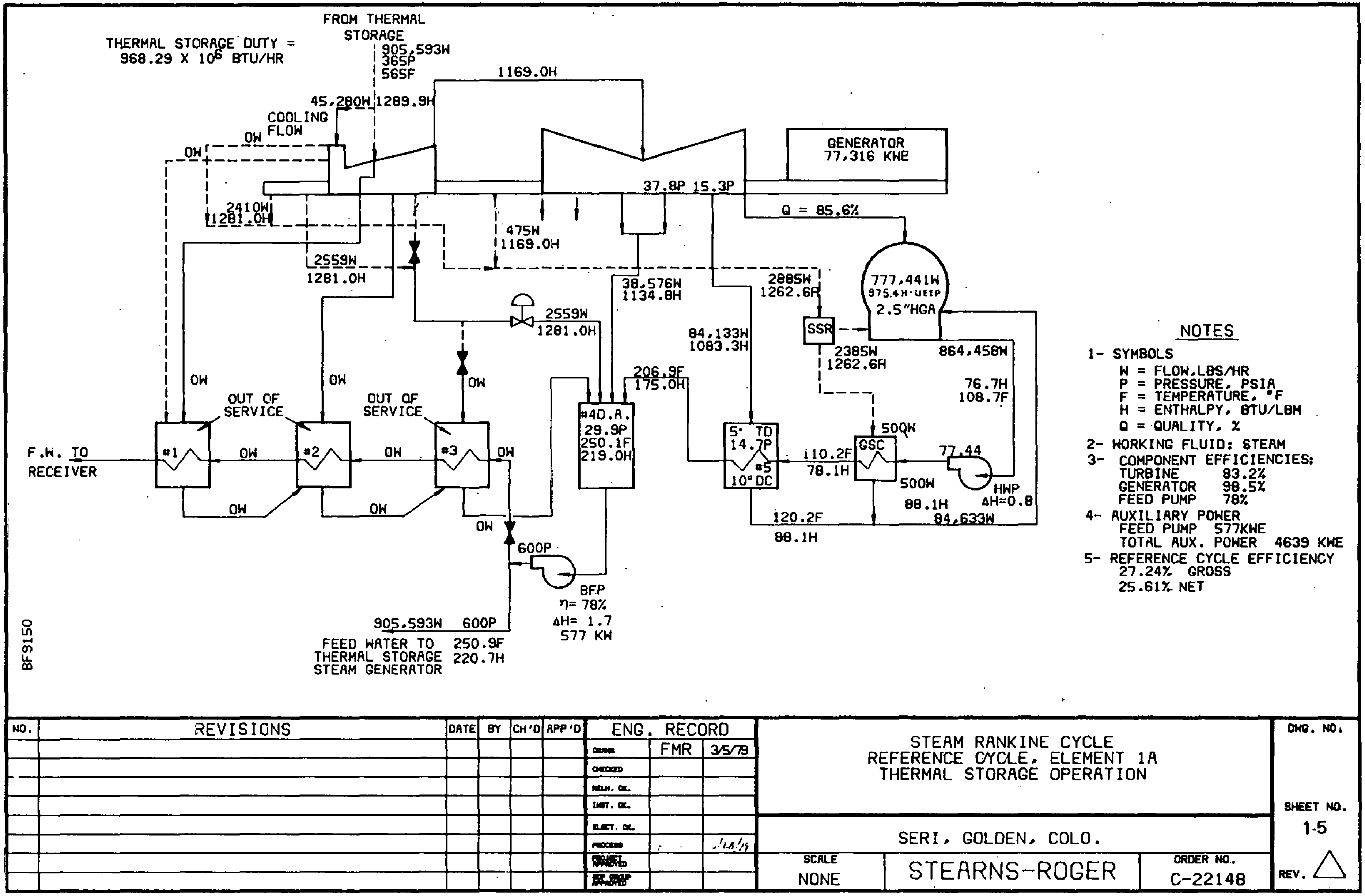


Figure 1–5. ELEMENT 1A - STEAM RANKINE NON-REHEAT CYCLE, THERMAL STORAGE OPERATION

SECTION 2
ELEMENT 1B - REHEAT STEAM RANKINE CYCLE

2.1 INTRODUCTION

Cycle heat balances were performed for a 100 MWe (nominal) reheat steam Rankine cycle employing six stages of regenerative feedwater heating. The cycle is based on the Advanced Central Receiver Power System using steam generated by a liquid metal solar receiver system. The turbine exhausts to a condenser at 2.5 inches HgA (8.46×10^3 Pa).

The cycle was analyzed using an in-house computer program (D135B), which performs a mass and energy balance around the specified cycle. The shape of the turbine expansion curve is as specified in Figure 25 in Reference 1. The program accesses subroutines to calculate the fluid state conditions around the cycle using the relationships specified in the ASME steam tables.

2.2 CYCLE DESCRIPTION

The throttle conditions used for the reference cycle are 2400 psia (16.55×10^6 Pa) and 1000°F (537.8°C), with the reheat temperature of 1000°F (537.8°C). The final feedwater temperature selected is 480°F (248.9°C), which allows a reasonable pressure ratio across the high pressure turbine. A 10 percent pressure loss is assumed across the steam generator, and a pressure loss of 15 percent is assumed across the reheater.

For this study the total heat input to the cycle was held constant while the throttle conditions and reheat temperatures were varied. The throttle and reheat temperatures were varied over the range of 800°F to 1100°F (426.7°C to 593.3°C) and the throttle pressure was varied over the range of 1250 psia to 2400 psia (8.62×10^6 Pa to 16.55×10^6 Pa). The lower temperature limit was selected to limit the turbine exhaust steam quality to 88 percent and the upper temperature limit was selected as the limit of existing steam turbine technology. The throttle pressures selected are those normally used in the power industry. Representative high pressure and low pressure turbine efficiencies were calculated using the method specified in Reference 1. The representative turbine efficiencies were based on the throttle conditions of the reference cycle. The turbine efficiencies were held constant over the range of throttle conditions in order to prevent distortion of the effect of throttle conditions on overall cycle efficiency. In reality, the efficiency of the turbines will increase slightly as the amount of superheat of the throttle steam increases. Generator output was allowed to vary with the throttle conditions.

Turbine extractions provide steam to three high-pressure closed feedwater heaters, an open deaerating feedwater heater, and two low-pressure closed feedwater heaters. The heater performance characteristics are derived from standard design values and are held constant over the range of throttle conditions. Most turbine steam leakages are not accounted for in the cycle, as these leakages are small when compared to other cycle flows. One leakage was included, the shaft leakage from the high pressure turbine to the low pressure turbine, as this is three percent of the throttle flow. The neglected leakages include shaft leakage from the exhaust of the HP turbine, the sealing flows to the LP turbine shaft seals, and packing leakage from the turbine stop and control valves. The total of these leakages is typically less than one percent of the throttle flow in current commercial units.

A cycle schematic showing component efficiencies and cycle flow data is shown in Figure 2—1 for the base cycle. Plots of cycle efficiency versus throttle temperature are shown in Figure 2—2 and Figure 2—3. A normalized representation of cycle efficiency with respect to base cycle efficiency versus throttle temperature is shown in Figure 2—4. A plot of generator output versus throttle temperature is shown in Figure 2—5. Gross cycle efficiency is defined as the total cycle input energy divided into the generator output. Net cycle efficiency is defined as the total cycle input energy divided into the generator output less the plant auxiliary power requirement (which is assumed to be 8 percent of the generator output). Plant auxiliary power includes that power used by the boiler feed pump, condensate pump, circulating water pump, controls, plant lighting, plant HVAC, solar collector field usage, cooling tower fans, etc.

REFERENCE

1. General Electric Company, "A Method for Predicting the Performance of Steam Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

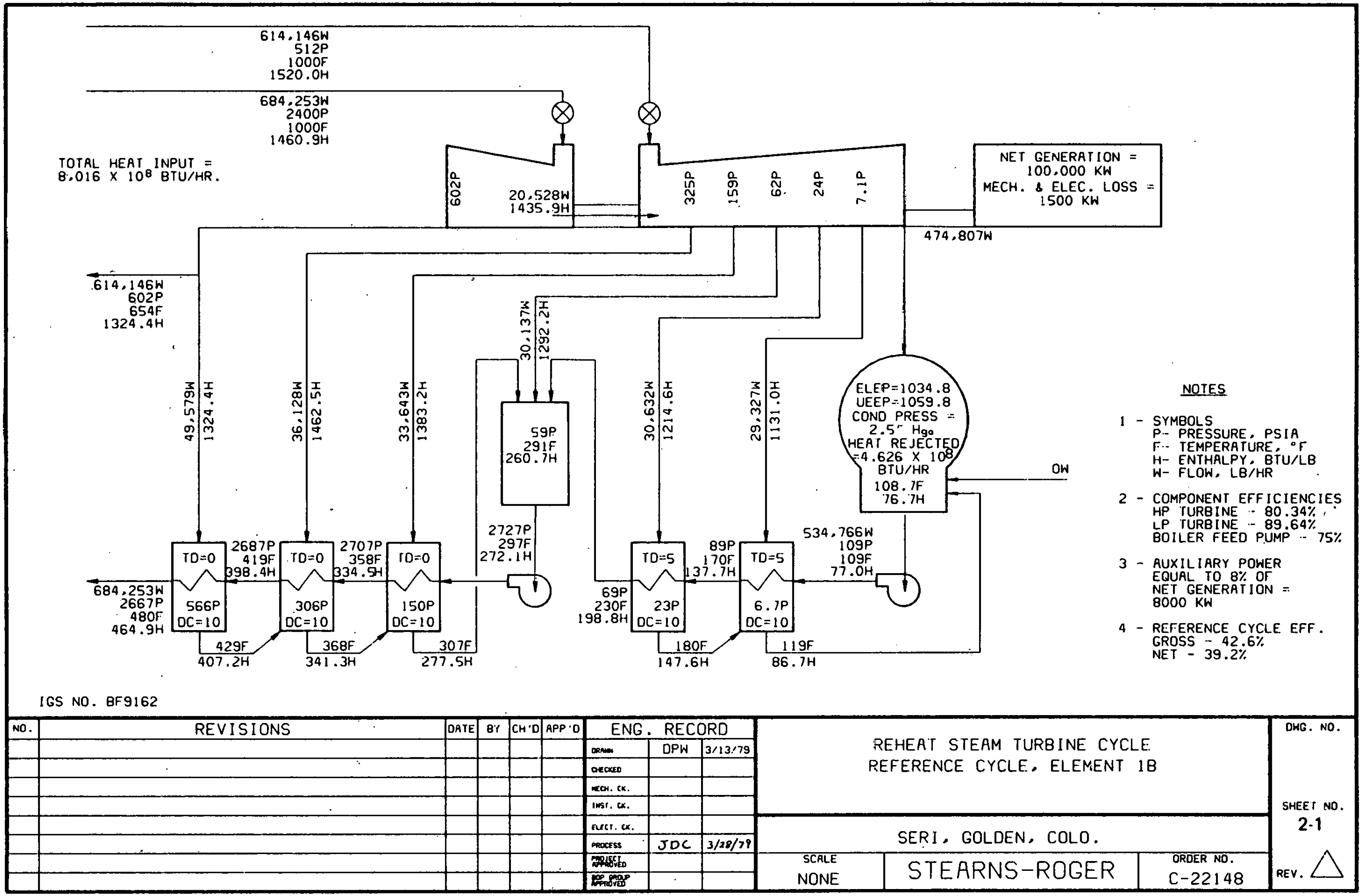


Figure 2–1. ELEMENT 1B - REHEAT STEAM RANKINE CYCLE, REFERENCE CYCLE

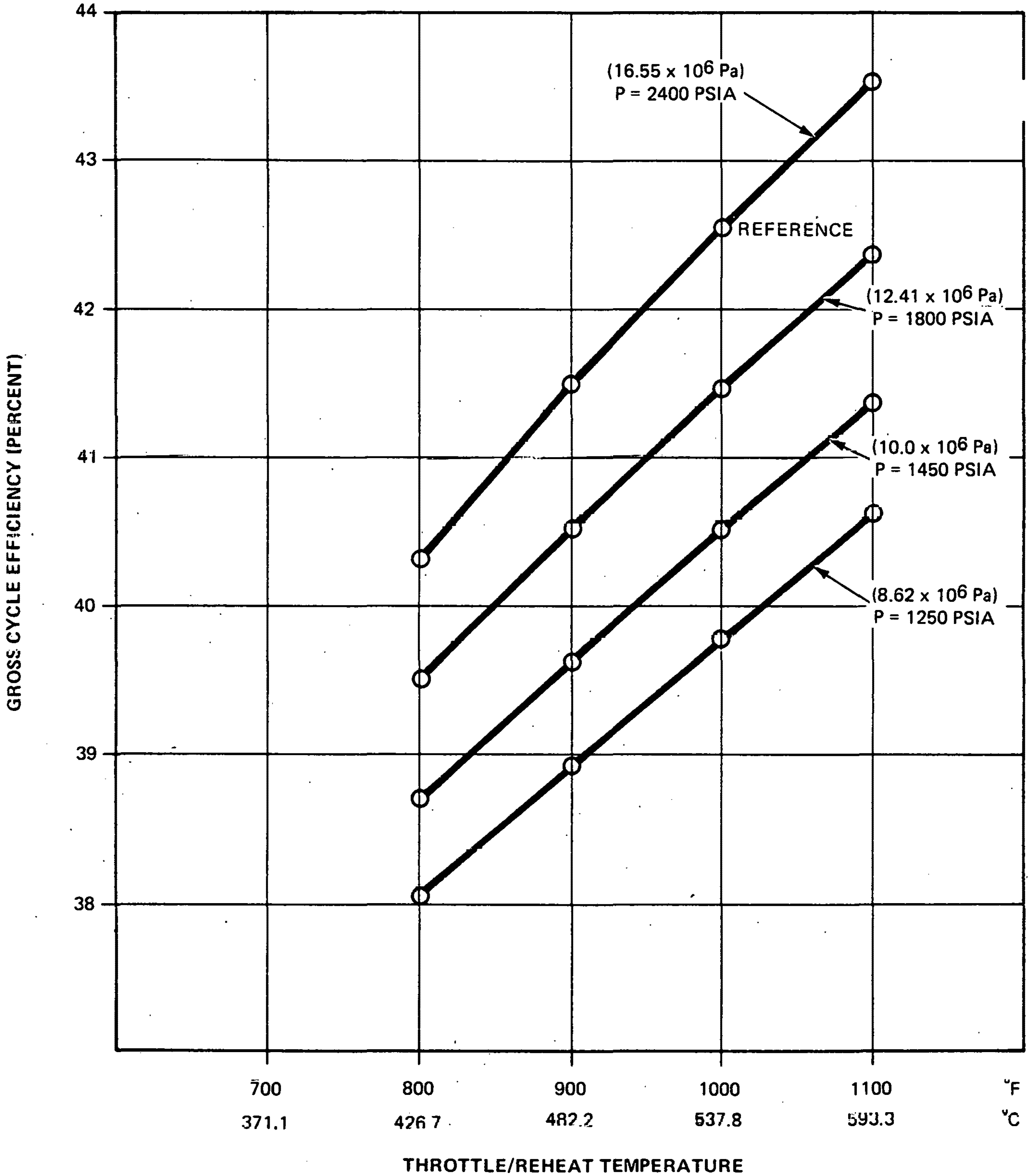


Figure 2–2. ELEMENT 1B - REHEAT STEAM RANKINE CYCLE, GROSS CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

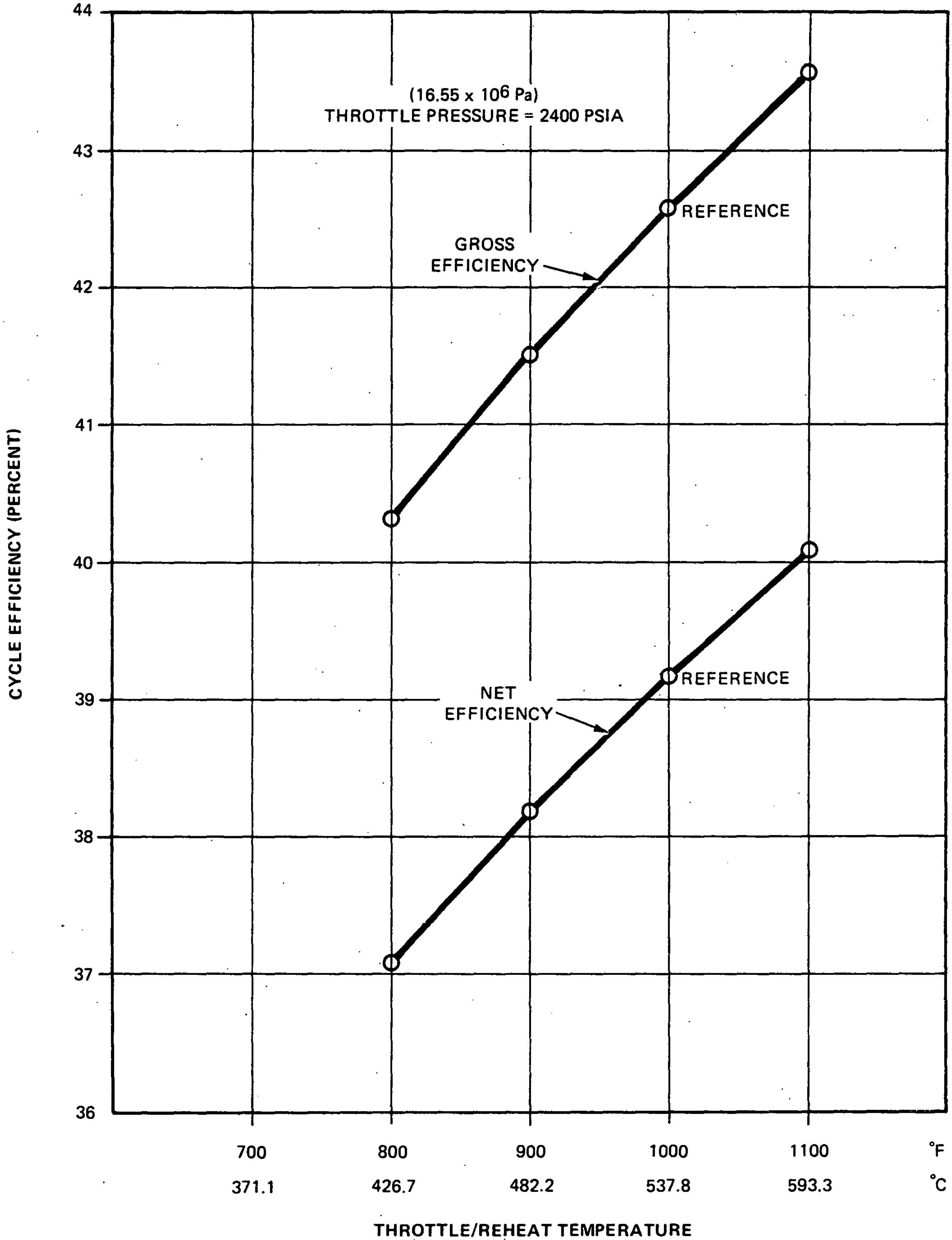


Figure 2–3. ELEMENT 1B - REHEAT STEAM RANKINE CYCLE, CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

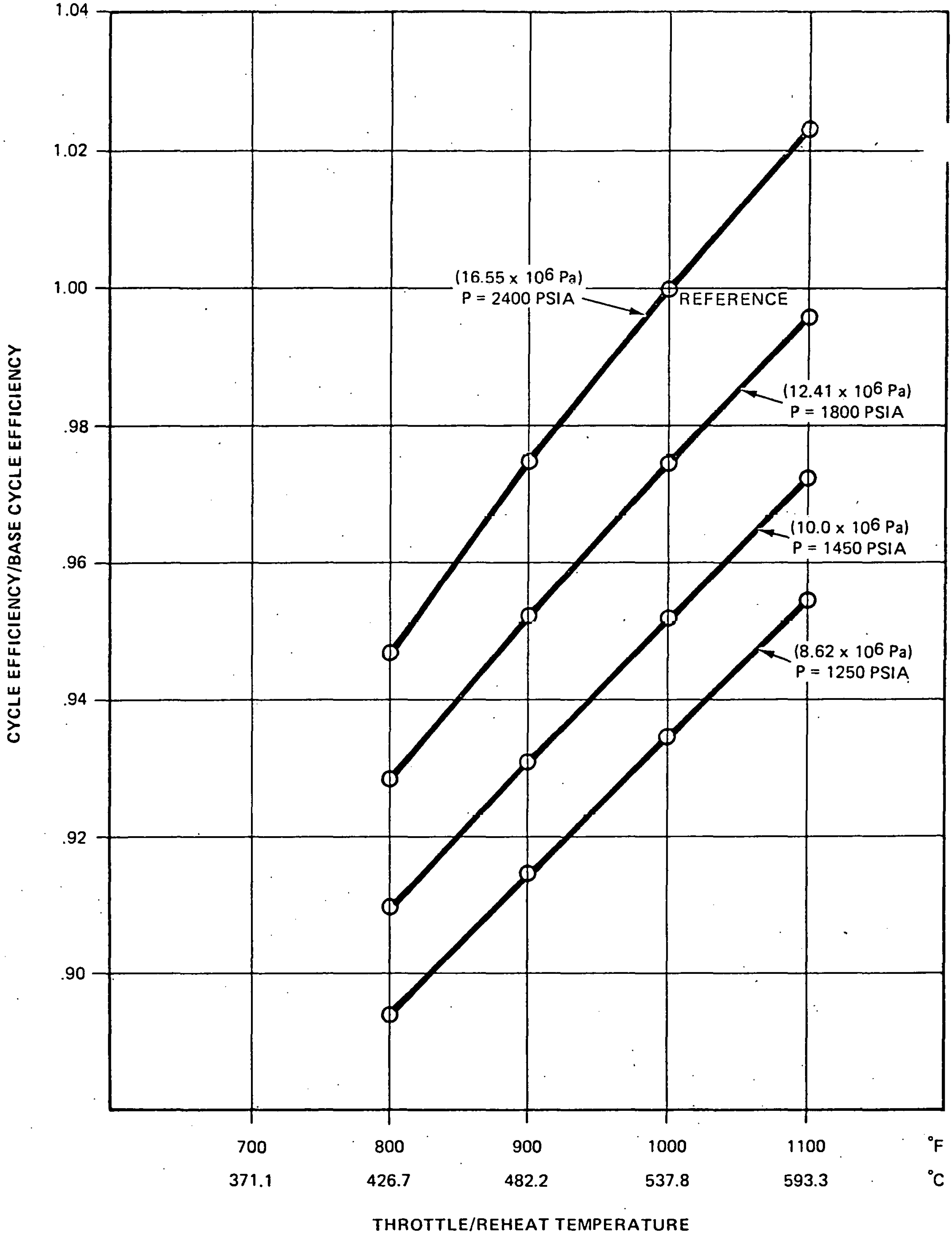


Figure 2—4. ELEMENT 1B - REHEAT STEAM RANKINE CYCLE, RELATIVE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

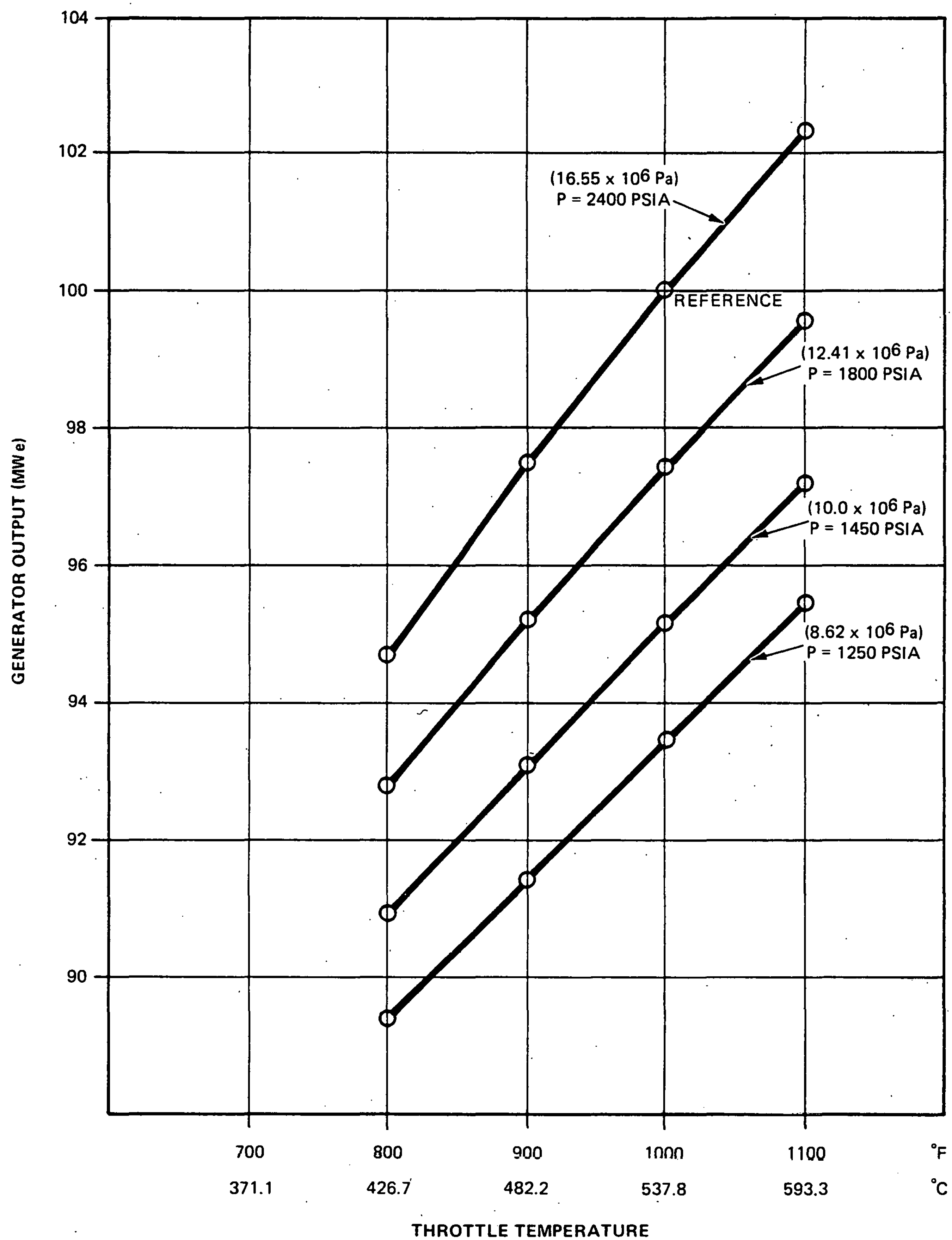


Figure 2–5. ELEMENT 1B - REHEAT STEAM RANKINE CYCLE, GENERATOR OUTPUT VS. THROTTLE TEMPERATURE

SECTION 3
ELEMENT 2 - OPEN REGENERATIVE BRAYTON CYCLE

3.1 INTRODUCTION

Cycle heat balances were performed for a 100 MWe (nominal) open regenerative, Brayton cycle. Heat input to the cycle is from a solar receiver/thermal storage system and from an oil fired air heater.

The cycle was analyzed using a computer program based on thermodynamic relationships contained in Reference 1 to perform the required mass and energy balance around the cycle. The program assumes a constant specific heat for the working fluid.

3.2 CYCLE DESCRIPTION

The reference cycle for this element is composed of a compressor with an efficiency of 80 percent and a pressure ratio of 4.75. Inlet air is compressed and discharged to the recuperator. Cooling air for the turbine is diverted from the compressor discharge to various parts of the turbine. Cooling air which is injected into the turbine in the flow path does useful work, while cooling air going to the turbine casing does no work. A general industry guide was used to determine how much of the total system flow was used as cooling air which did no work in the turbine. This guide states that cooling air is required at a turbine inlet temperature of 1700°F (926.7°C) and will be one percent of the compressor flow for every 100°F (55.6°C) of temperature increase.

The compressor flow is ducted to the recuperator where it is heated by the turbine exhaust air. For this cycle, the recuperator effectiveness is 90 percent and a pressure drop of two percent is assumed across the recuperator. The air temperature is further increased by the solar receiver/thermal storage system to a temperature of 1250°F (676.7°C). A pressure drop of three-and-one-half percent is assumed across the solar system. The air is finally heated to 2000°F (1093.3°C) by the oil-fired air heater. A pressure drop of one percent is assumed across the heater. The hot air is expanded through the turbine and produces shaft work which drives the compressor and the generator. The turbine efficiency used for this cycle is 90 percent. The air from the turbine exhaust passes through the recuperator where it is cooled by air from the compressor. A pressure drop of two percent is again assumed across the recuperator. The spent air is then exhausted to the atmosphere.

The turbine and compressor efficiencies, the recuperator effectiveness, and the amount of cooling air required are based on verbal information from several industry sources, Reference 2. These values are considered to be conservative.

For this study, the generator output was held constant for the non-reference cycles while the pressure ratio of the compressor was varied over the range of 2 to 10. The turbine inlet temperature was held at a constant 2000°F (1093.3°C). As the compressor pressure ratio is increased, the pressure ratio of the turbine also increases. The exhaust temperature of the compressor increases and the exhaust temperature of the turbine decreases with increasing pressure ratios. The overall effect is to reduce the duty of the recuperator. This relationship is shown in Figure 3—1.

For any specific compressor ratio the relationship between the temperature of the air out of the solar receiver/thermal storage system and amount of oil burned in the air heater can be calculated. In defining this relationship it is assumed that the generator output and the turbine inlet temperature are constant. This relationship is shown in Figure 3—2.

The cycle schematic showing component efficiencies and reference cycle flow data is shown in Figure 3—3. A plot of cycle efficiency versus compressor pressure ratio is shown in Figure 3—4. A normalized representation of cycle efficiency with respect to base cycle efficiency versus compressor pressure ratio is shown in Figure 3—5.

Gross cycle efficiency is defined as the total cycle input energy divided into the generator output. Net cycle efficiency is defined as the total cycle input energy divided into the generator output less the plant auxiliary requirement (which is assumed to be 8 percent of the generator output).

REFERENCES

1. "Gas Dynamics", A. D. Lewis, 1964.

2. Telephone conversations with manufacturers concerning turbine efficiencies, compressor efficiencies, recuperator effectiveness, and cooling flows.

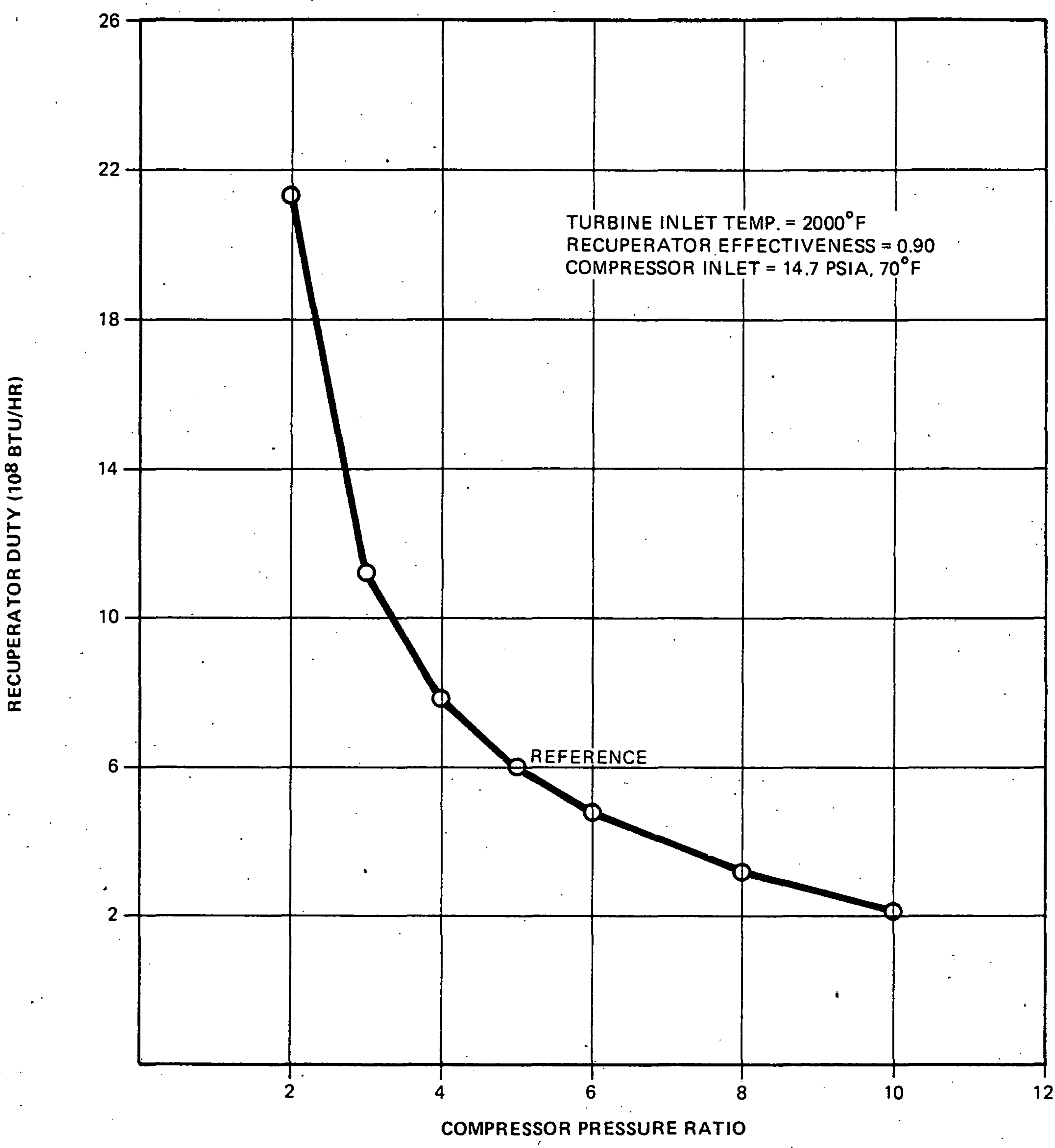


Figure 3–1. ELEMENT 2 - OPEN REGENERATIVE CYCLE, RECUPERATOR DUTY VS. COMPRESSOR PRESSURE RATIO

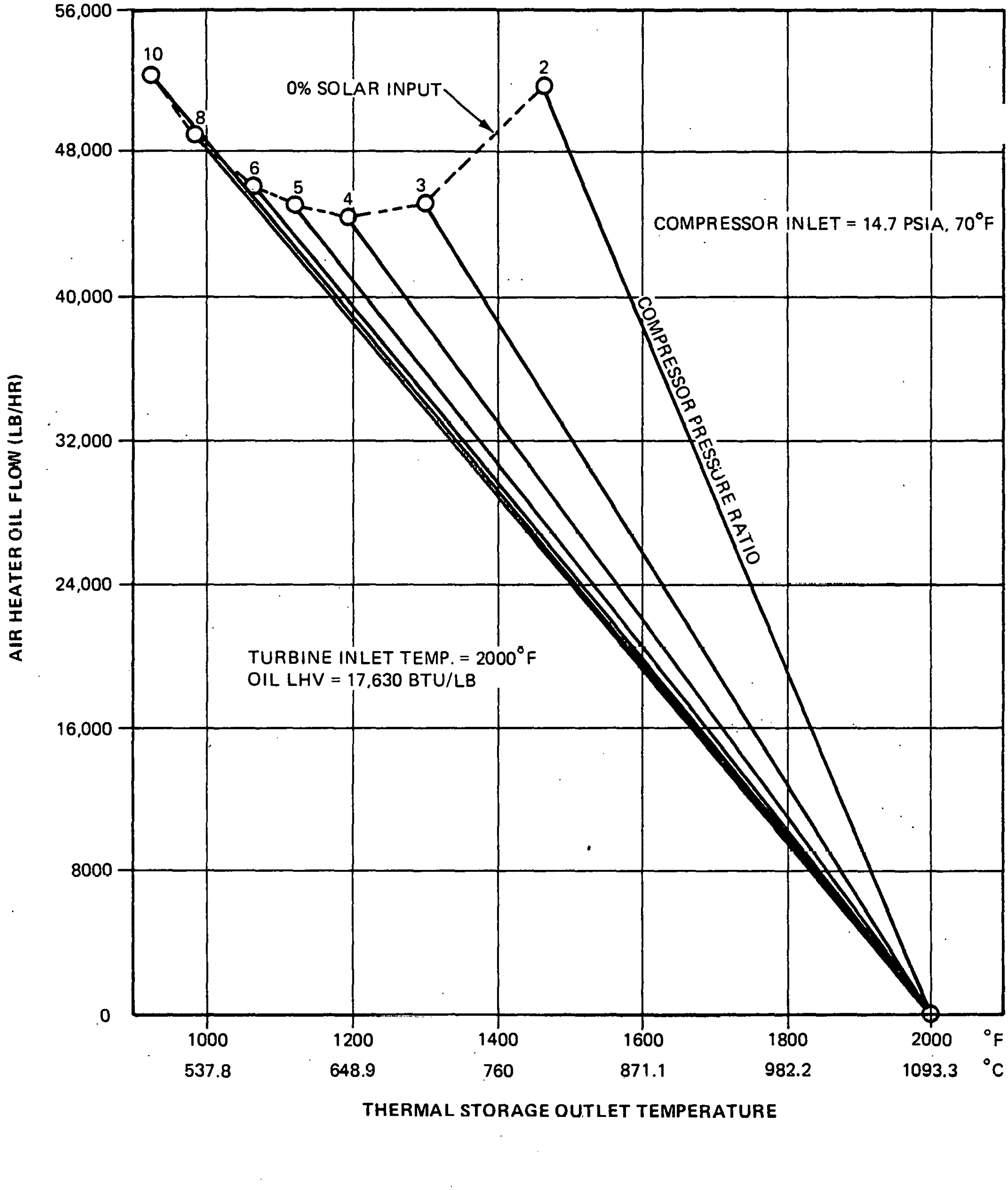


Figure 3–2. ELEMENT 2 - OPEN REGENERATIVE CYCLE, AIR HEATER OIL FLOW VS. THERMAL STORAGE OUTLET TEMPERATURE

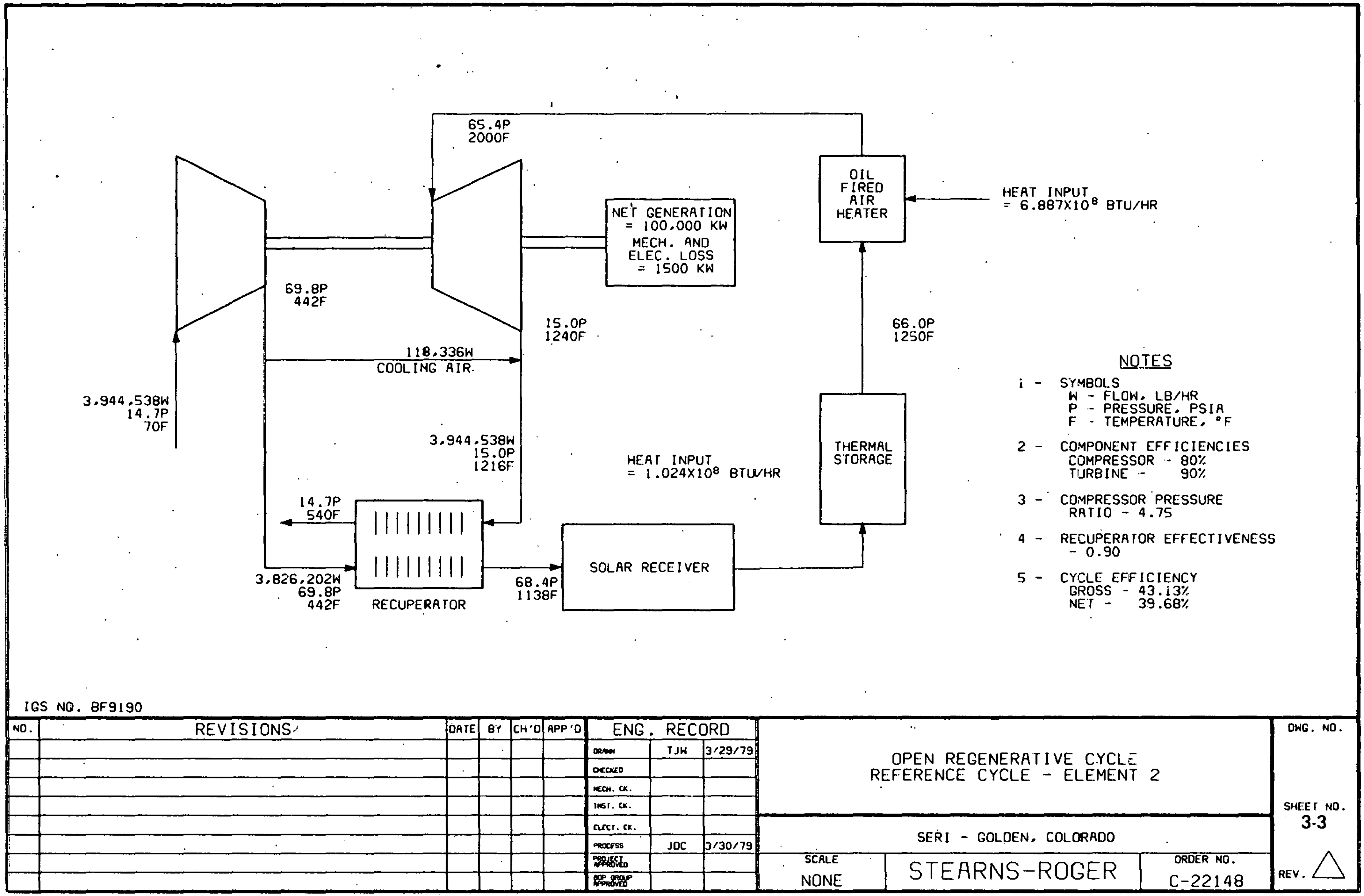


Figure 3–3. ELEMENT 2 - OPEN REGENERATIVE CYCLE

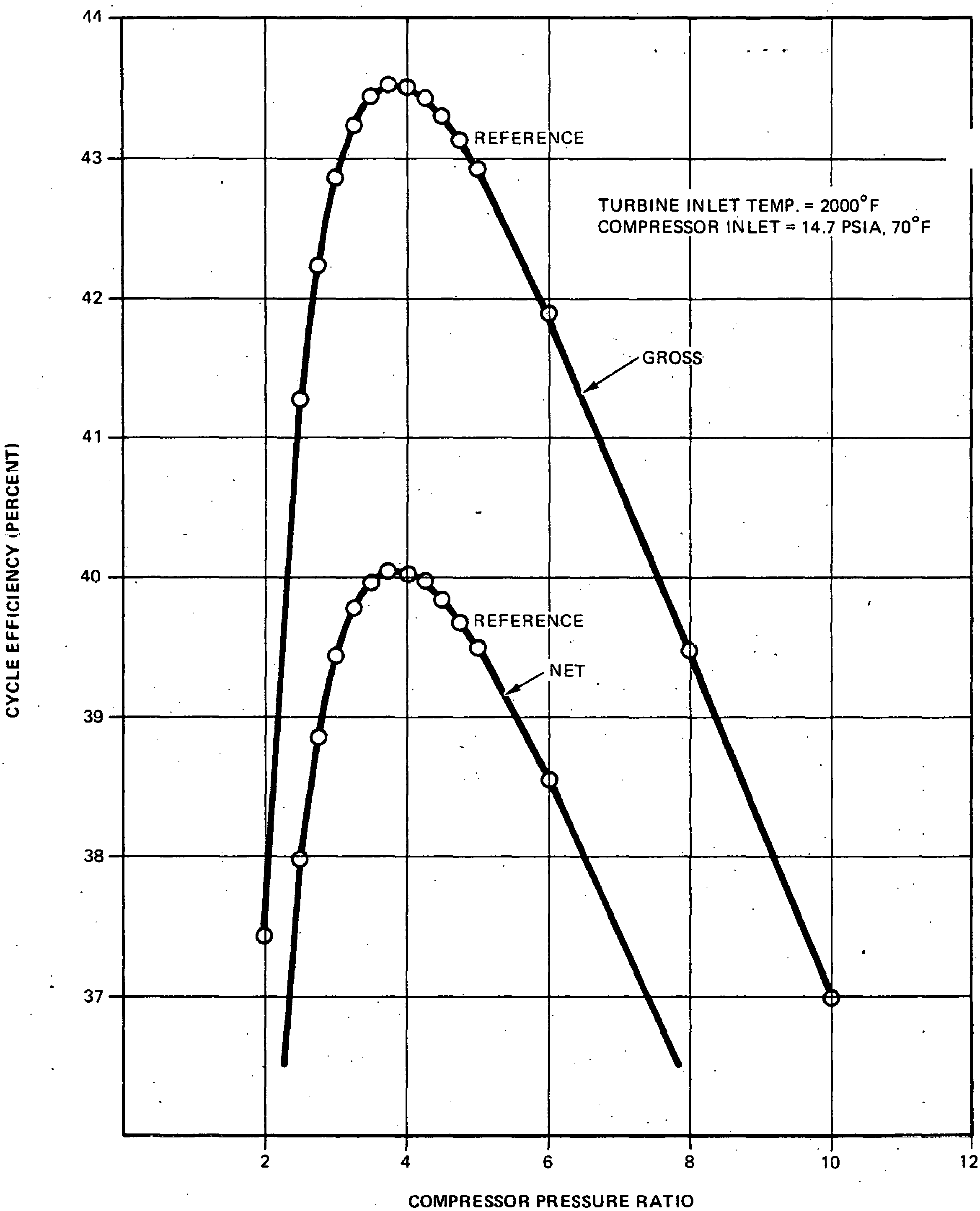


Figure 3–4. ELEMENT 2 - OPEN REGENERATIVE CYCLE, CYCLE EFFICIENCY VS. COMPRESSOR PRESSURE RATIO

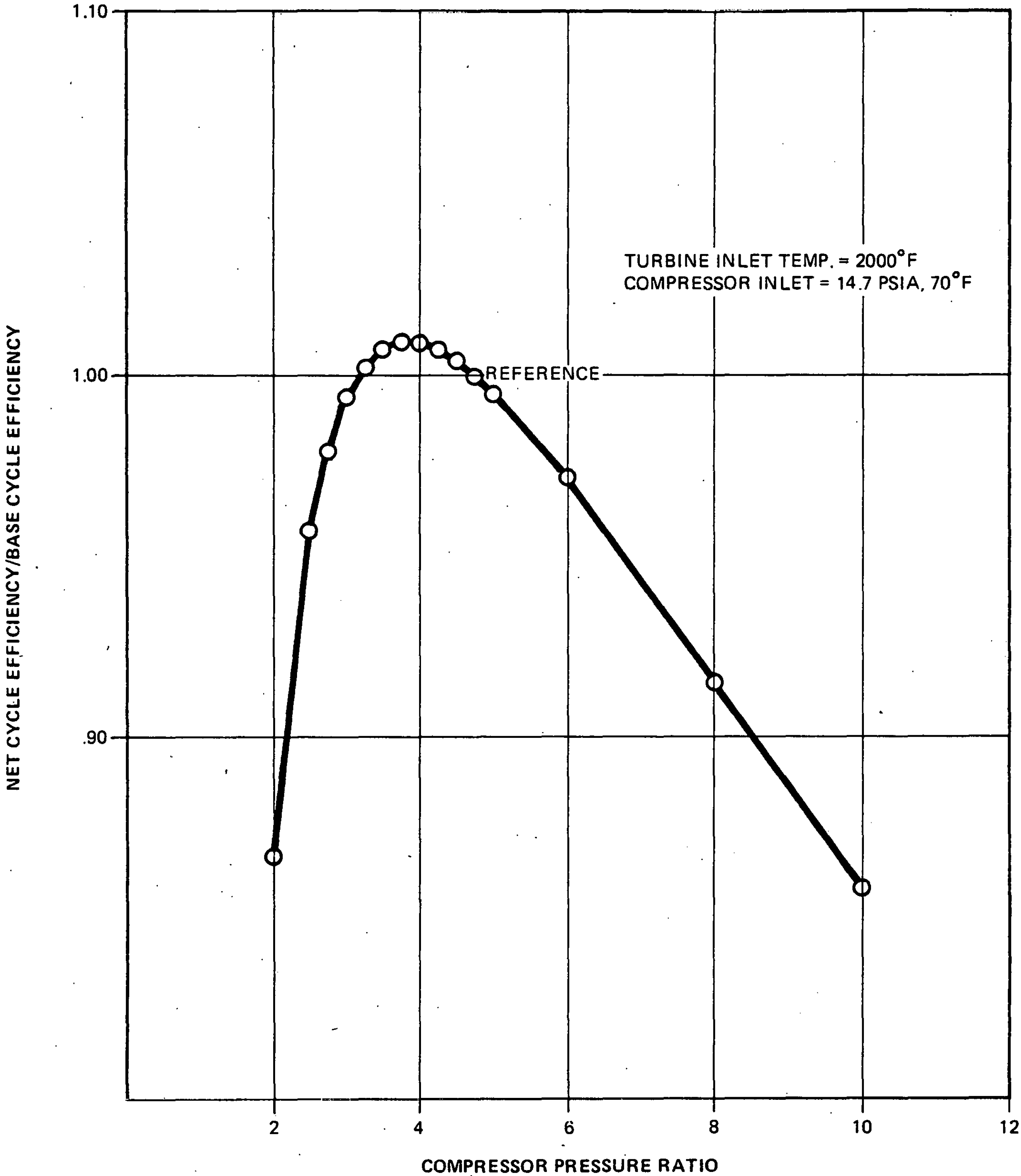


Figure 3–5. ELEMENT 2 - OPEN REGENERATIVE CYCLE,
RELATIVE CYCLE EFFICIENCY VS. COMPRESSOR PRESSURE RATIO

SECTION 4
ELEMENT 3A - STEAM RANKINE NON-REHEAT CYCLE

4.1 INTRODUCTION

This cycle is similar to Element 1A, but is modeled after the Barstow 10 MWe Pilot Plant, Reference 1. A four-heater feedwater system (including two closed high-pressure heaters, one open deaerating heater and one closed low-pressure heater) is utilized. The gland steam condenser is eliminated from the feedwater system for this cycle configuration.

4.2 CYCLE DESCRIPTION

For throttle steam supplied from the receiver, the steam conditions are 1465 psia (10.10×10^6 Pa) and 950°F (510°C). The turbine exhaust conditions and basis for throttle temperature and pressure variations are identical to Element 1A for receiver operation. The throttle temperatures were varied from 800°F (427°C) up to 1100°F (593°C), while holding total heat input to the cycle constant. These calculations were based on the method described in the Appendix and Reference 2.

For the case of operation from thermal storage, the admission steam conditions are 384.7 psia (2.65×10^6 Pa) and 525°F (274°C), with 5 percent of admission steam used for cooling the high-pressure turbine. The two top heaters are out of service for operation from thermal storage. Throttle temperatures were varied from 500°F (260°C) up to 750°F (399°C), using last stage quality as an indicator of the lower temperature limit and holding heat input constant. Throttle pressures were varied to illustrate the effects on the cycle of changing the admission point up to the normal admission (receiver operation) pressure.

Reference cycle diagrams are presented as Figure 4—1 (receiver operation) and Figure 4—2 (thermal storage operation), with plots of gross and net cycle efficiencies versus throttle temperature and pressure given in Figure 4—3, for receiver and thermal storage operation (assuming 8 percent and 6 percent for total auxiliaries, respectively, including pumping power). Plots of the ratio of non-reference cycle efficiency to reference cycle efficiency versus throttle temperature and pressure, and generator output (gross and net) versus throttle temperature and pressure are shown in Figures 4—4 and 4—5.

REFERENCES

1. McDonnell Douglas Astronautics Company, "Central Receiver Solar Thermal Power System, Phase 1," (Volume 6, EPGS, MDC-G-6776), October 1977.

2. General Electric Company, "A Method for Predicting the Performance of Steam Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

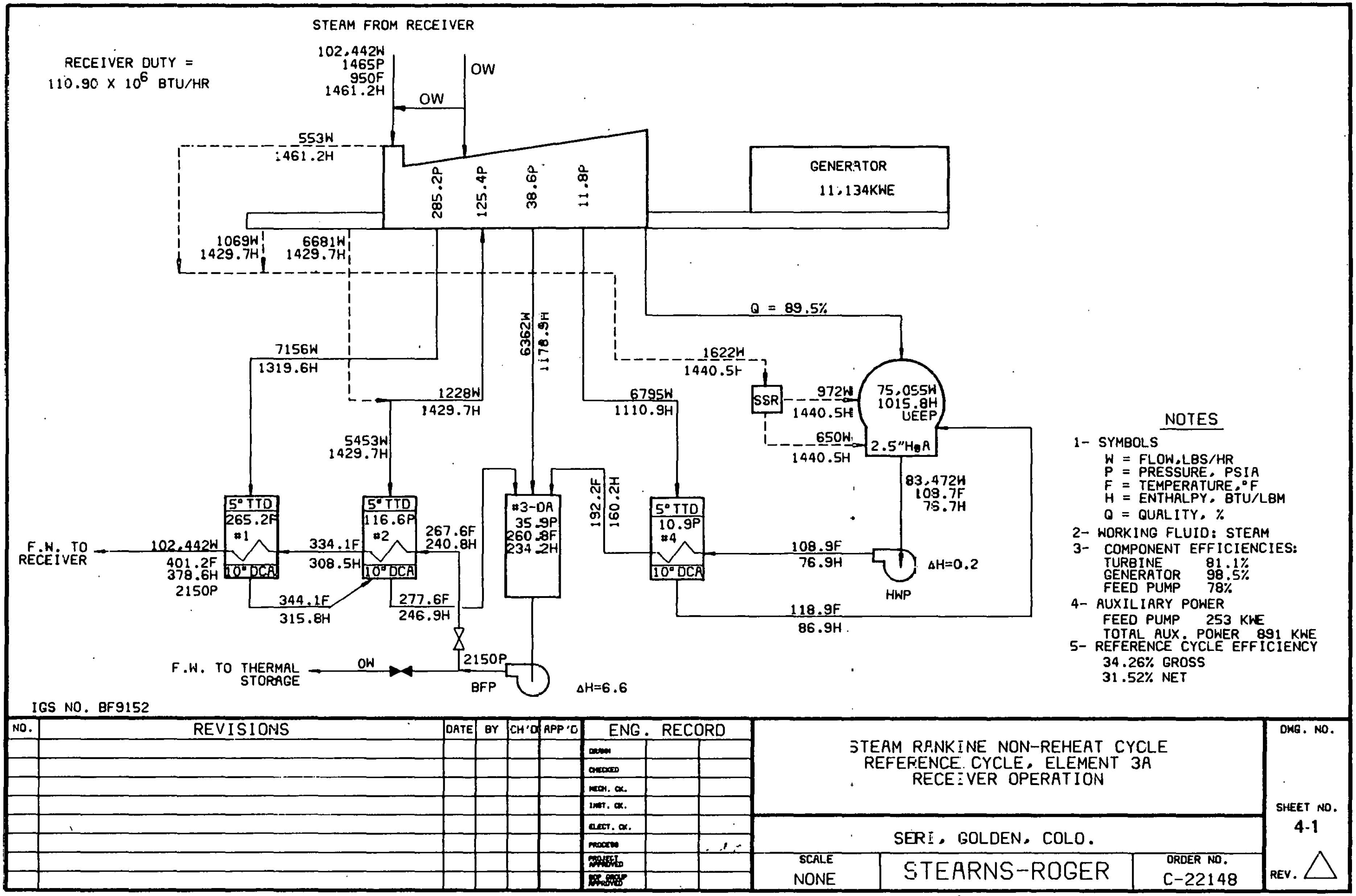


Figure 4–1. ELEMENT 3A - STEAM RANKINE NON-REHEAT CYCLE, RECEIVER OPERATION

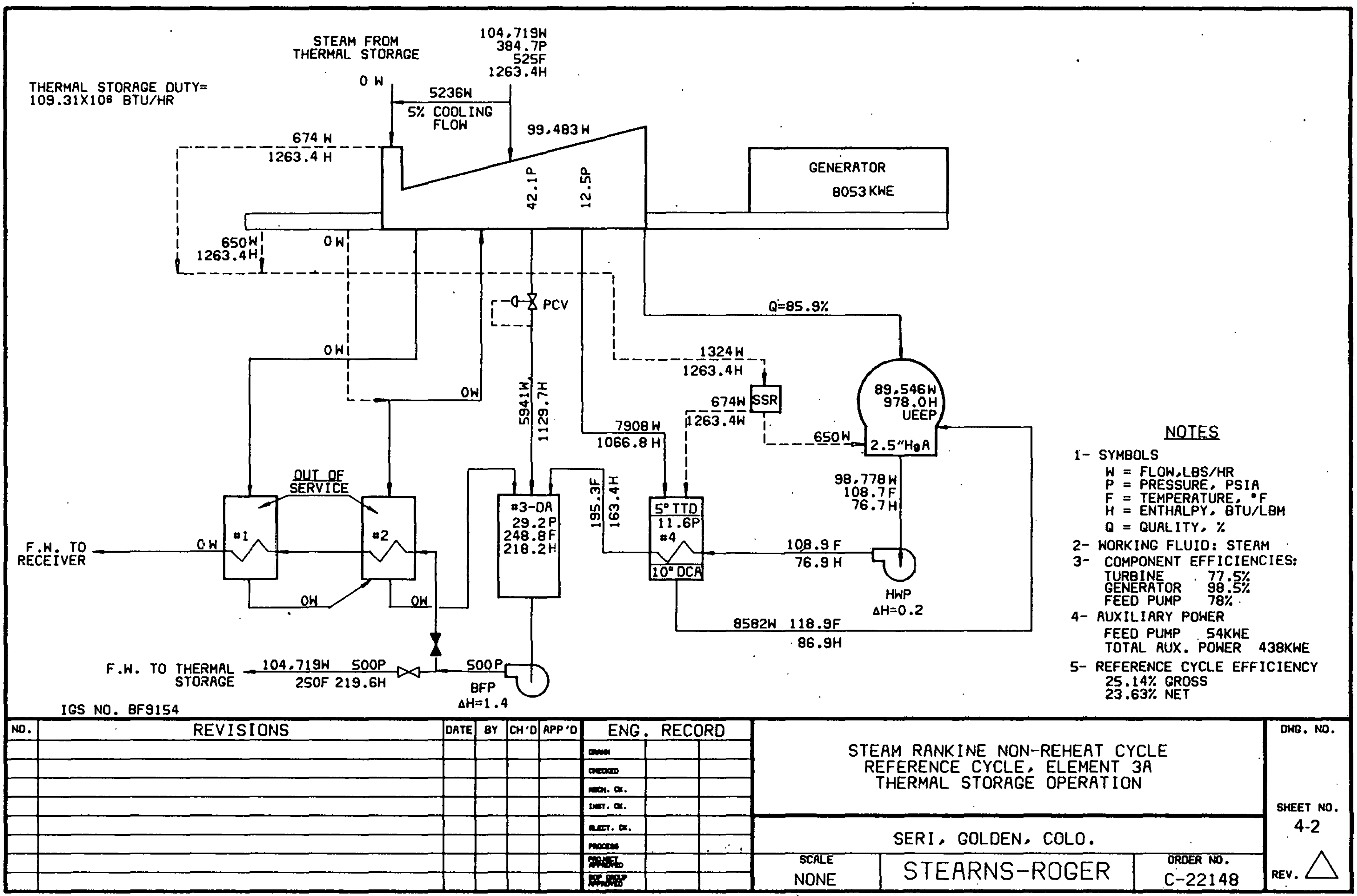


Figure 4–2. ELEMENT 3A - STEAM RANKINE NON-REHEAT CYCLE, THERMAL STORAGE OPERATION

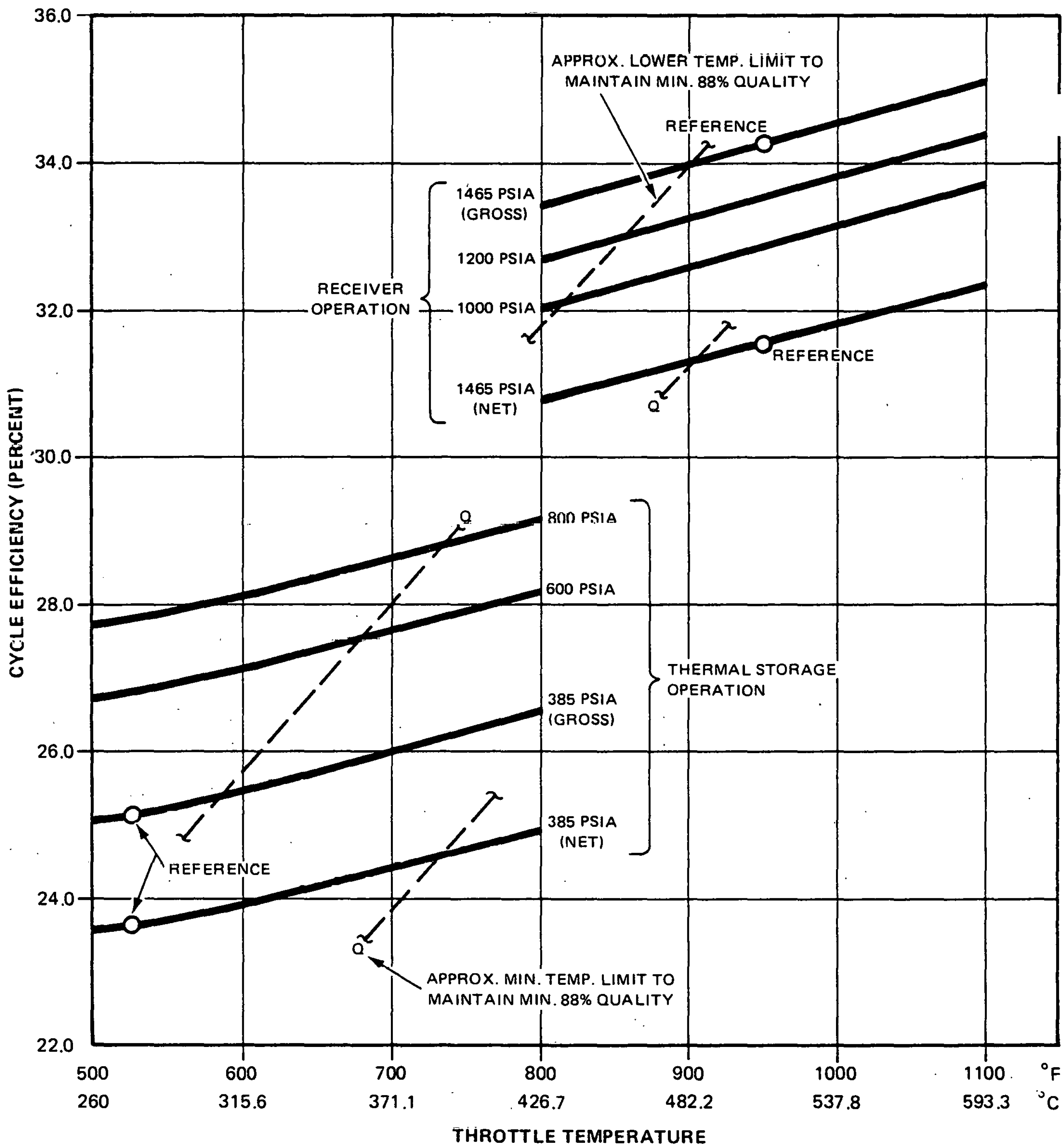


Figure 4–3. ELEMENT 3A - STEAM RANKINE NON-REHEAT CYCLE, CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

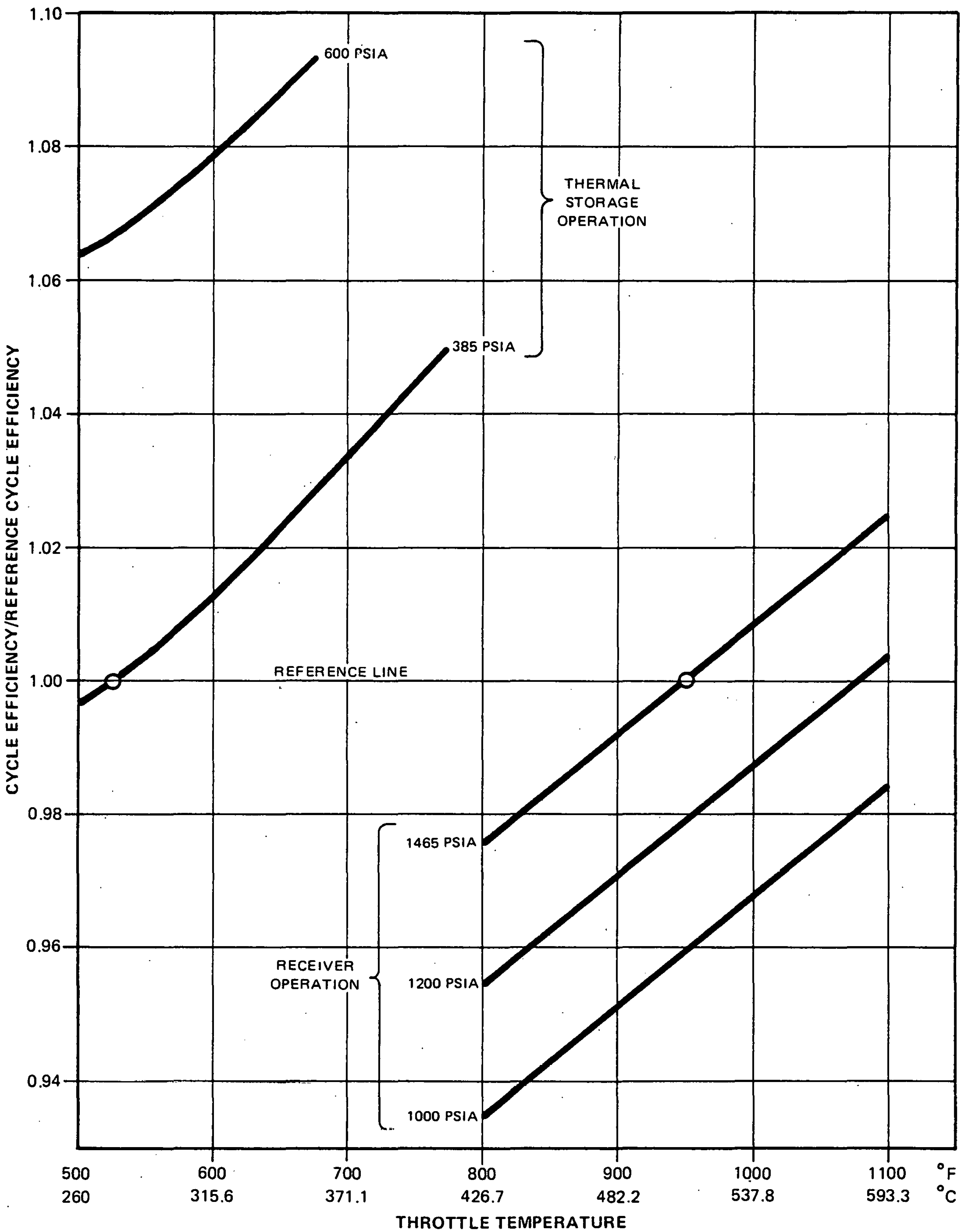


Figure 4—4. ELEMENT 3A - STEAM RANKINE NON-REHEAT CYCLE, CYCLE EFFICIENCY VS. REFERENCE CYCLE EFFICIENCY

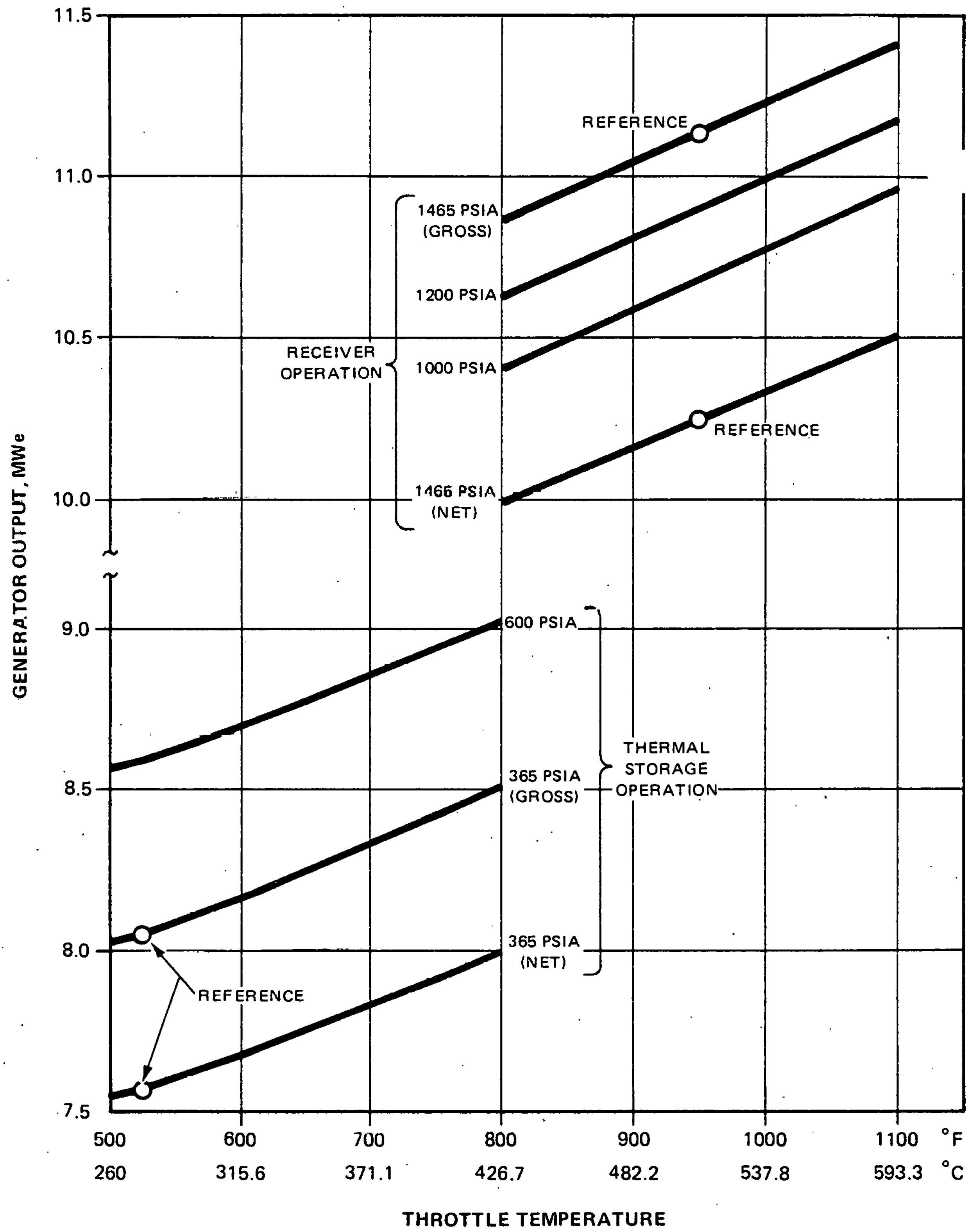


Figure 4–5. ELEMENT 3A - STEAM RANKINE NON-REHEAT CYCLE, GENERATOR OUTPUT VS. THROTTLE TEMPERATURE

SECTION 5
ELEMENT 3B - ORGANIC RANKINE CYCLE

5.1 INTRODUCTION

Performance data were calculated for the reference cycle as defined in the project statement of work for a 150 KWe net generation, organic Rankine Cycle Power System. Toluene (Monsanto Chemical Co. designation CP-25) was used as the working fluid as specified in the statement of work.

5.2 CYCLE DESCRIPTION

The reference cycle is based on a maximum boiler outlet temperature of 350°F (176.7°C) saturated. Saturation pressure is 71.0 psia (489.54 x 10^3 Pa).

The Organic Rankine Cycle is identical, thermodynamically to a steam Rankine cycle used in large utility generating plants, however due to the thermal properties of organic fluids there are certain component differences.

Figure 5—1 shows a schematic of the Organic Rankine Cycle analyzed together with the state points and performance for the 150 KWe (net) reference cycle.

Liquid Toluene is preheated and vaporized in the vaporizer with heat supplied by the solar receiver or thermal storage systems. In most Organic Rankine Systems, the working fluid is admitted to the turbine from the vaporizer with little or no superheat. The reason for this is that for the organic fluids and Toluene specifically, the saturated vapor line when plotted on a T-S diagram has a positive slope. This means that if the fluid leaving the vaporizer and entering the turbine is saturated, vapor expansion through the turbine will result in the turbine exit fluid being considerably superheated, with no danger of wet fluid causing blade erosion as in a conventional steam turbine.

After the working fluid is expanded in the turbine, the vapor is passed through a regenerator which is a vapor/liquid heat exchanger used to remove the superheat from the vapor and transfer this heat to the liquid Toluene at the discharge of the feed pump. The utilization of this superheat to preheat the liquid to the vaporizer improves cycle efficiency since it is not rejected in the condenser.

The condenser used in this analysis is a conventional shell-and-tube, water-cooled heat exchanger. As specified in the work statement, a condensing temperature of 100°F (37.8°C) was used. Two degrees of subcooling was assumed to take place in the condenser to provide adequate net positive suction head at the feed pump inlet.

The feed pump is assumed to be a centrifugal type, and could be powered either by an electric motor drive or be driven directly off the turbine shaft.

Typically, Organic Rankine Cycle Turbines use impluse type blade design and operate at significantly higher speeds than do conventional steam turbines due to the thermodynamic properties of the fluid (Reference 1 and 2). For this reason, Figure 5—1 shows a speed reducer between the turbine and the generator. Details of the turbine/generator system can only be determined after an engineering design of the system has been performed.

Component efficiences which were used in this analysis are shown on Figure 5—1 and were obtained from Reference 3 and are considered typical for this cycle. These efficiencies were held constant for all the cycle temperatures investigated.

Pressure losses through the system were based on the assumption in Reference 3. The most significant pressure loss relating to cycle performance is the hot-side of the regenerator, since this loss affects the available energy in the turbine when condenser temperature is held constant. Regenerator hot-side losses will directly affect regenerator size and the losses assumed for this study are judged to be typical.

The Toluene fluid property data were obtained from Monsanto Chemical Co., Reference 4.

The analysis for the reference cycle consisted of performing a heat balance around the cycle using a 150 KWe net generator output, 350°F (176.7°C) saturated boiler output and 100°F (37.8°C) condensing temperature. A 5 percent auxiliary power requirement was assumed resulting in a gross generation of 157.5 KWe. The cycle analysis showed that the vaporizer feed pump will require 2.25 KWe with 5.25 KWe available for the remainder of the auxiliary power requirements such as circulating water pump power to the condenser and cooling tower fan power.

A vaporizer heat input requirement was calculated for the reference cycle and was held constant for the cycle calculations at the other maximum temperature conditions. Generator output was allowed to vary for each of the nonreference cycles.

A net and gross cycle efficiency was calculated for a number of fluid temperatures. Gross cycle efficiency is defined as the ratio of the gross generator output divided by the total heat into the cycle. Net efficiency includes the auxiliary power.

The data for the reference cycle are shown on Figure 5—1. The efficiency data for each of the non-reference cycles are shown on Figures 5—2 and 5—3. Cycle efficiency is plotted as a function of maximum cycle temperature on Figure 5—2. Figure 5—3 shows the ratio of cycle efficiency to reference cycle efficiency plotted as a function of maximum cycle temperature. Data were calculated for cycles both below and above the critical point of Toluene. All of the supercritical cycles were calculated at a pressure of 800 psi (5516 x 10^3 Pa). The discontinuity that exists between the subcritical and supercritical cycles is due to the pressure change and the fluid data inconsistancy at the critical point. Auxiliary power requirements of 7 to 9 percent were used for the supercritical cycles. The reason for the higher auxiliary power for these cycles is the increase in feed pump power for the supercritical pressures.

Figure 5—4 shows a plot of gross and net generator output as a function of maximum cycle temperature and based on a constant cycle heat input as determined from the reference cycle.

REFERENCES

1. Bjerklie, J. W. "Working Fluid as a Design Variable for a Family of Small Rankine Power Systems", *ASME* 67-GT-6.

2. Luchter, S. "A Quantitative Method of Screening Working Fluids for Rankine Cycle Power Plants", *ASME* 67-GT-12.

3. Barber, R. E. "Current Costs of Solar Powered Organic Rankine Cycle Engines", *Solar Energy*, Volume 20, pp 1-6, 1978.

4. Monsanto, "Toluene Thermodynamic Grid", Data Ref. 73294, St. Louis 10/12/73.

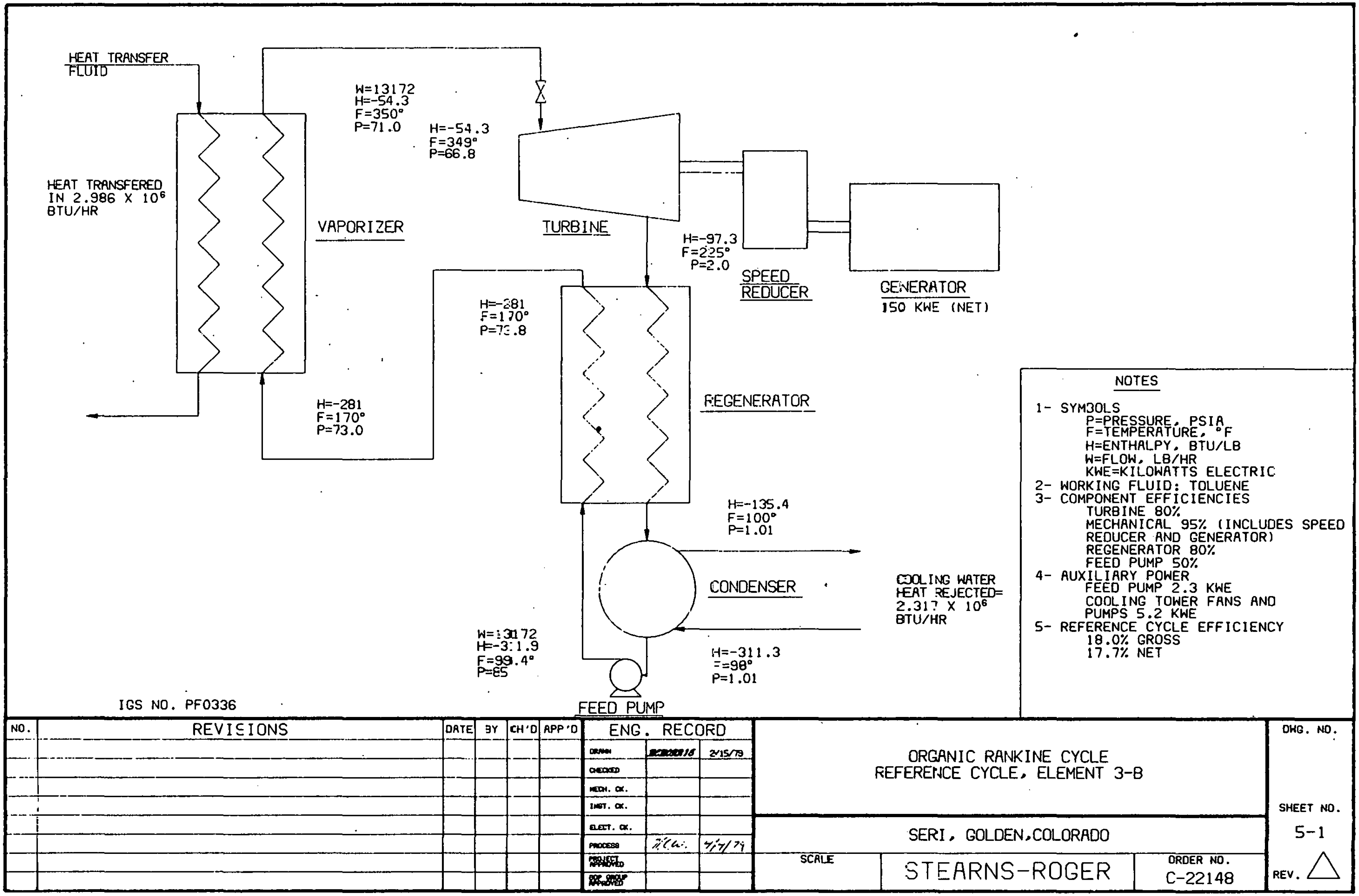


Figure 5–1. ELEMENT 3B - ORGANIC RANKINE CYCLE

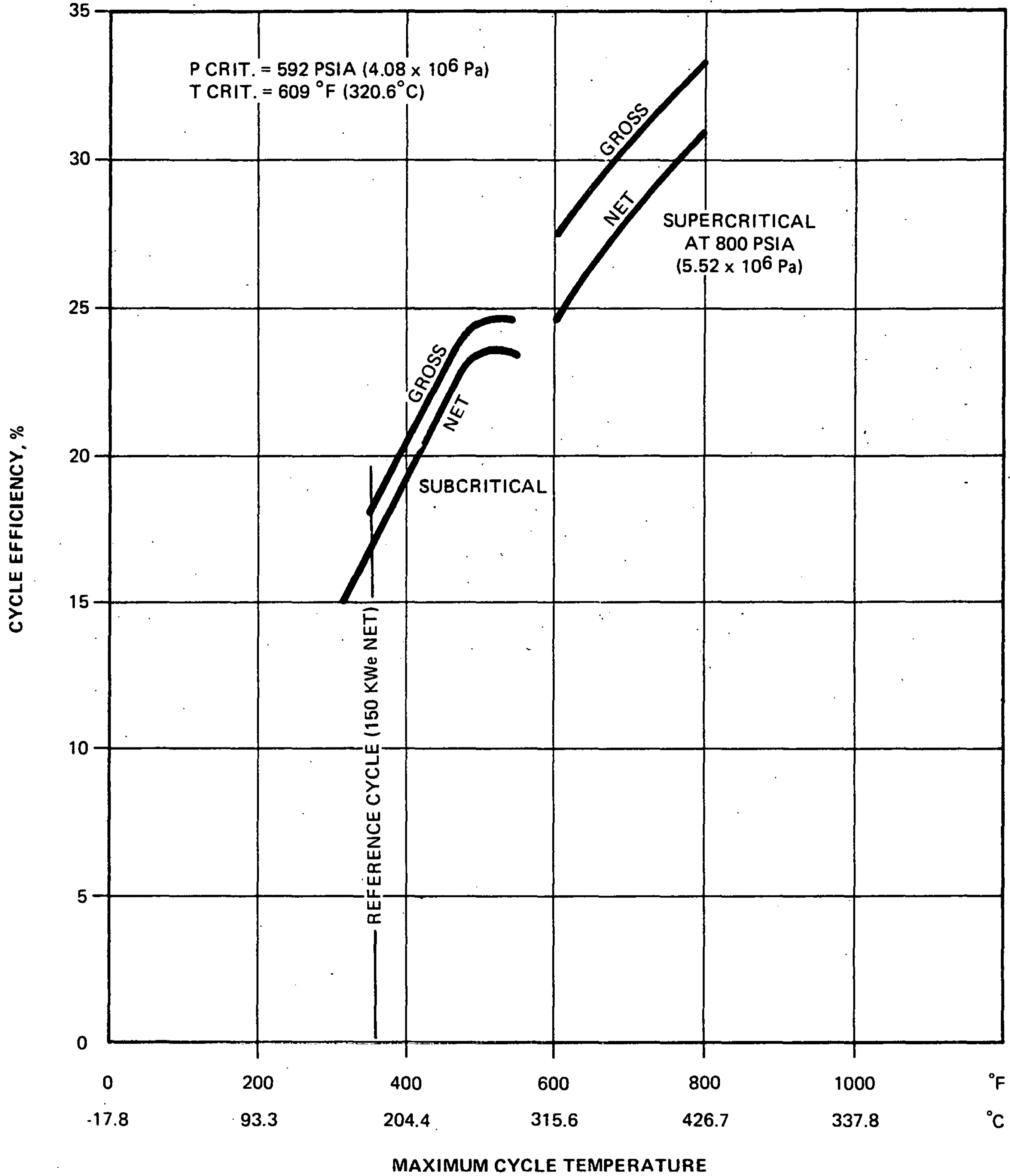


Figure 5–2. ELEMENT 3B - ORGANIC RANKINE CYCLE,
CYCLE EFFICIENCY VS. MAXIMUM CYCLE TEMPERATURE

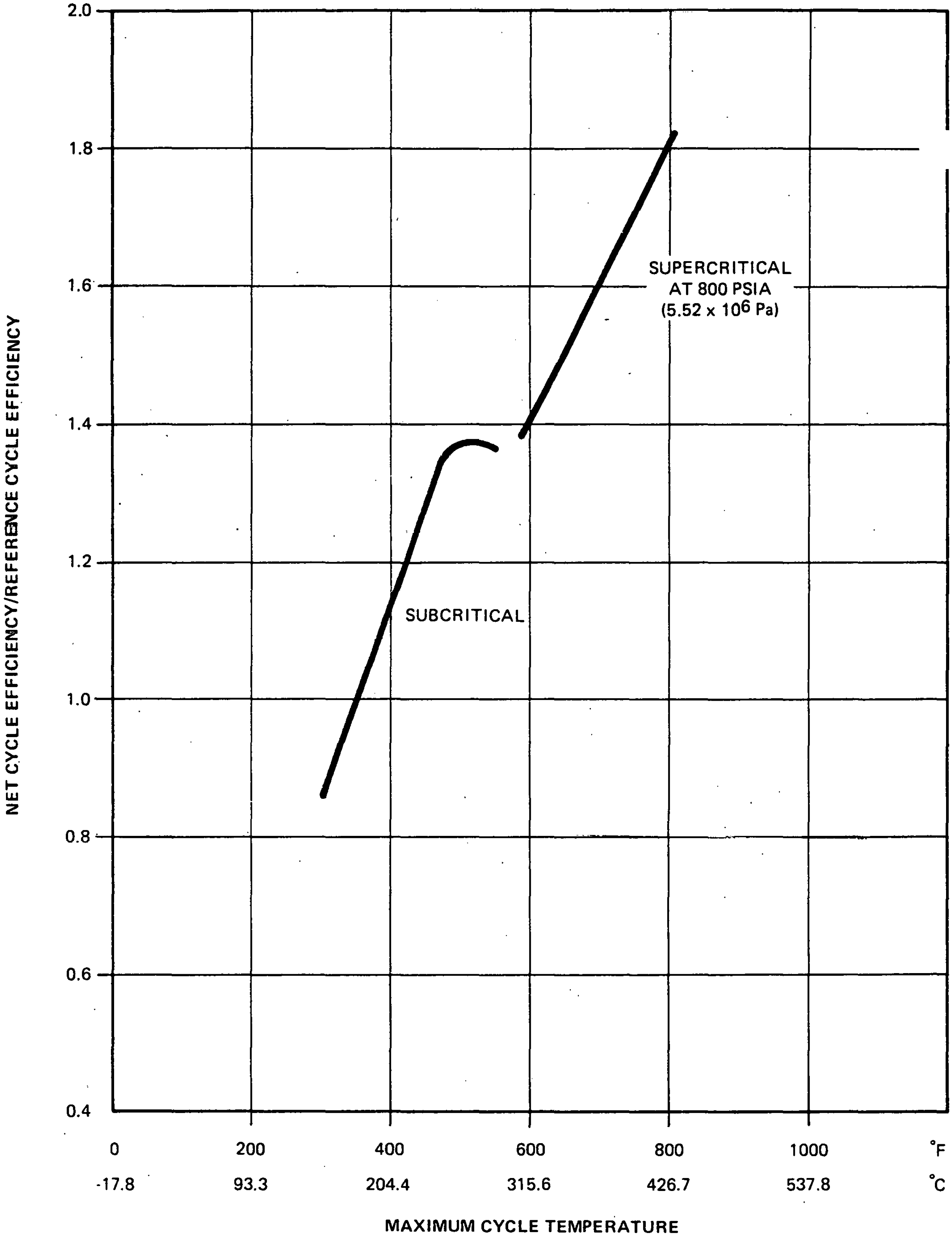


Figure 5–3. ELEMENT 3B - ORGANIC RANKINE CYCLE,
NET CYCLE EFFICIENCY VS. REFERENCE CYCLE EFFICIENCY

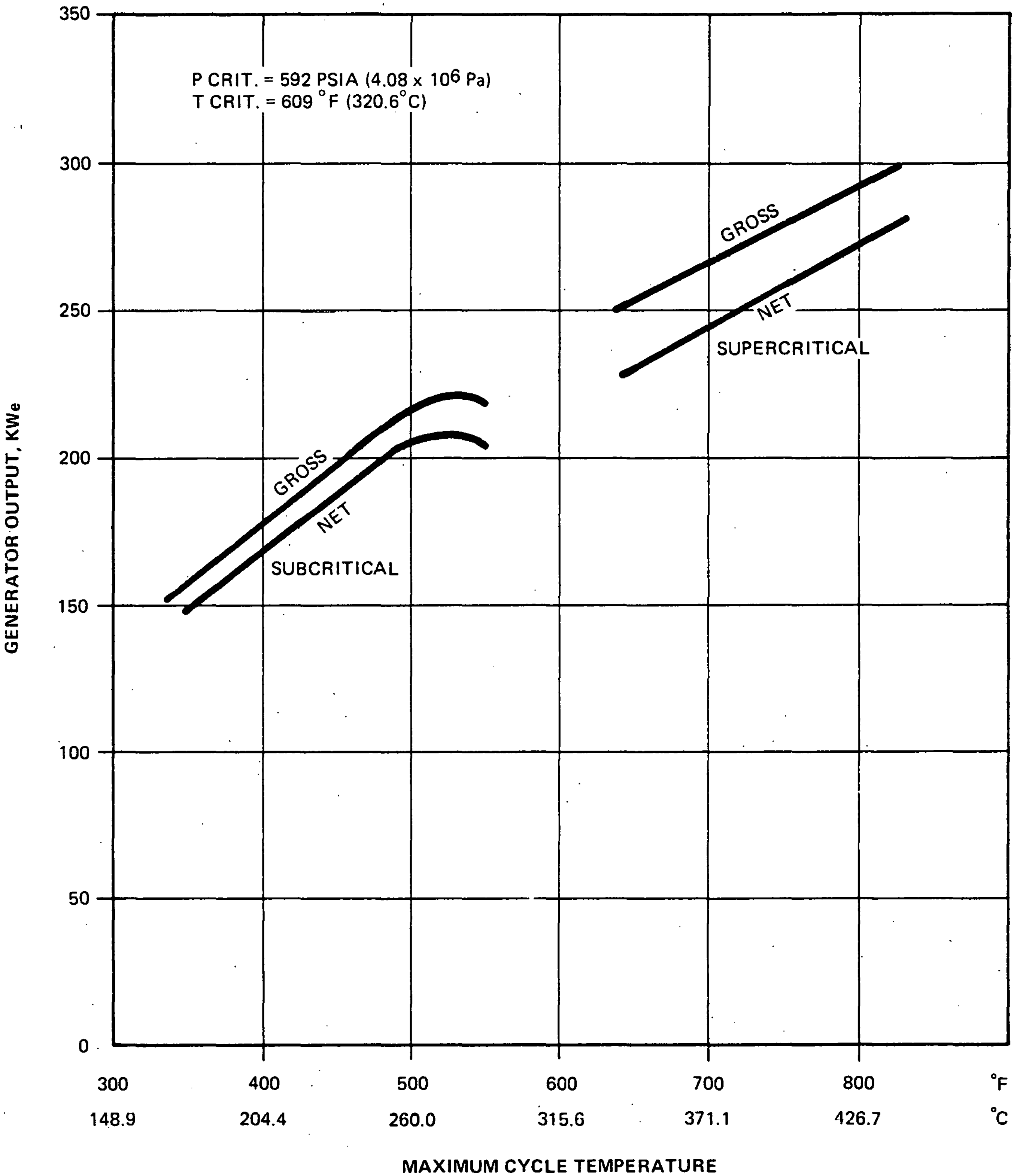


Figure 5–4. ELEMENT 3B - ORGANIC RANKINE CYCLE, GENERATOR OUTPUT VS. MAXIMUM CYCLE TEMPERATURE

SECTION 6
ELEMENT 3C - STEAM RANKINE CYCLE, TOTAL ENERGY SYSTEM

6.1 INTRODUCTION

Cycle heat balances were performed for a 400 KWe (nominal) steam Rankine cycle used in a total energy system. This cycle uses a single automatic extraction condensing turbine with one stage of regenerative feedwater heating. The extraction point on the turbine provides steam at 125 psia (8.62 x 10^5 Pa) which supplies the deaerator requirement and is desuperheated to 340°F (171.1°C) to supply the process requirement. The condensate is returned to the condenser at a temperature of 230°F (110°C). The turbine exhausts to a condenser at 20.78 psia (1.43 x 10^5 Pa). The cycle is largely based on the General Electric Solar Total Energy Cycle as referenced in the statement of work.

The cycle was analyzed using a computer program to perform the required mass and energy balances around the cycle. The program assumes that the expansion of the steam through the turbine is a straight line on a Mollier diagram. The program accesses subroutines to calculate the fluid state conditions around the cycle using the relationships specified in the ASME steam tables.

6.2 CYCLE DESCRIPTION

Throttle conditions used for the reference cycle are 715 psia (4.93 x 10^6 Pa) and 720°F (382.2°C). A pressure loss of 10 percent is assumed across the steam generator.

For this study, the total heat input to the cycle and the process steam usage was held constant while the throttle conditions were varied. The throttle temperature was varied over the range of 650°F to 1100°F (343.3°C to 593.3°C), and the throttle pressure was varied over the range of 715 psia to 1450 psia (4.93 x 10^6 Pa to 10.0 x 10^6 Pa). The throttle pressures selected are those normally used in the power industry. The upper temperature limit was selected as the limit of existing steam turbine technology. The lower temperature limit varied with throttle pressure such that the required process steam temperature could be achieved at the extraction point. For the 715 psia (4.93 x 10^6 Pa) pressure, the minimum temperature is 650°F (343.3°C); for the 850 psia (5.86 x 10^4 Pa) pressure, the minimum temperature is 700°F (371.1°C); and for the 1250 psia and 1450 psia (8.62 x 10^6 Pa and 10.0 x 10^6 Pa) pressures, the minimum temperature is 800°F (426.7°C). The turbine efficiency was calculated from the throttle and exhaust conditions of the reference cycle, the General Electric Solar Total Energy System. The turbine efficiency was held constant over the range of throttle conditions in order to prevent distortion of the effect of throttle conditions on the overall cycle efficiency. In reality, the turbine efficiency will increase slightly as the amount of superheat of the throttle steam increases. Generator output was allowed to vary with the throttle conditions.

A cycle schematic showing component efficiencies and flow data is shown in Figure 6—1 for the base cycle. Plots of cycle efficiency versus throttle temperature are shown in Figure 6—2 and Figure 6—3. A normalized representation of cycle efficiency with respect to base cycle efficiency versus throttle temperature is shown in Figure 6—4. A plot of generator output versus throttle temperature is shown in Figure 6—5. Gross cycle efficiency is defined as the total cycle input energy divided into the generator output. Net cycle efficiency is defined as the total cycle input energy divided into the generator output less the plant auxiliary power requirement (which is assumed to be 8 percent of the generator output). Plant auxiliary power includes that power used by the boiler feed pump, condensate pump, circulating water pump, cooling tower fans, controls, plant lighting, plant HVAC, solar collector field usage, etc.

Figure 6—6 is a plot of gross cycle efficiency versus process flow for the base case throttle conditions. The efficiency of the cycle decreases as process flow increases because the boiler duty is held constant. As more steam is extracted to the process, less steam is available to produce electric power.

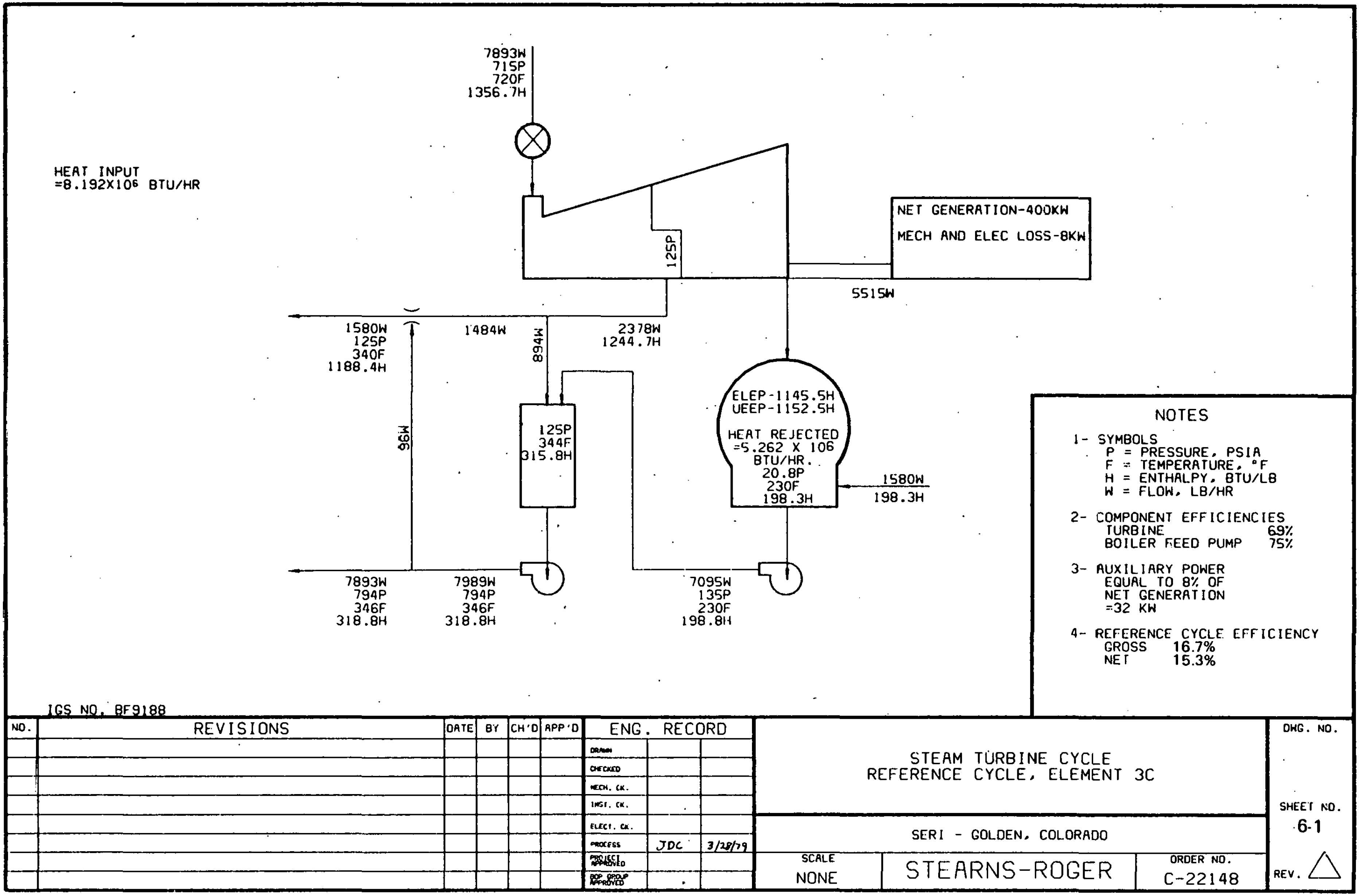


Figure 6–1. ELEMENT 3C - STEAM RANKINE CYCLE

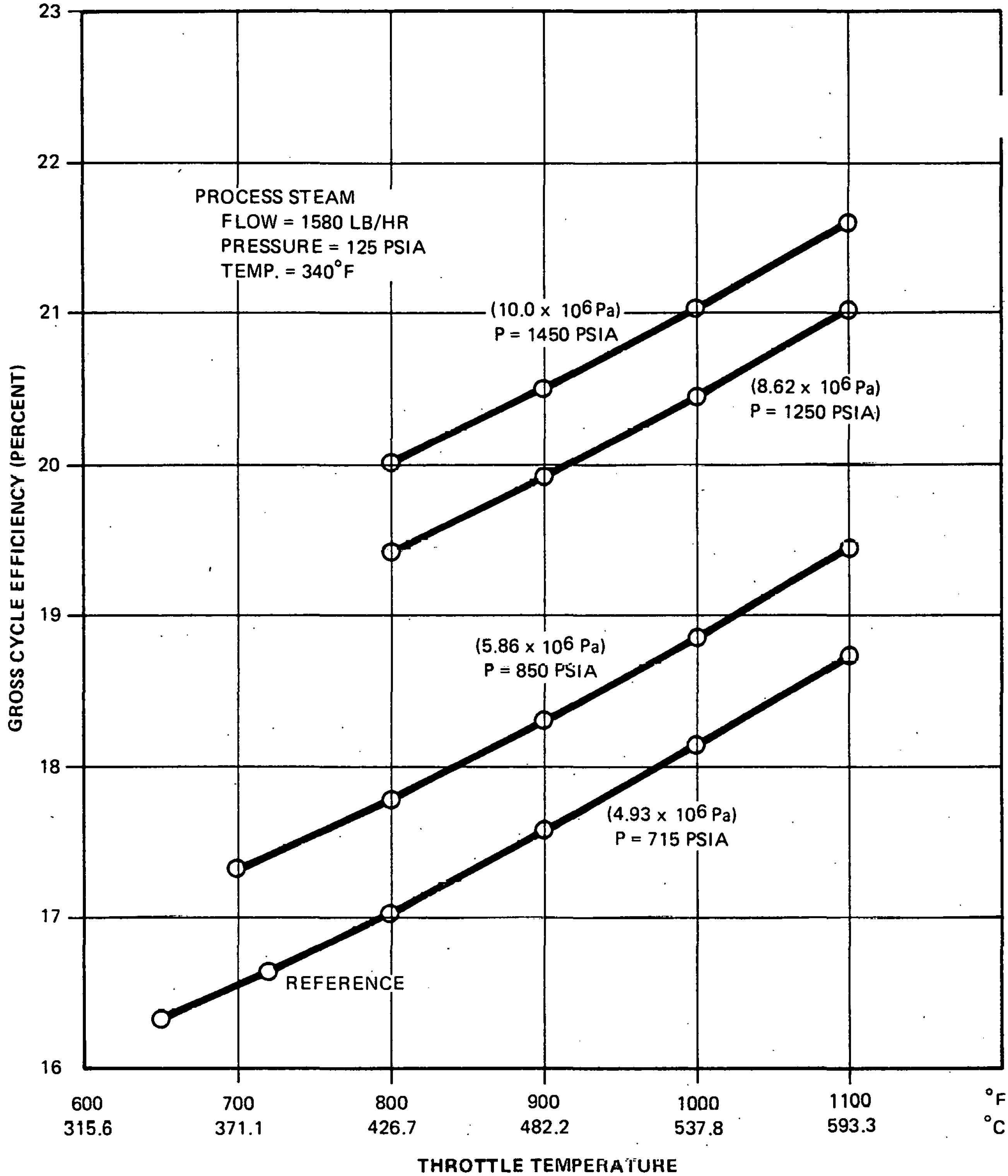


Figure 6–2. ELEMENT 3C - STEAM RANKINE CYCLE, GROSS CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

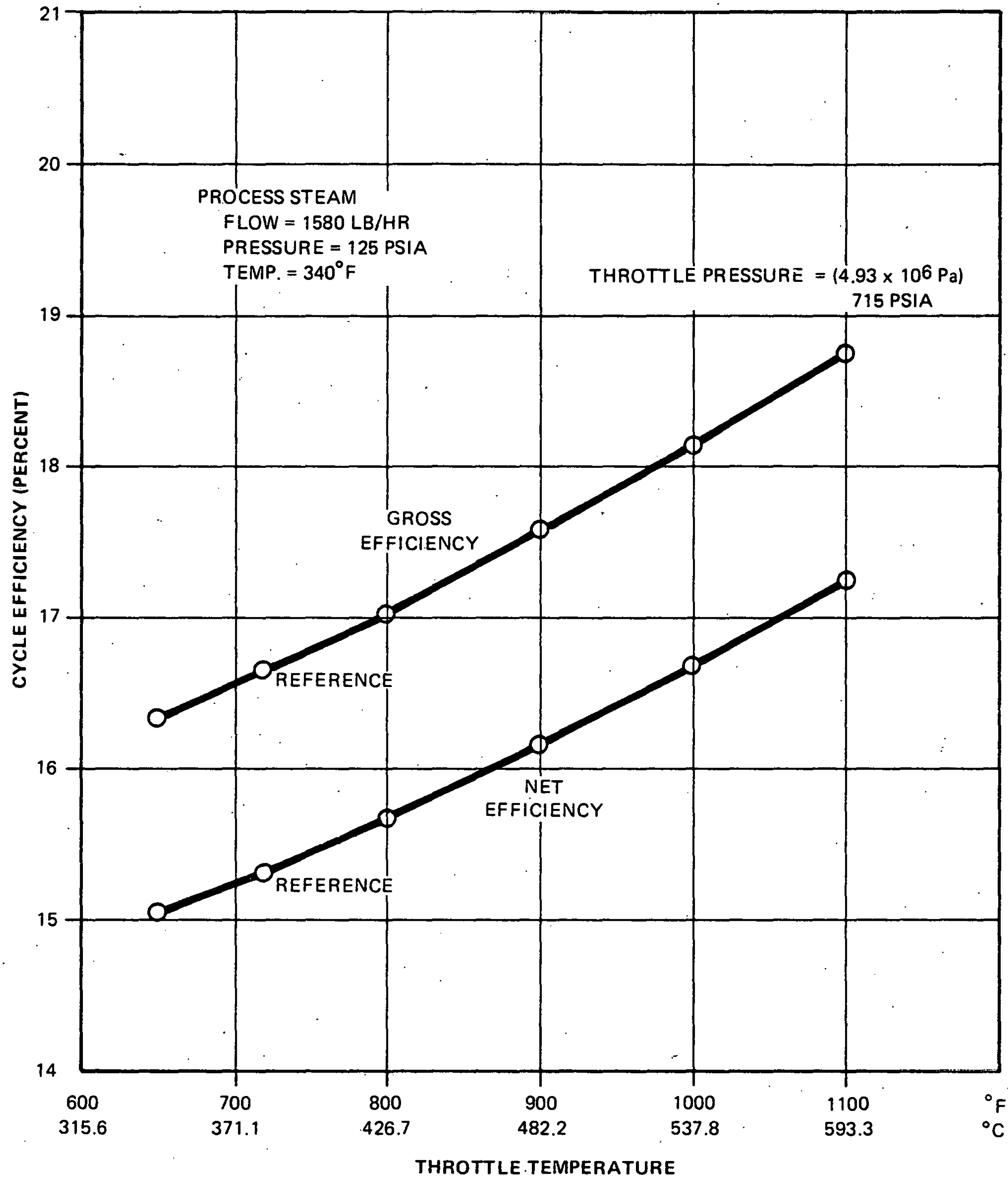


Figure 6–3. ELEMENT 3C - STEAM RANKINE CYCLE, EFFICIENCY VS. THROTTLE TEMPERATURE

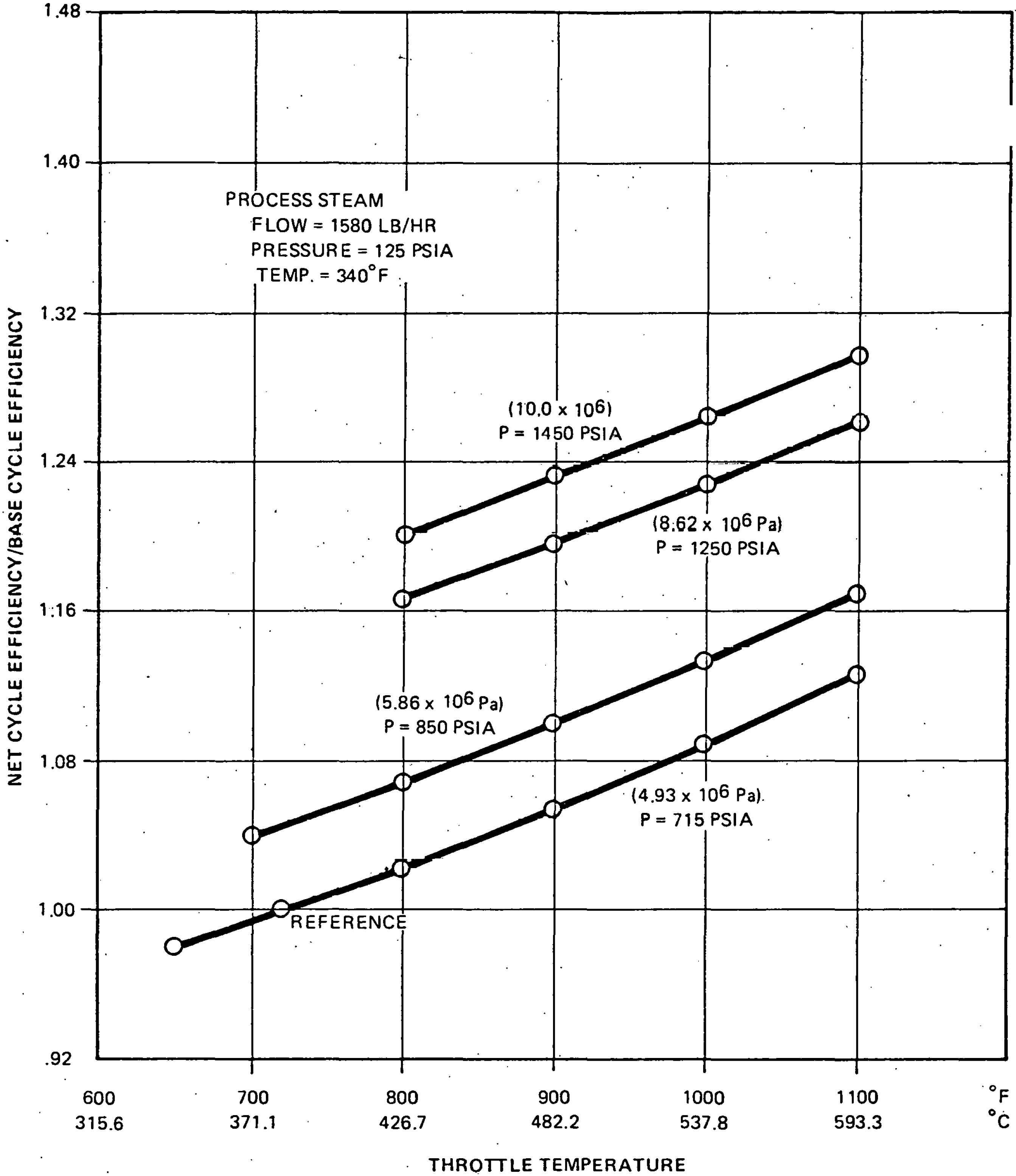


Figure 6–4. ELEMENT 3C - STEAM RANKINE CYCLE,
RELEATIVE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

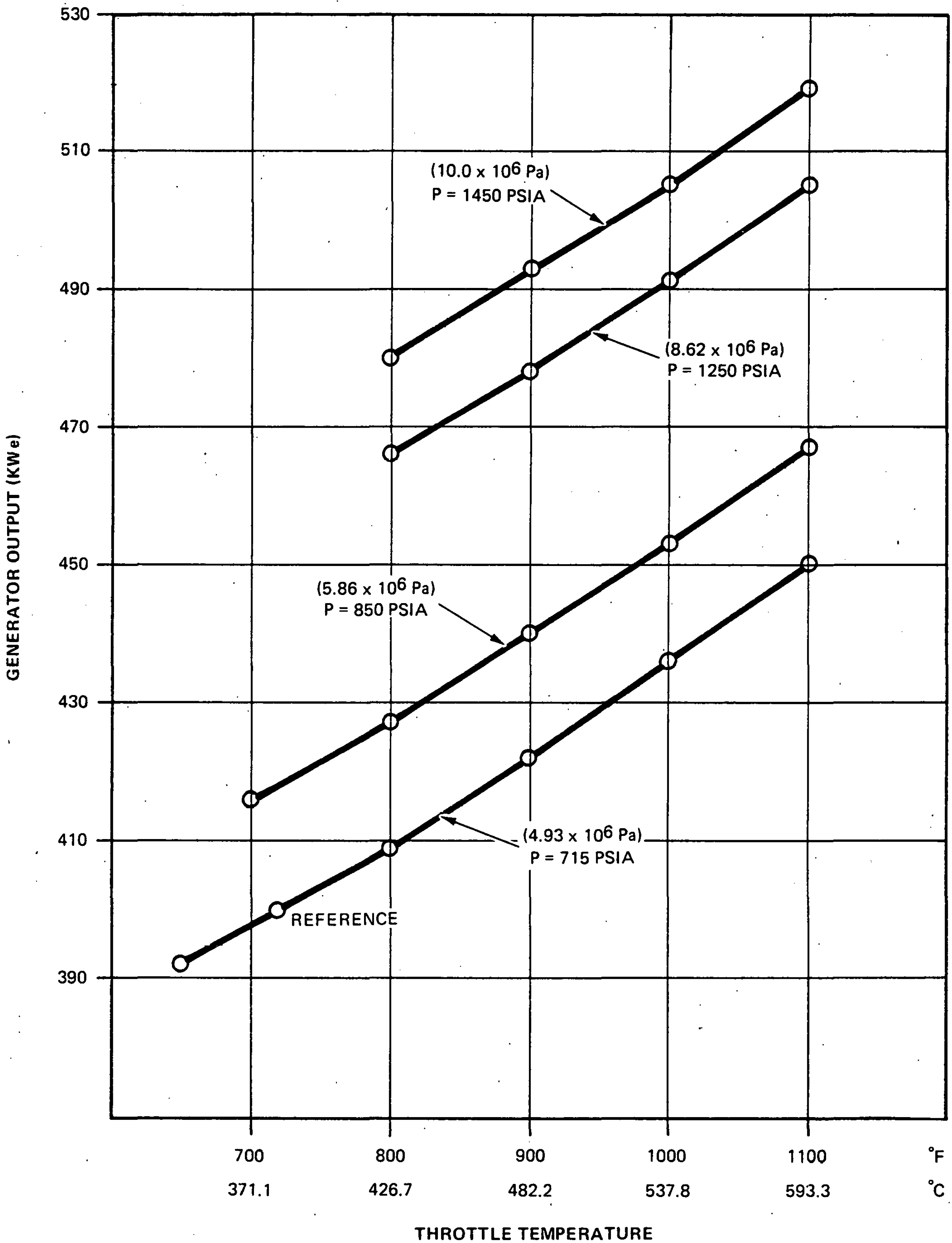


Figure 6–5. ELEMENT 3C - STEAM RANKINE CYCLE, GENERATOR OUT PUT VS. THROTTLE TEMPERATURE

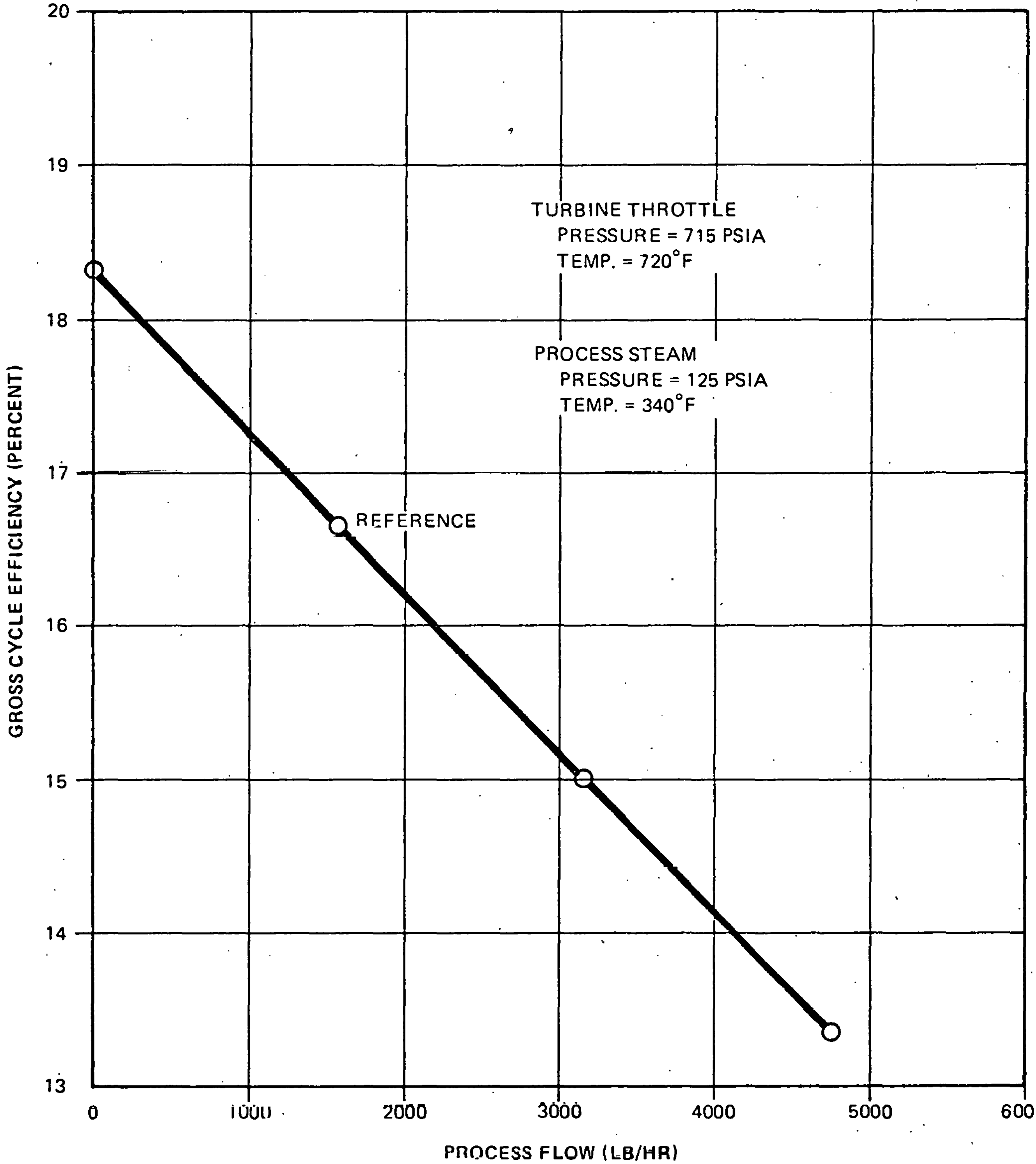


Figure 6–6. ELEMENT 3C - STEAM RANKINE CYCLE, CYCLE EFFICIENCY VS. PROCESS FLOW

SECTION 7
ELEMENT 4A - REPOWER - STEAM RANKINE NON-REHEAT CYCLE

7.1 INTRODUCTION

Cycle heat balances were performed for a 100 MWe (nominal) steam Rankine, non-reheat cycle incorporating a six heater feedwater heating system. The cycle was duplicated from that of the Public Service of New Mexico Repowering Study, Reference 1, using steam from a solar receiver to repower an existing fossil-fuel-fired unit. A seventh condensing feedwater heater is added to the top of the cycle to recover heat from the discharge of the thermal storage system. It is assumed that the amount of heat absorbed in the thermal storage unit reduces the steam conditions from superheated (throttle temperature and pressure) to saturated conditions, after a pressure drop of 10 percent through the thermal storage unit. For this study the fossil-fuel-fired boiler was out of service.

7.2 CYCLE DESCRIPTION

The turbine used in this model is a standard-frame General Electric utility, non-reheat steam turbine exhausting at an assumed 2.5 inches HgA (0.0984 mm HgA) to a tube-and-shell condenser. The turbine's six extractions are connected to two closed high-pressure heaters, an open deaerating heater, and three closed low-pressure heaters operating at various pressures. The heater operating characteristics are as shown on the cycle diagram Figure 7—1.

Throttle conditions are 1250 psia (8.62 x 10^6 Pa) and 950°F (510°C) for the reference cycle. At these throttle conditions (no steam to thermal storage, and constant final feedwater conditions of 2400 psia (16.55 x 10^6 Pa) and 425°F (218.3°C)), the heat input to the reference cycle was determined. This heat input was held constant for the varying throttle temperatures and pressures, and thermal storage duties studied. The amount of steam to thermal storage was varied from zero up to the point at which the feedwater temperature leaving the thermal storage heater equals the temperature of the heater shell (i.e., saturation temperature at heater shell pressure). This yields a hot end terminal temperature difference (TTD) of 0°, for which it is assumed the thermal storage heater is designed. The throttle temperatures were varied from the reference cycle conditions down to a temperature of 800°F (427°C) which yields approximately 84 percent quality steam leaving the last stage of the turbine, and up to a temperature of 1100°F (593°C) (the upper limit for existing steam turbine technology). It is expected that the turbine will actually operate at a minimum quality of approximately 88 percent steam leaving the last stage. Generator output was allowed to vary with the throttle conditions.

A cycle schematic diagram showing all component efficiencies is presented in Figure 7—1. A parametric plot of cycle efficiency versus throttle temperature at different throttle pressures and thermal storage duties is presented in Figure 7—2. Component efficiencies were determined from Reference 2 (turbine) and from existing conventional power plant operating data (pumps). It must be kept in mind that each pressure, temperature and extraction as a percent of boiler duty shown on Figure 7—2 represents a discrete storage system and heater design. Off design conditions were not considered. The cycle efficiencies are presented as gross (total energy output divided by total energy input to the cycle) and net (assuming 8 percent of total energy output including pumping power goes to auxiliary demand). The ratio of non-reference cycle efficiency to reference cycle efficiency, and generator output (gross and net) are also presented in this Figure. The maximum percent of receiver duty to thermal storage versus throttle temperature is plotted in Figure 7—3 to illustrate the limits of heat to the thermal storage heater before the feedwater temperature reaches a maximum, based on the assumptions above.

REFERENCES

1. Maddox, J. D., Public Service Company of New Mexico, "A Technical and Economic Assessment of Solar Hybrid Repowering," SAMD 78-8511, November 1978, p. 65.

2. General Electric Company, "A Method for Predicting the Performance of Steam-Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

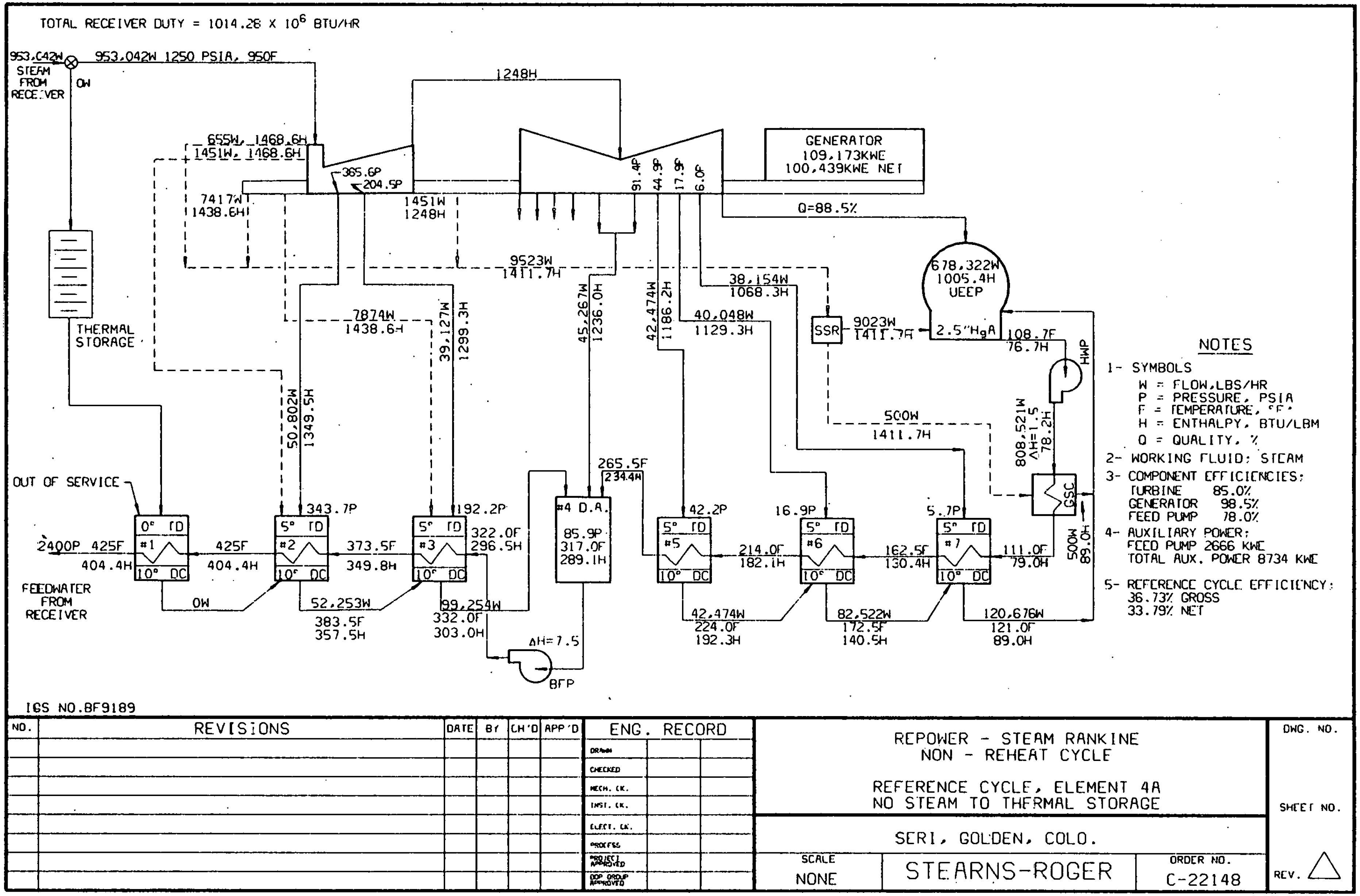
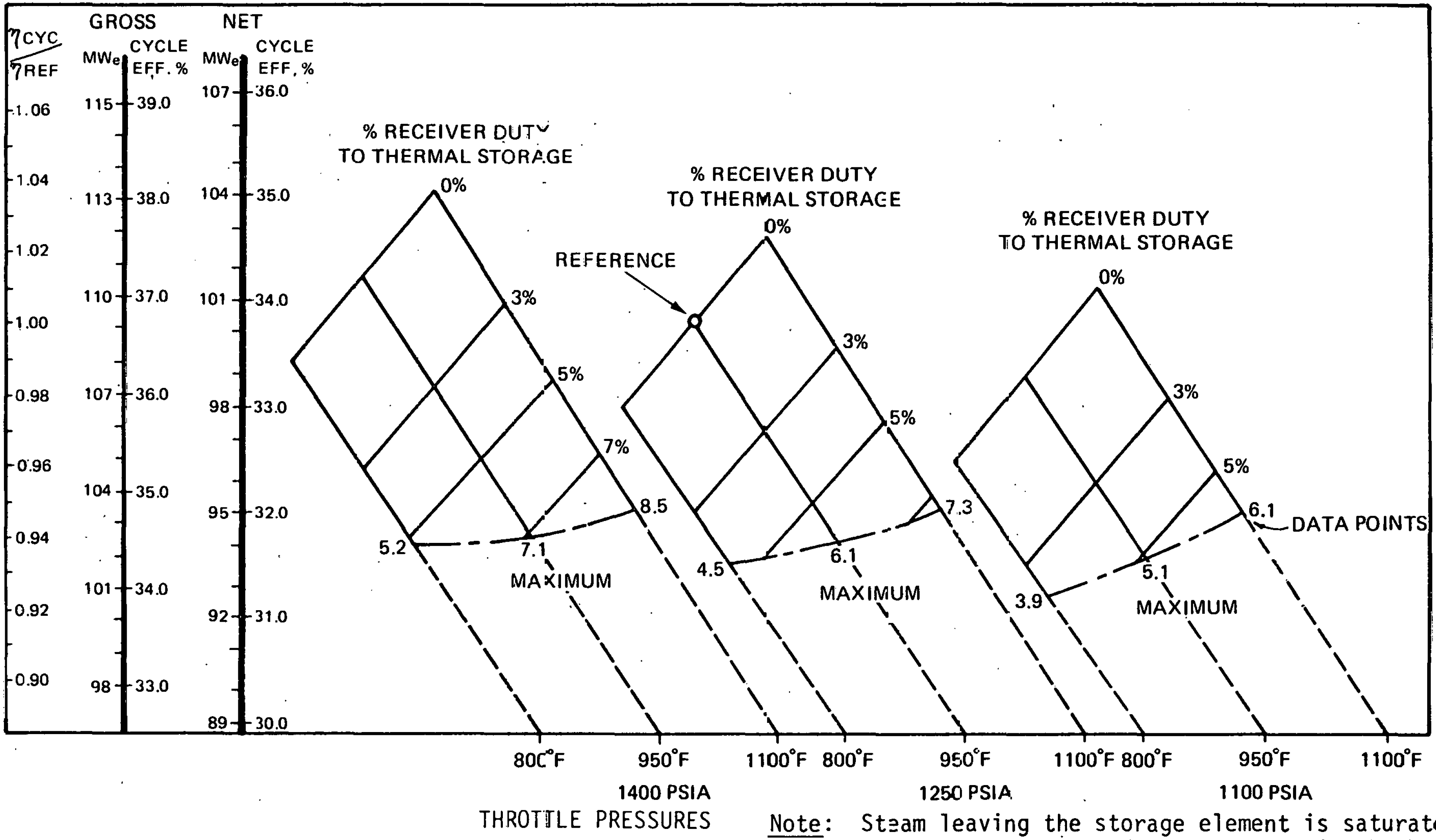


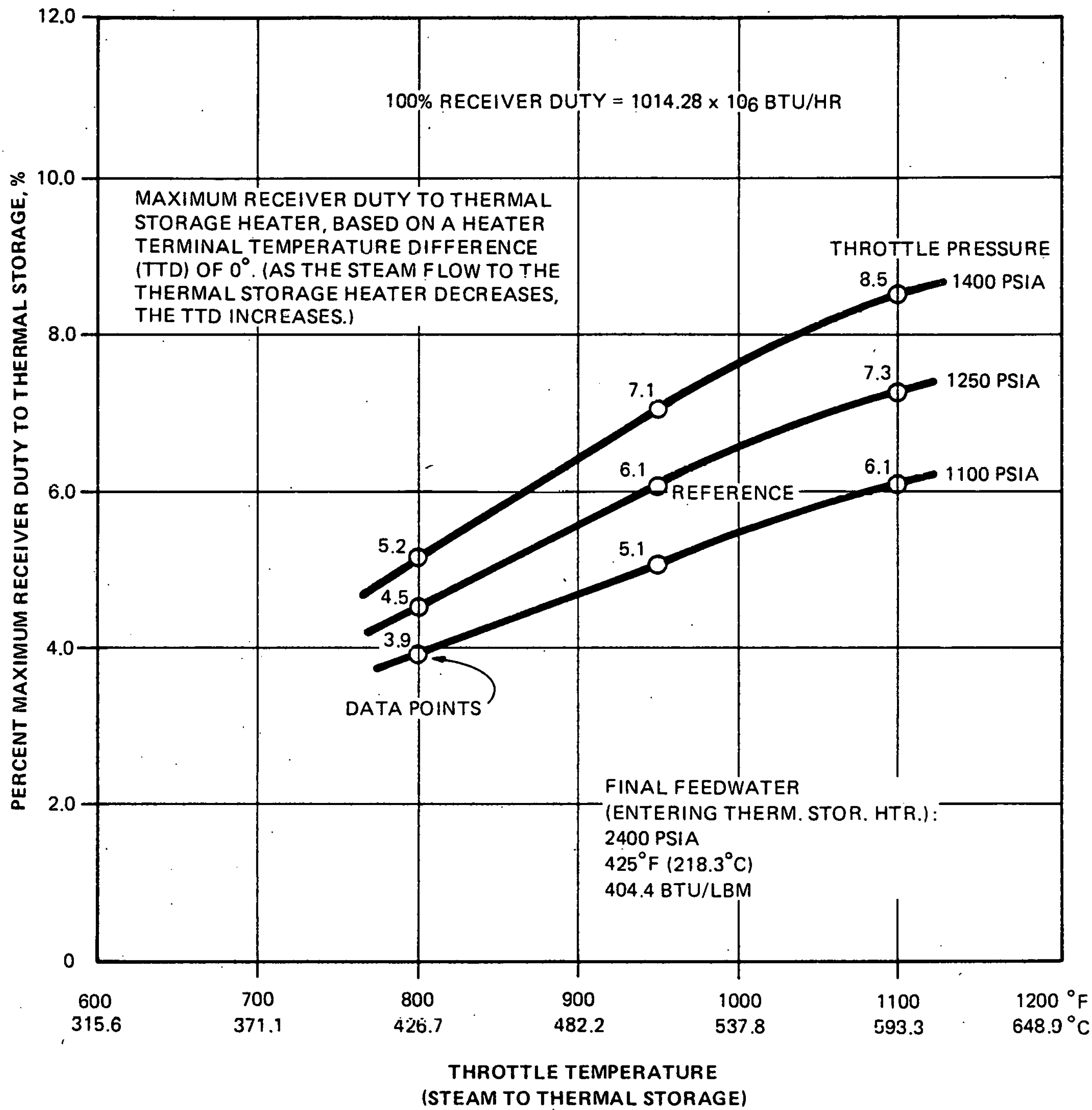
Figure 7–1. ELEMENT 4A - REPOWER - STEAM RANKINE NON-REHEAT CYCLE



REFERENCE CYCLE: 1250 PSIA; 950°F; 0% RCVR. DUTY TO THERM. STOR.
CONSTANT RECEIVER DUTY: 1014.28 MILLION BTU/HR

Note: Steam leaving the storage element is saturated vapor. The heat transferred into storage is the heat contained in the vapor superheat. The % receiver heat duty to thermal storage is therefore ~1/3 of the % flow. At 7% receiver heat duty to storage, ~21% of the steam flow is diverted to storage.

Figure 7–2. ELEMENT 4A - REPOWER - STEAM RANKINE NON-REHEAT CYCLE; CYCLE EFFICIENCY, GENERATOR OUTPUT, CYCLE EFFICIENCY/REFERENCE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE AT VARYING THERMAL STORAGE DUTIES AND CONSTANT RECEIVER DUTY



Note: The terminal temperature difference is the ΔT between the feedwater leaving the last feedwater heater and the steam entering the feedwater heater from the storage element.

Figure 7–3. ELEMENT 4A - REPOWER - STEAM RANKINE NON-REHEAT CYCLE, PERCENT RECEIVER DUTY TO THERMAL STORAGE VS. THROTTLE TEMPERATURE

SECTION 8
ELEMENT 4C - CLOSED ADVANCED BRAYTON AIR REGENERATIVE CYCLE

8.1 INTRODUCTION

Cycle heat balances were performed for a 100 MWe (nominal) closed advanced Brayton air regenerative cycle. Heat input to the cycle is from a solar receiver/thermal storage system. References 1, 2 and 3 were used to set up the thermodynamic relationships to be used in the computer model for performance calculations and to determine component efficiencies.

8.2 CYCLE DESCRIPTION

The reference cycle for this element based on Reference 4, is composed of a two-stage compressor with efficiencies of 80 percent each and a total combined pressure ratio of 4.75. Inlet air is compressed in the first stage, cooled in a water-cooled intercooler, compressed further in the second stage, and discharged to a recuperator. Cooling air for the turbine is diverted from the compressor discharge to various parts of the turbine. Cooling air which is injected into the turbine in the flow path does useful work, while cooling air going to the turbine casing does no work. From the compressor, the air flows to the recuperator where it is heated by the turbine exhaust. The air temperature is further increased by the solar receiver/thermal storage system to a temperature of 1500°F (815.6°C) for the reference cycle. The hot air is expanded in a 90 percent efficient turbine to produce shaft work which drives the compressor and the generator. The turbine exhaust passes through the recuperator where it releases heat to the compressor discharge air. From the recuperator, the turbine exhaust is further cooled by a water-cooled precooler to a constant 100°F (37.8°C) prior to reentering the compressor.

A general industry guide was used to determine how much of the total system flow was used as cooling air which did no work in the turbine. This guide is that cooling air is required above a turbine inlet temperature of 1700°F (926.7°C), and will be one percent of the compressor flow for each 100°F (55.6°C) above 1700°F. The cooling air flow considered in this study represents that amount of flow that does no work in the turbine.

For this study, the generator output was held constant at 100 MWe gross, while the pressure ratio of the compressor was varied over the range of 2 to 9 at various turbine inlet temperatures ranging from 1500°F (815.6°C) to 2400°F (1315.6°C). A cycle schematic showing component efficiencies, pressure drops, heat exchanger effectiveness, and cycle flow data is shown in Figure 8-1 for the reference cycle. A plot of cycle efficiency (gross and net) versus compressor pressure ratio at several turbine inlet temperatures is shown in Figure 8-2. A normalized representation of cycle efficiency with respect to reference cycle efficiency versus compressor pressure ratio is also shown in Figure 8-2. The effects of the cooling air flow on cycle efficiency are demonstrated by the temperature lines of Figure 8-2 crossing each other at lower pressure ratios. Note that there is no cooling flow for turbine inlet temperatures below 1700°F (926.7°C).

Gross cycle efficiency is defined as the gross generator output divided by the total cycle input energy. Net cycle efficiency is defined as the net generator output divided by the total cycle input energy, where the total plant auxiliary requirements are assumed to be 8 percent of the gross generator output.

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3. Telephone conversations with various gas turbine manufacturers relative to current compressor and turbine efficiencies and cooling flows.

4. Gintz, J., Boeing Engineering and Construction, "Closed Cycle Brayton Advanced Central Receiver Solar Thermal Electric Power Plant," SAND 78-8511, November 1978, p. 85.

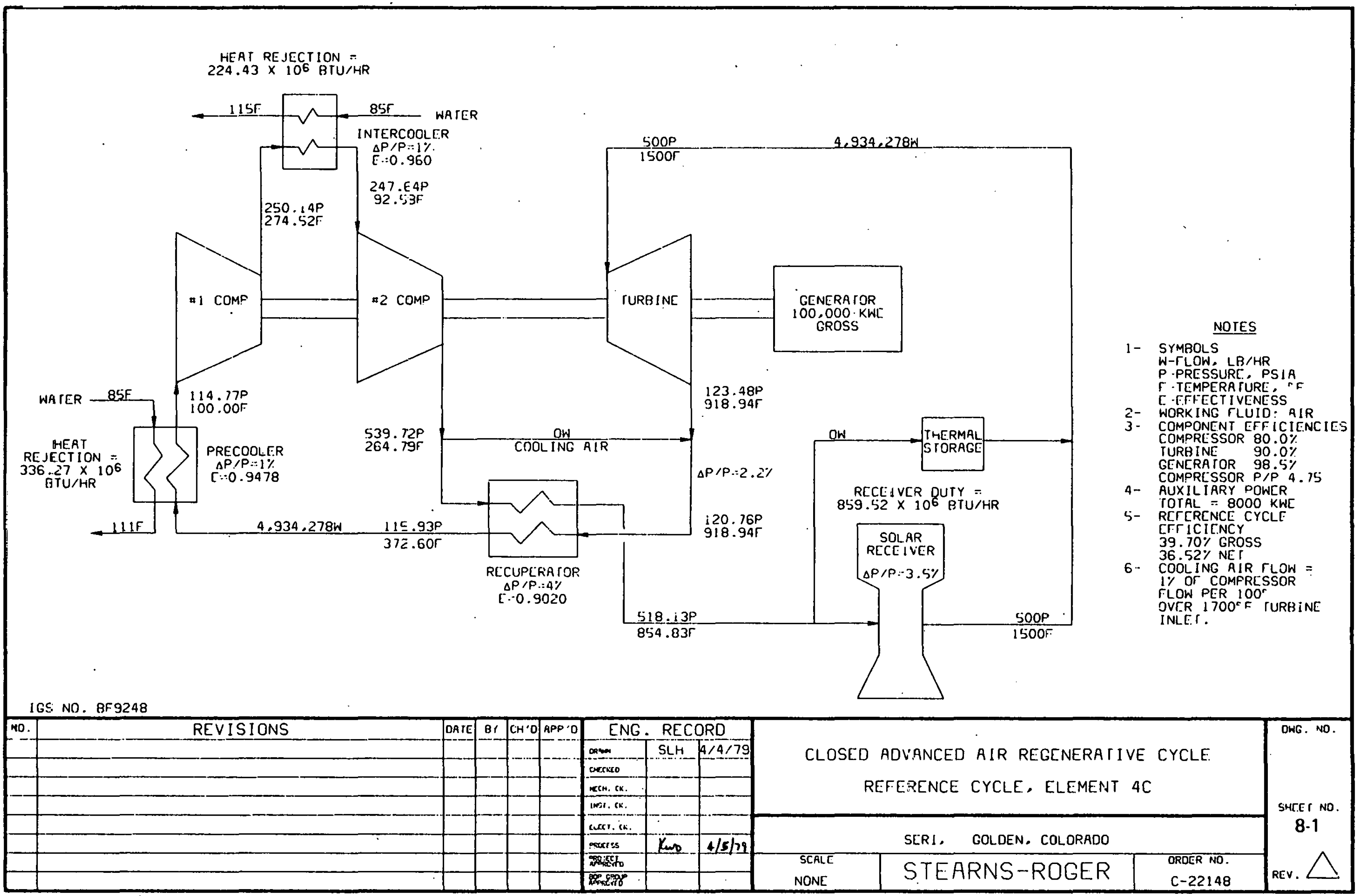

Figure 8—1. ELEMENT 4C - CLOSED ADVANCED AIR REGENERATIVE CYCLE, REFERENCE CYCLE

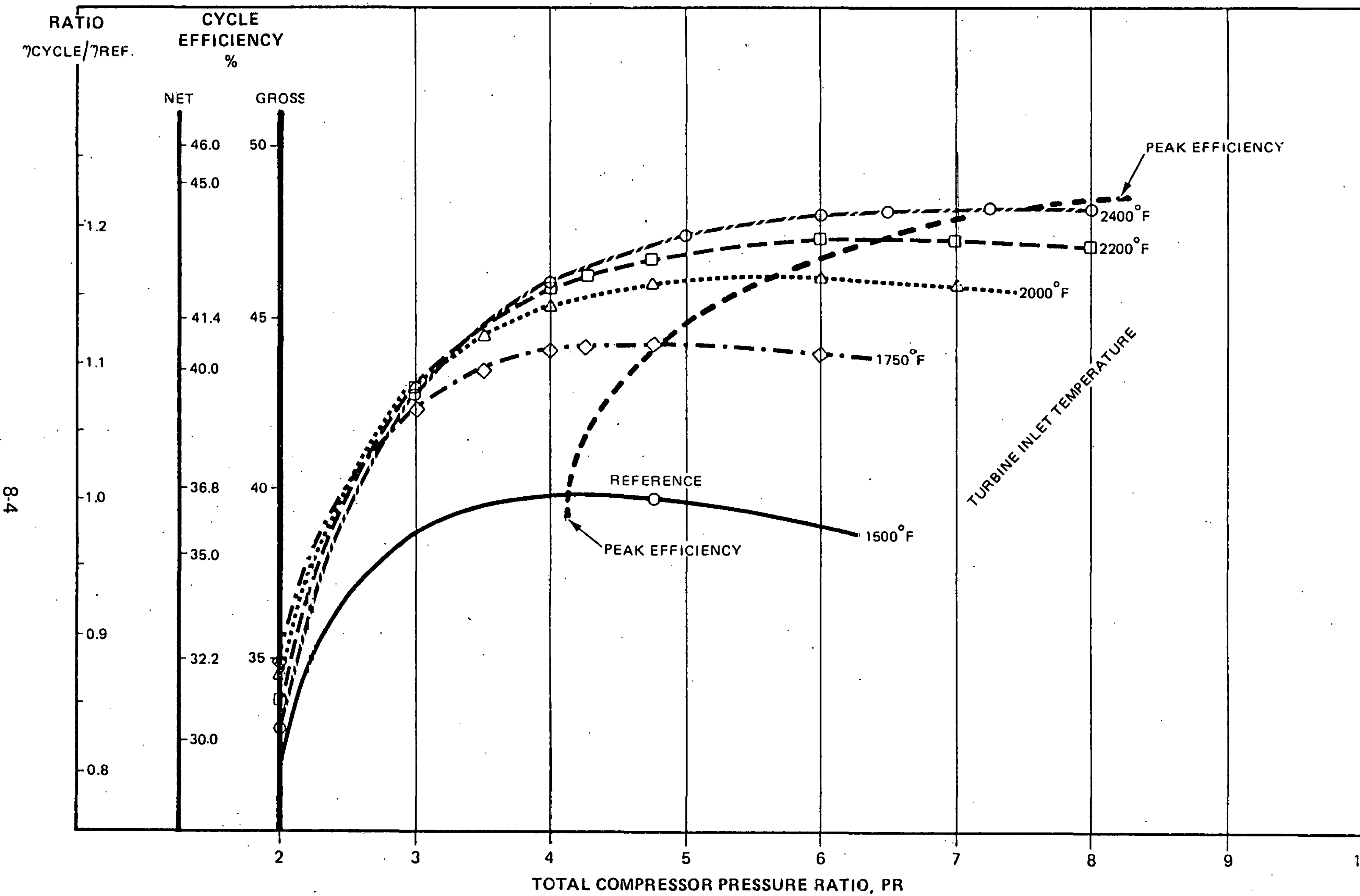

Figure 8–2. ELEMENT 4C - CLOSED ADVANCED AIR REGENERATIVE CYCLE, CYCLE EFFICIENCY VS. COMPRESSOR PRESSURE RATIO & TURBINE INLET TEMPERATURE

SECTION 9
ELEMENT 5 - REHEAT STEAM RANKINE CYCLE

9.1 INTRODUCTION

Cycle heat balances were performed for a 100 MWe (nominal) reheat steam Rankine cycle employing six stages of regenerative feedwater heating. The turbine exhausts to a condenser at 2.5 inches HgA (8.40×10^3 Pa). An allowance is made to extract up to 50 percent of the throttle flow from the second extraction on the low pressure turbine for use by the thermal storage system in a thermochemical reaction. The extraction pressure was varied to be 100, 150, and 200 psia (6.90×10^5, 1.03×10^6, and 1.38×10^6 Pa). Condensate is returned to the cycle in the condenser and is assumed to be at a temperature of 200°F (93.3°C).

The cycle was analyzed using an in-house computer program (D135E) which performs a mass and energy balance around the cycle. The shape of the turbine expansion curve is as specified in Figure 25 in Reference 1. The program accesses subroutines to calculate the fluid state conditions around the cycle using the relationships specified in the ASME steam tables.

9.2 CYCLE DESCRIPTION

The throttle conditions used for the reference cycle are 2400 psia (16.55×10^6 Pa) and 1000°F (537.8°C), with the reheat temperature of 1000°F (537.8°C). The final feedwater temperature selected is 480°F (248.9°C), which allows a reasonable pressure ratio across the high pressure turbine. A 10 percent pressure loss is assumed across the steam generator, and a pressure loss of 15 percent is assumed across the reheater.

For this study, the total heat input to the cycle was held constant while the throttle conditions and reheat temperature were varied. The throttle and reheat temperatures were varied over the range of 800°F to 1100°F (426.7°C to 593.3°C) and the throttle pressure was varied over the range of 1250 psia to 2400 psia (8.62×10^6 Pa to 16.55×10^6 Pa). The lower temperature limit was selected to limit the turbine exhaust steam quality to 88 percent and the upper temperature limit was selected as the limit of existing steam turbine technology. The throttle pressures selected are those normally used in the power industry. Representative high pressure and low pressure turbine efficiencies were calculated using the method specified in Reference 1. The representative turbine efficiencies were based on the throttle conditions of the reference cycle. The turbine efficiencies were held constant over the range of throttle conditions in order to prevent distortion of the effect of throttle conditions on overall cycle efficiency. In reality, the efficiency of the turbines will increase slightly as the amount of superheat of the throttle steam increases. Generator output was allowed to vary with the throttle conditions.

The turbine extractions provide steam to three high-pressure closed feedwater heaters, an open deaerating feedwater heater, and two low-pressure closed feedwater heaters. The heater performance characteristics are derived from standard design values and are held constant over the range of throttle conditions. Most turbine steam leakages are not accounted for in the cycle, as these leakages are small when compared to other cycle flows. One leakage was included, the shaft leakage from the high pressure turbine to the low pressure turbine, as this is three percent of the throttle flow. The neglected leakages include shaft leakage from the exhaust of the HP turbine, the sealing flows to the LP turbine shaft seals, and packing leakage from the turbine stop and control valves. The total of these leakages are typically less than one percent of the throttle flow in current commercial units.

A cycle schematic showing component efficiencies and cycle flow data is shown in Figure 9—1 for the base cycle, and in Figure 9—2 for the base cycle with an extraction flow of 50 percent of throttle flow. Plots of cycle efficiency versus throttle temperature are shown in Figure 9—3 and Figure 9—4. A normalized representation of cycle efficiency with respect to base cycle efficiency versus throttle temperature is shown in Figure 9—5. A plot of generator output versus throttle temperature is shown in Figure 9—6. The change in cycle efficiency versus the extraction flow is shown in Figure 9—7. The change in generator output versus the extraction flow is shown in Figure 9—8. Gross cycle efficiency is defined as the total cycle input energy divided into the generator output. Net cycle efficiency is defined as the total cycle input energy divided into the generator output less the plant auxiliary power required (which is assumed to be 8 percent of the generator output). Plant auxiliary power includes that power used by the receiver feed pump, condensate pump, circulating water pump, cooling tower fans, controls, plant lighting, plant HVAC, solar collector field usage, etc.

REFERENCE

1. General Electric Company, "A Method for Predicting the Performance of Steam Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

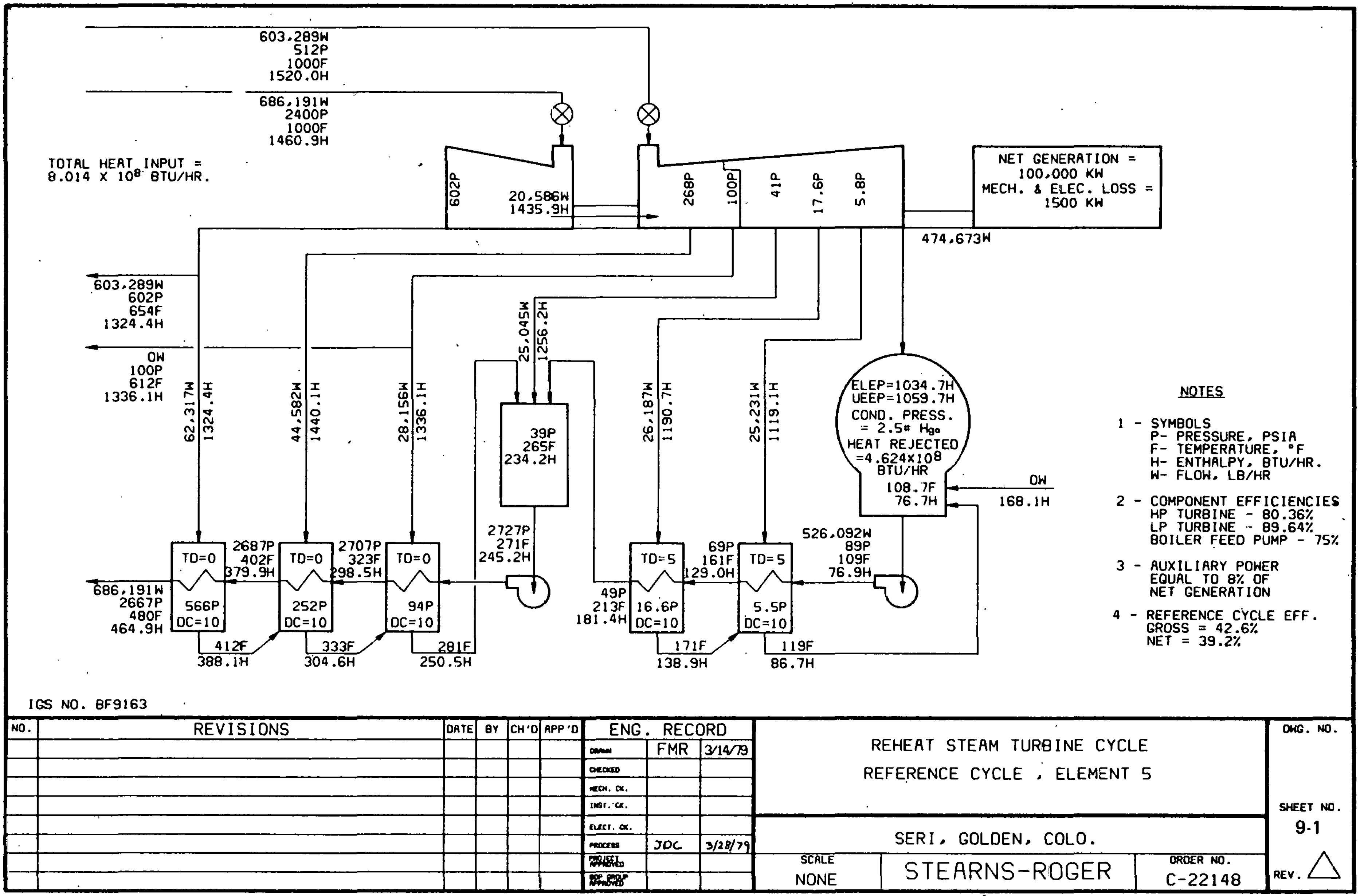

Figure 9—1. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, REFERENCE CYCLE

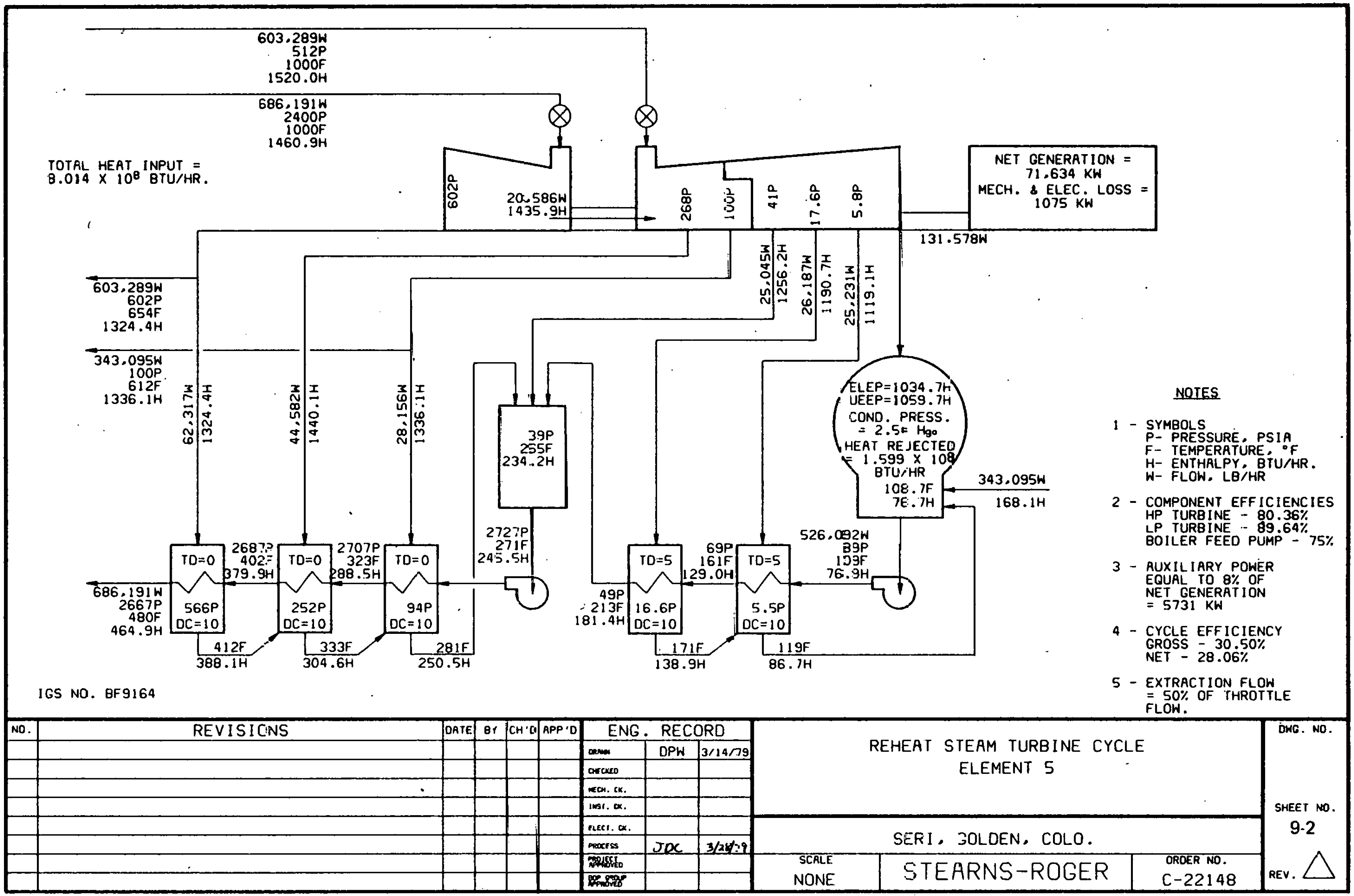

Figure 9–2. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE

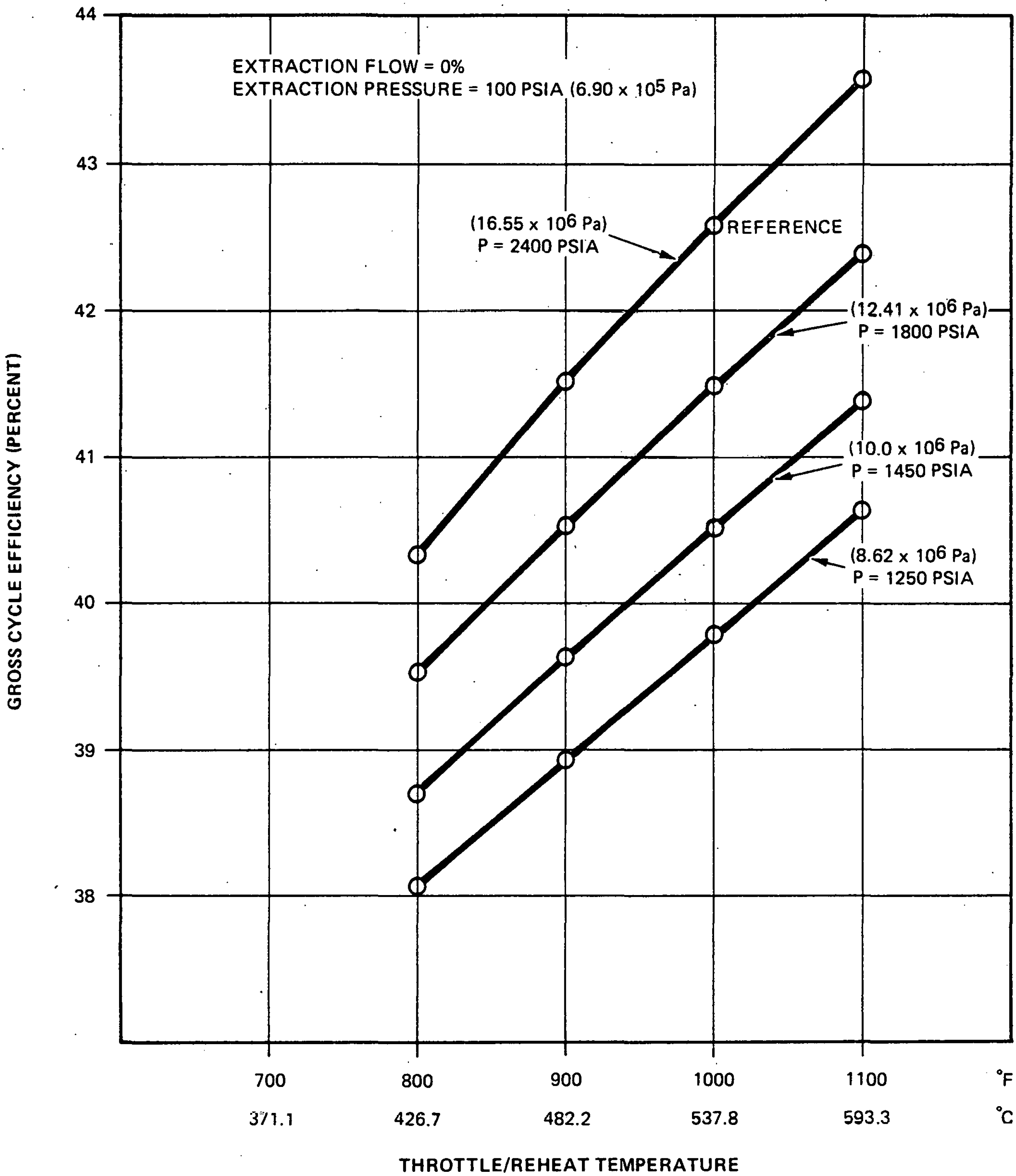

Figure 9–3. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, GROSS CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

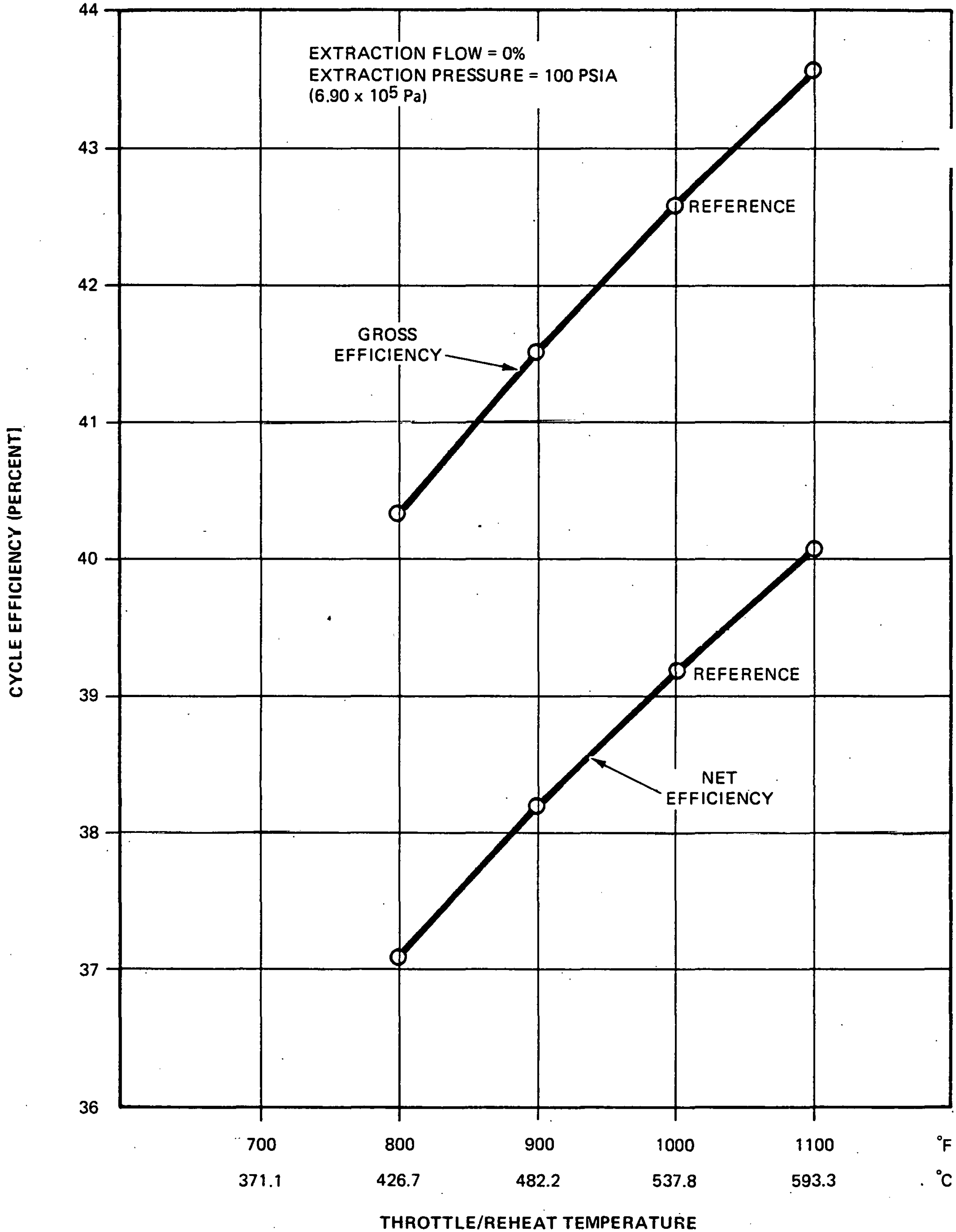

Figure 9–4. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

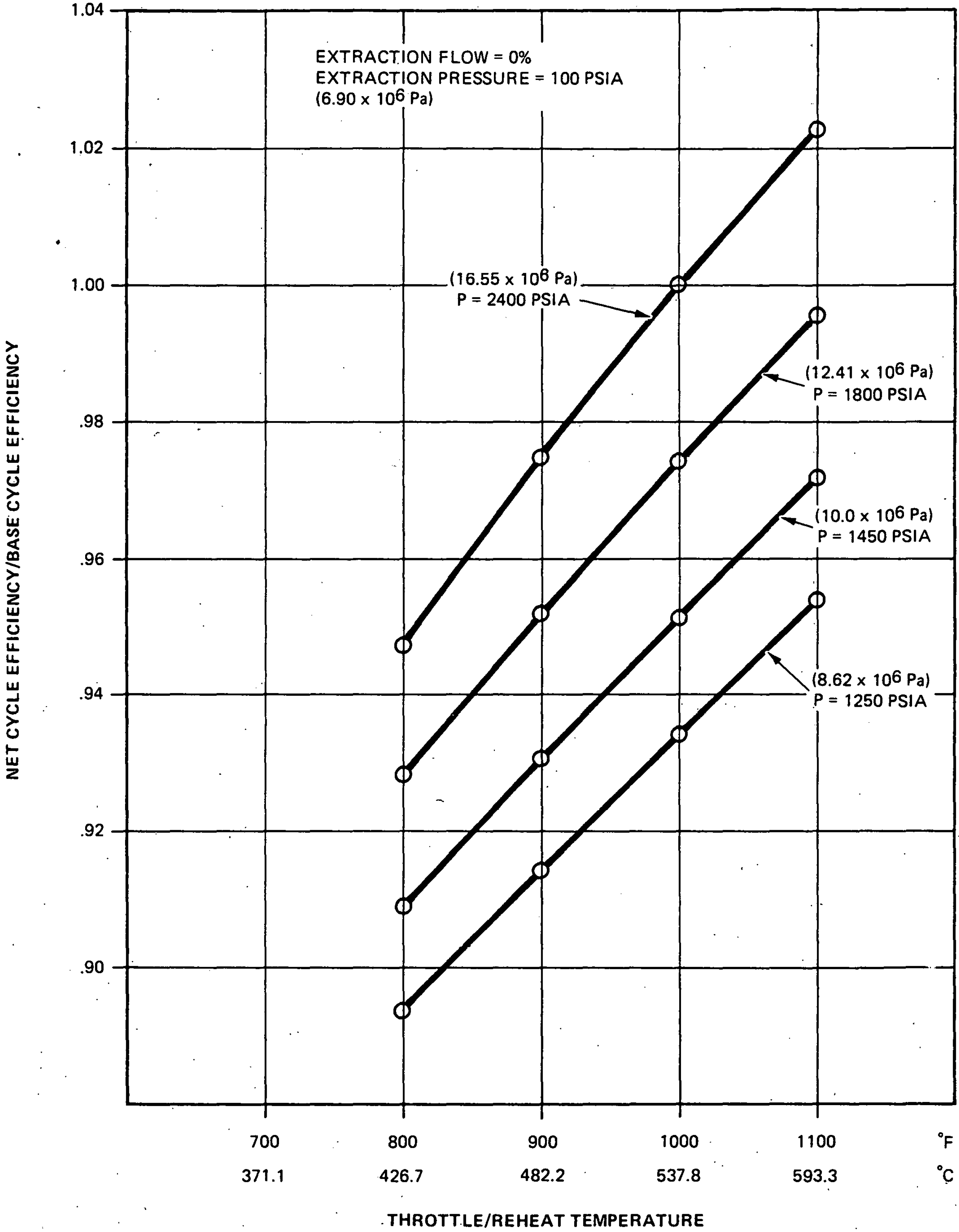

Figure 9–5. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, RELATIVE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

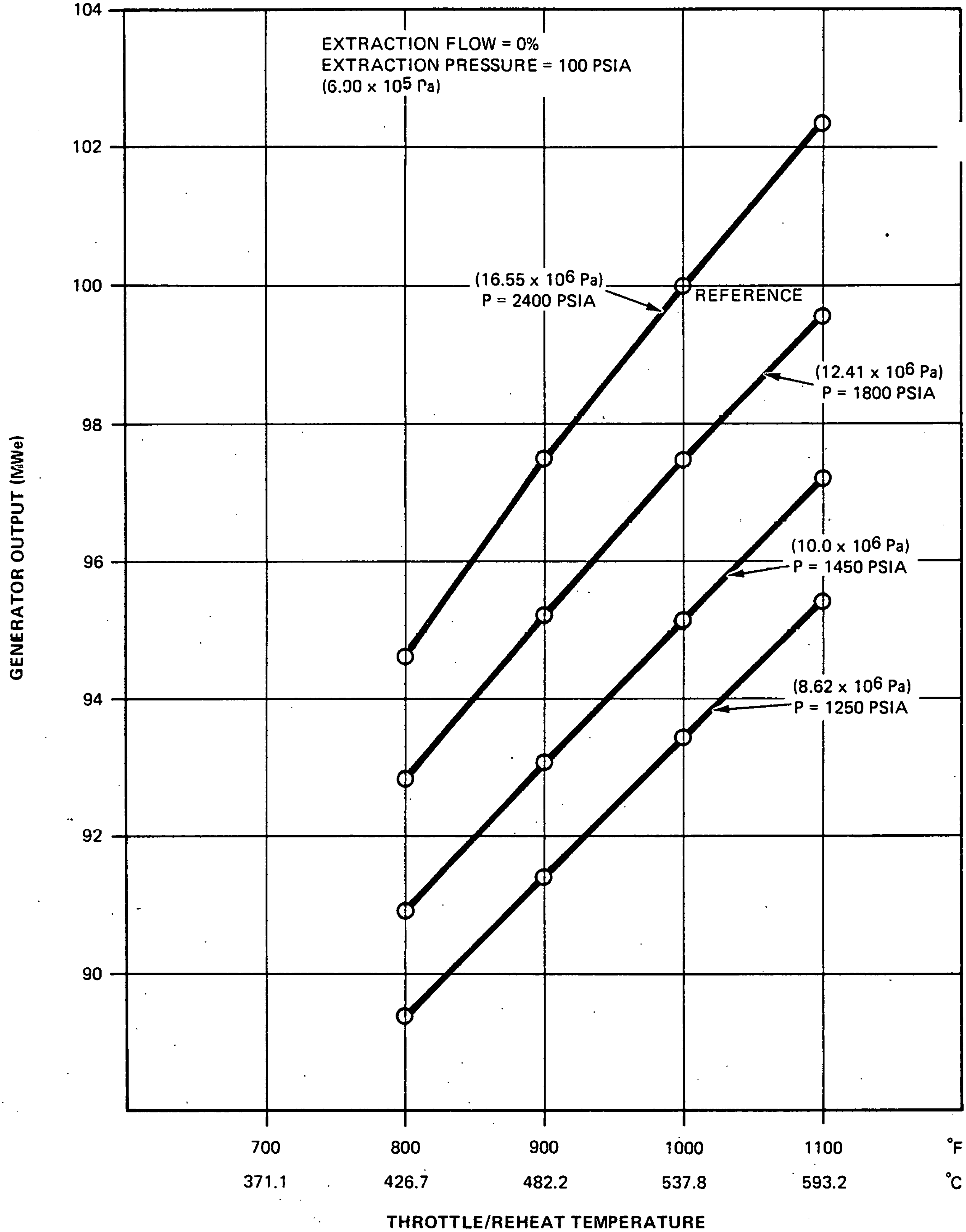

Figure 9–6. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, GENERATOR OUTPUT VS. THROTTLE TEMPERATURE

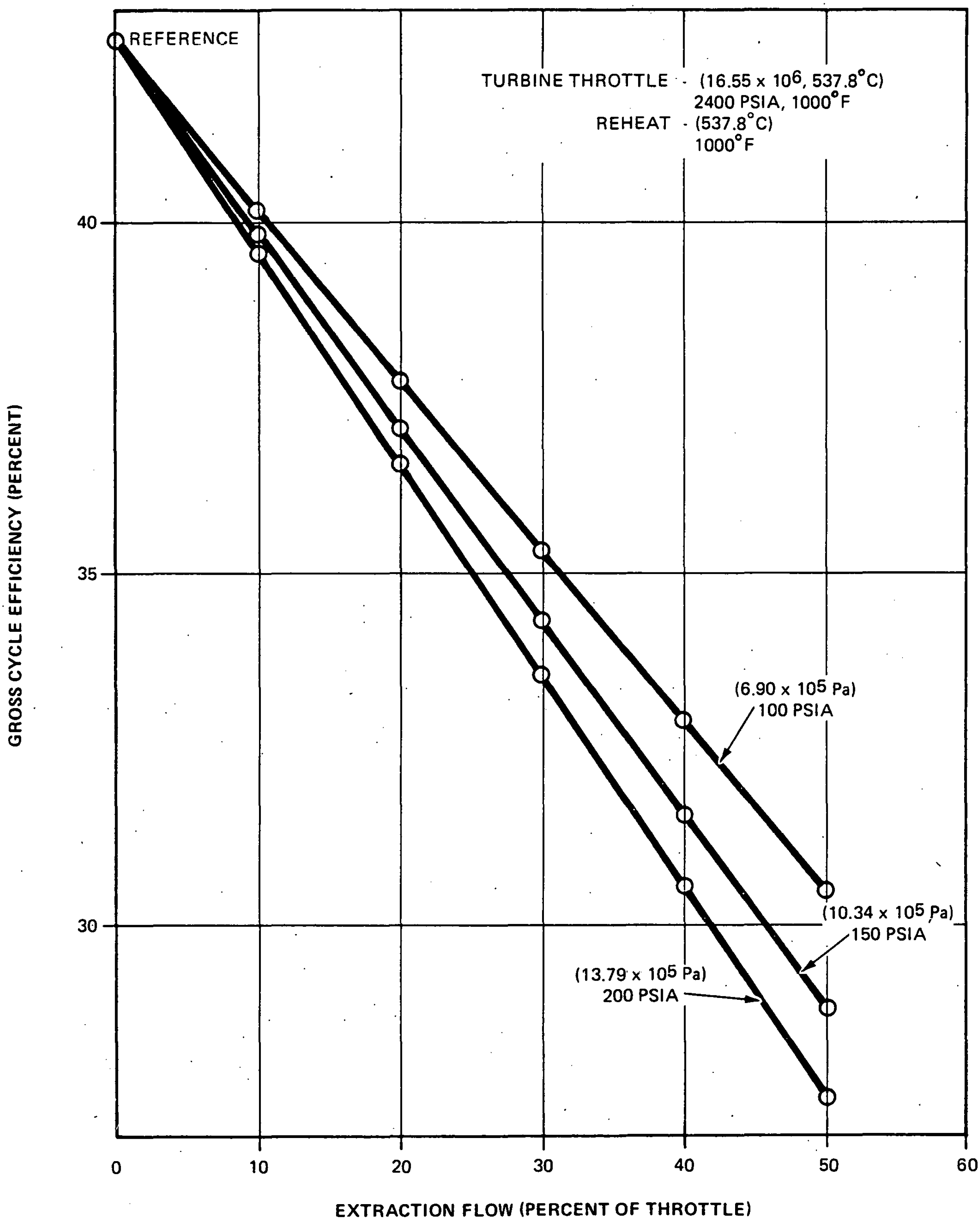

Figure 9–7. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, GROSS CYCLE EFFICIENCY VS. EXTRACTION FLOW

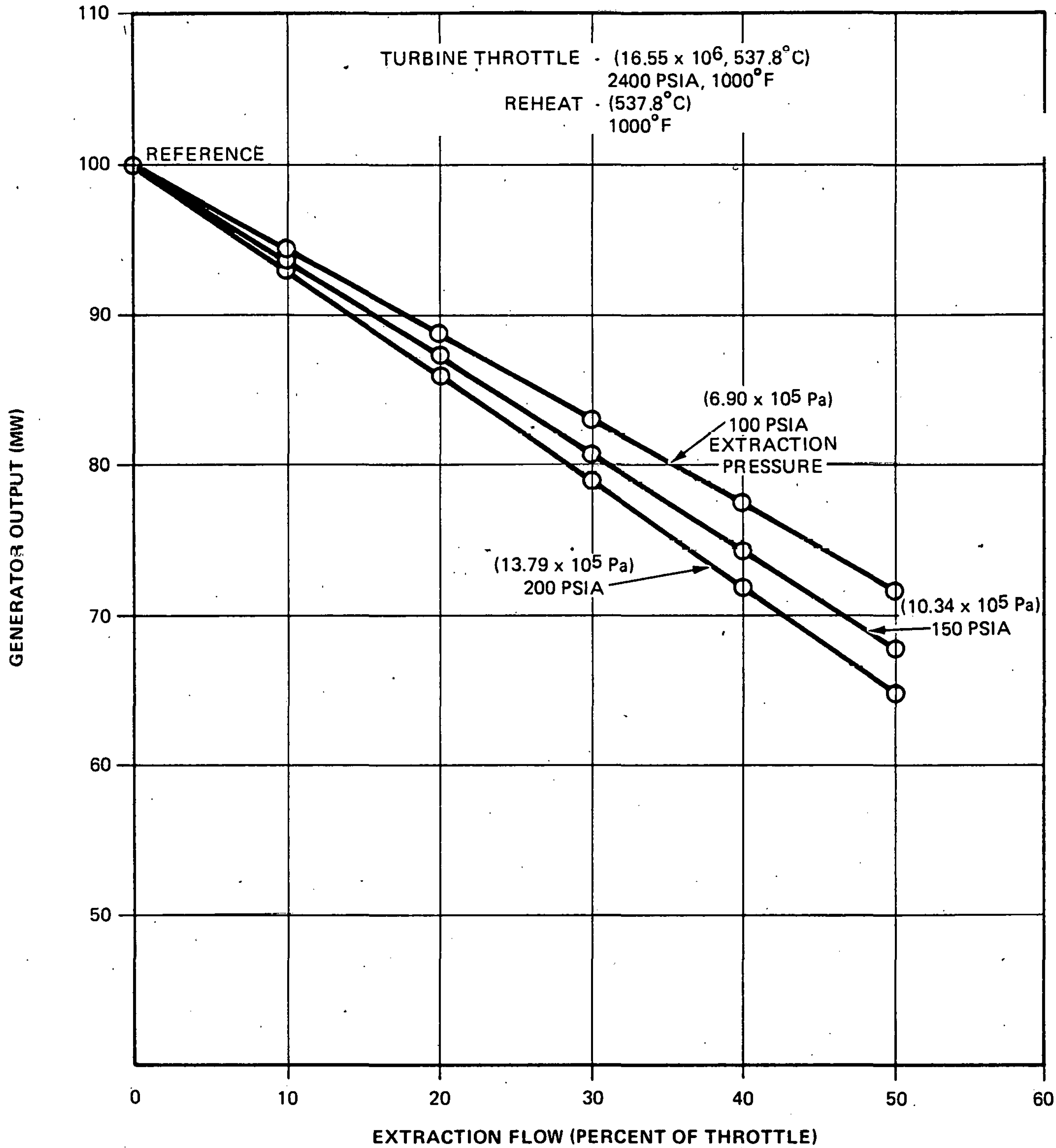

Figure 9–8. ELEMENT 5 - REHEAT STEAM RANKINE CYCLE, GENERATOR OUTPUT VS. EXTRACTION FLOW

SECTION 10
ELEMENT 6A - STEAM RANKINE CYCLE

10.1 INTRODUCTION

Cycle heat balances were performed for a 10 MWe (nominal) non-reheat steam Rankine cycle employing four stages of regenerative feedwater heating. The turbine exhausts to a condenser at 2.5 inches HgA (8.46×10^3 Pa).

The cycle was analyzed using an in-house computer program (D135A), which performs a mass and energy balance around the specified cycle. The shape of the turbine expansion curve is as specified in Figure 25 of Reference 1. The program accesses subroutines to calculate the fluid state conditions around the cycle using the relationships specified in the ASME steam tables.

10.2 CYCLE DESCRIPTION

The throttle conditions used for the reference cycle are 1250 psia (8.62×10^6 Pa) and 950°F (510°C). The final feedwater temperature is 400°F (204.4°C), which allows a reasonable pressure ratio from the throttle to the first extraction point. A 10 percent pressure loss is assumed across the steam generator.

For this study, the total heat input to the cycle was held constant while the throttle conditions were varied. The throttle temperature was varied over the range of 700°F to 1100°F (371.1°C to 593.3°C), and the throttle pressure was varied over the range of 850 psia to 1800 psia (5.86×10^6 Pa to 12.41×10^6 Pa). The pressures selected are those which are normally used in the power industry. The upper temperature limit was selected as the limit of existing steam turbine technology. The lower limit of temperature varies with the throttle pressure, as it is desirable to maintain the turbine exhaust steam quality at a value greater than 84 percent. For the 1800 psia (12.41×10^6 Pa) pressure, the minimum temperature is 825°F (440.6°C); for the 1450 psia (10.0×10^6 Pa) pressure, the minimum temperature is 775°F (412.8°C); for the 1250 psia (8.62×10^6 Pa) pressure, the minimum temperature is 725°F (385°C); and for the 850 psia (5.86×10^6 Pa) pressure, the minimum temperature is 650°F (343.3°C). A representative turbine efficiency was calculated using the method specified in Reference 1. The turbine efficiency was based on the throttle conditions of the reference cycle. The turbine efficiency was held constant over the range of throttle conditions in order to prevent distortion of the effect of throttle conditions on overall cycle efficiency. In reality, the efficiency of the turbine will increase slightly as the amount of superheat of the throttle steam increases. Generator output was allowed to vary with the throttle conditions.

Turbine extractions provide steam to two high-pressure closed feedwater heaters, an open deaerating feedwater heater, and a low-pressure closed feedwater heater. The heater performance characteristics are derived from standard design values and are held constant over the range of throttle conditions. Turbine steam leakages are not accounted for in the cycle, as these leakages are small when compared to other cycle flows.

These leakages include shaft leakage from the high pressure end of the turbine, seal steam flow to the low pressure end of the turbine, and throttle stop and control valve packing leakage. The total of these leakages is typically less than one percent of the throttle flow in current commercial units.

A cycle schematic showing component efficiencies and cycle flow data is shown in Figure 10—1 for the base cycle. Plots of cycle efficiency versus throttle temperature are shown in Figure 10—2 and Figure 10—3. A normalized representation of cycle efficiency with respect to the base cycle efficiency versus throttle temperature is shown in Figure 10—4. A plot of generator output versus throttle temperature is shown in Figure 10—5. Gross cycle efficiency is defined as the total cycle input energy divided into the generator output. Net cycle efficiency is defined as the total cycle input energy divided into the generator output less the plant auxiliary power requirement (which is assumed to be 8 percent of the generator output). Plant auxiliary power includes that power used by the boiler feed pump, condensate pump, circulating water pump, cooling tower fans, controls, plant lighting, plant HVAC, solar collector field usage, etc.

REFERENCE

1. General Electric Company, "A Method for Predicting the Performance of Steam Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

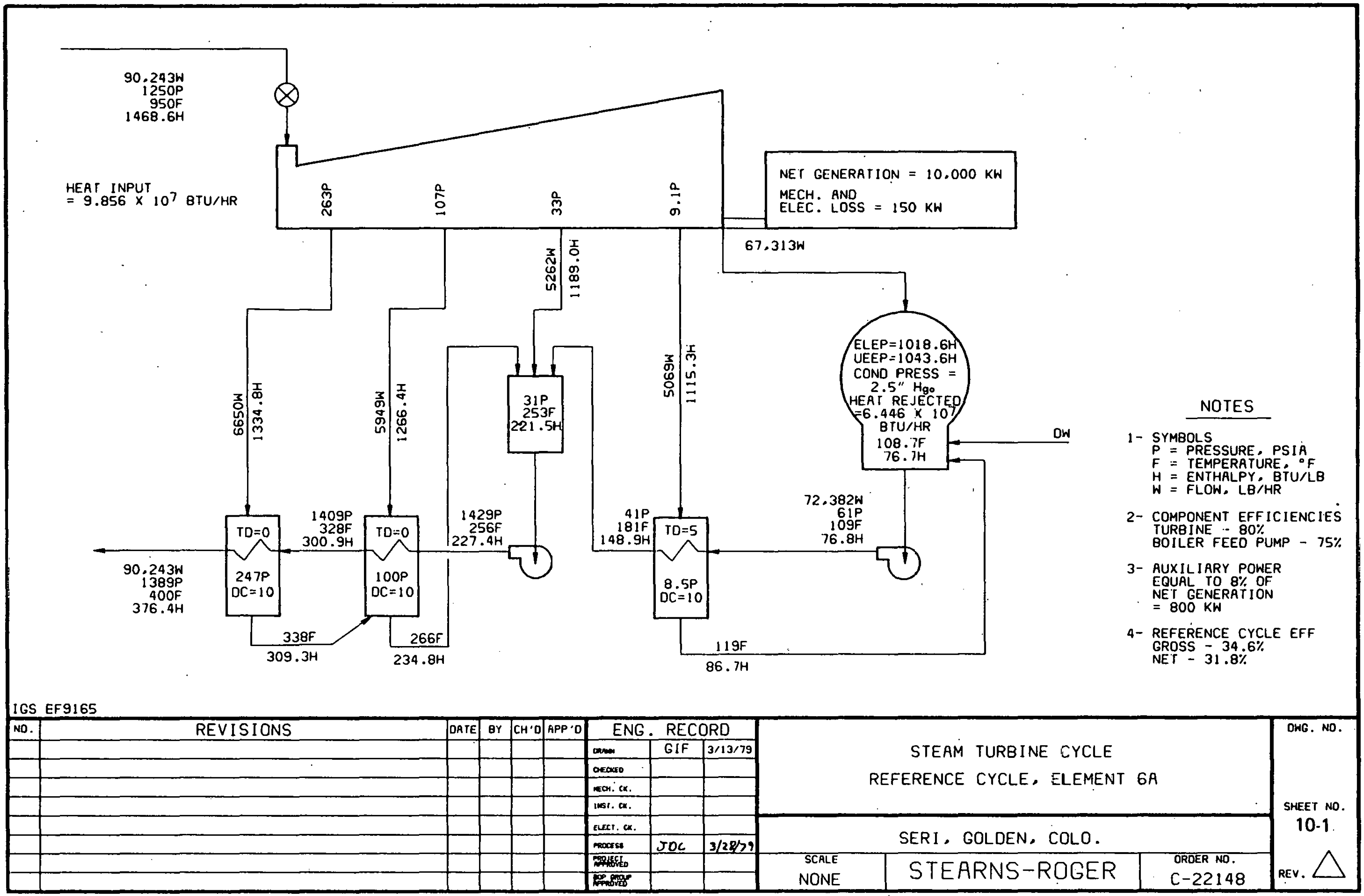

Figure 10–1. ELEMENT 6A - STEAM RANKINE CYCLE, REFERENCE CYCLE

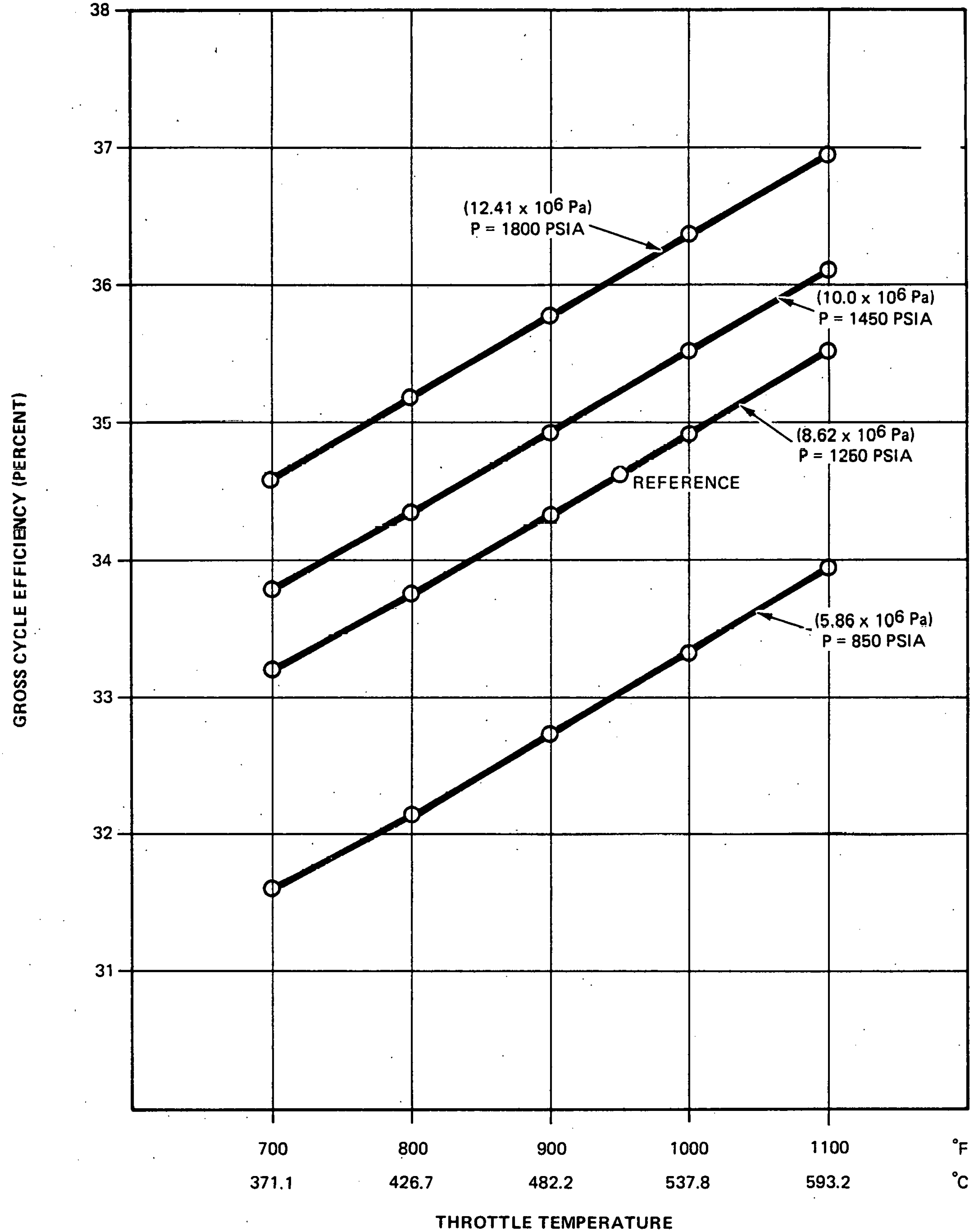

Figure 10–2. ELEMENT 6A - STEAM RANKINE CYCLE,
GROSS CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

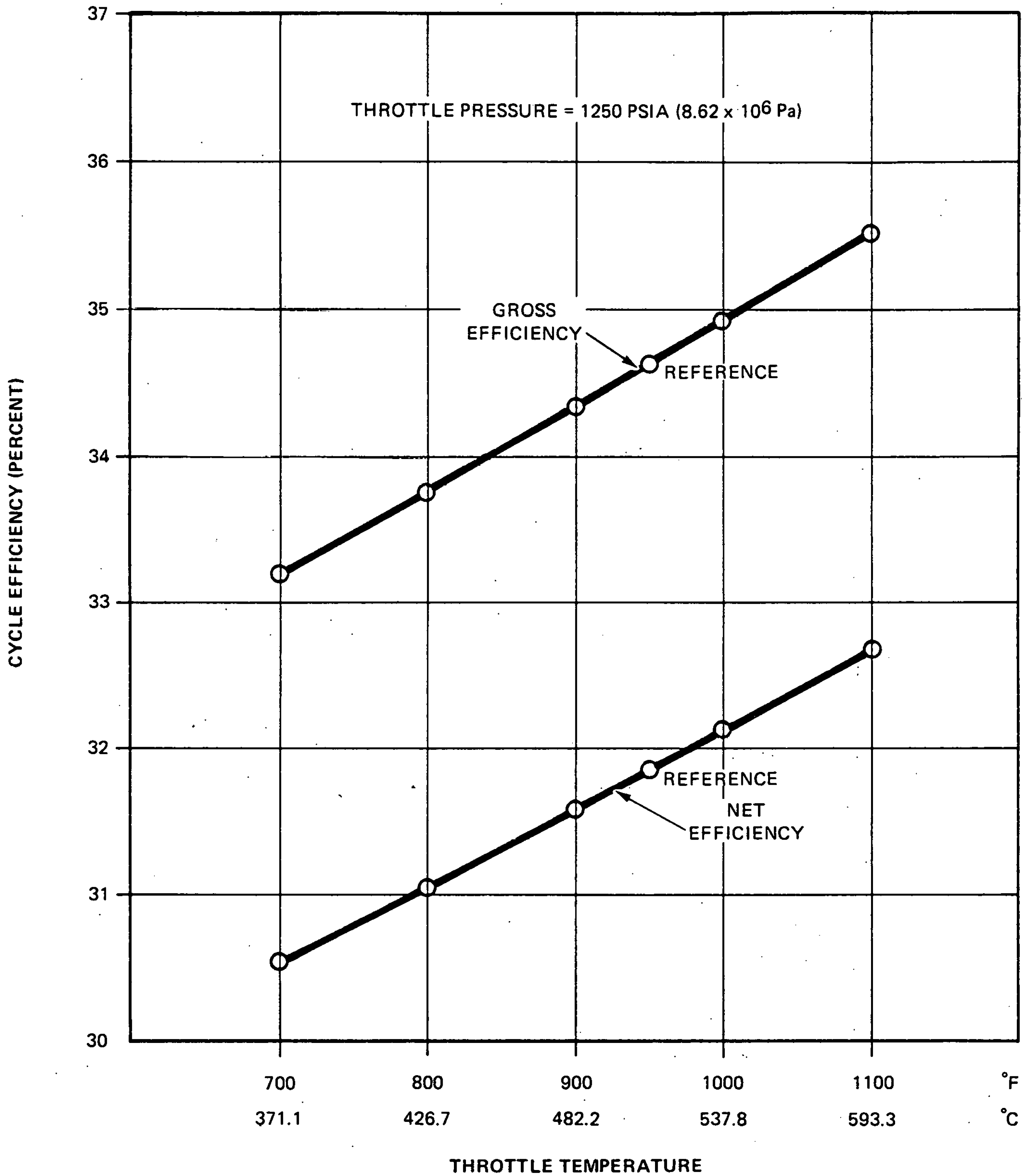

Figure 10–3. ELEMENT 6A - STEAM RANKINE CYCLE, CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

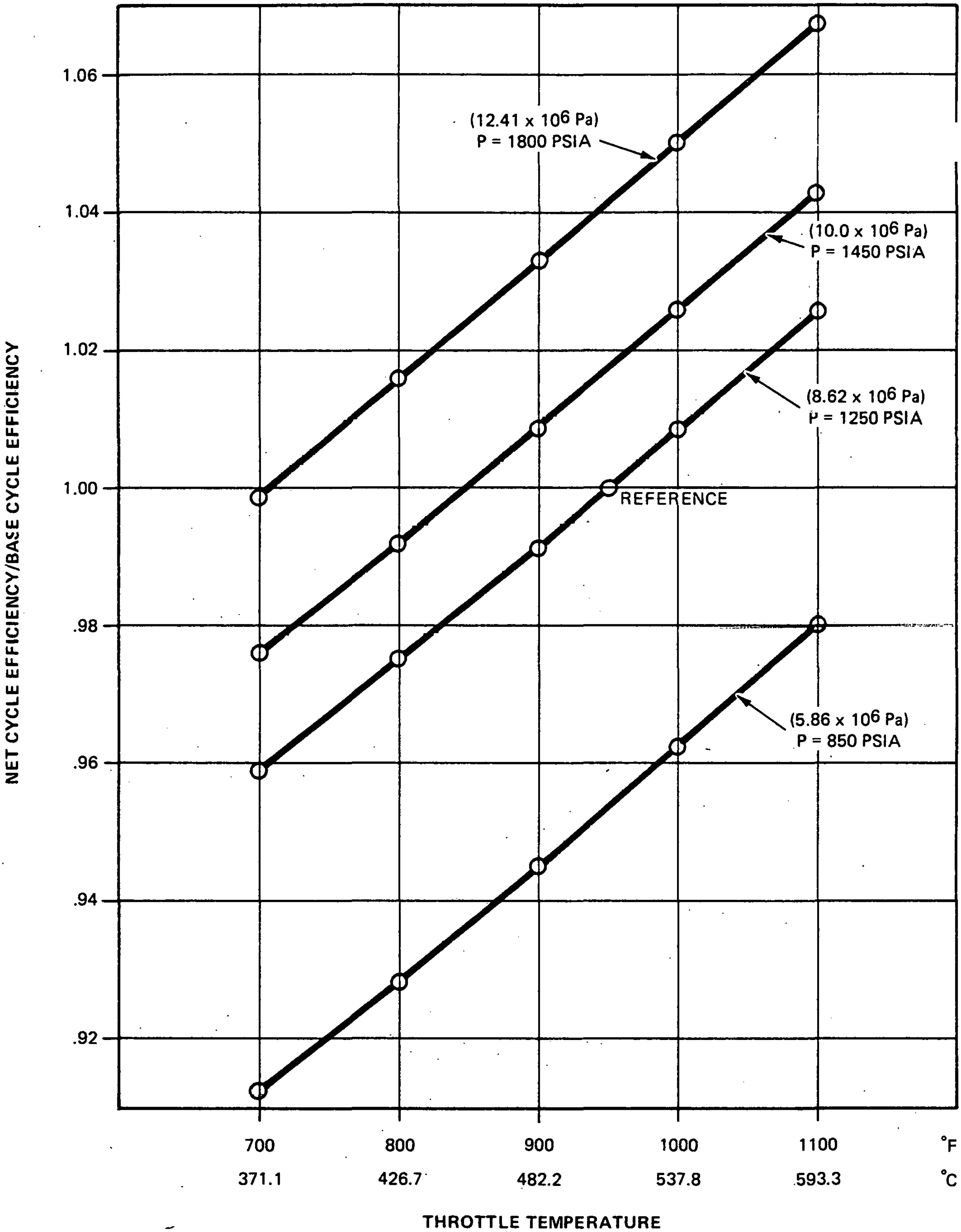

Figure 10—4. ELEMENT 6A - STEAM RANKINE CYCLE, RELATIVE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

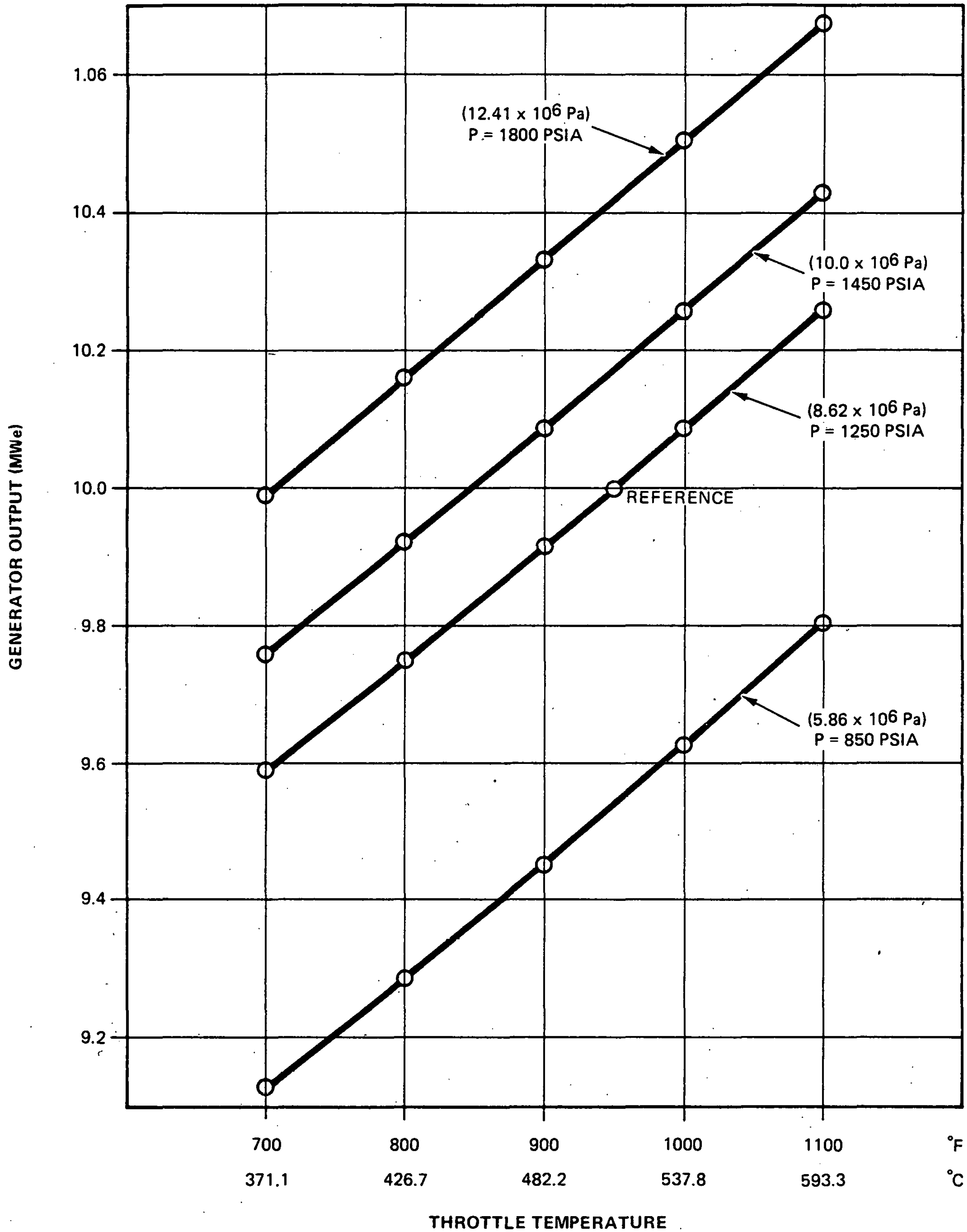

Figure 10–5. ELEMENT 6A - STEAM RANKINE CYCLE, GENERATOR OUTPUT VS. THROTTLE TEMPERATURE

SECTION 11
ELEMENT 6B - STEAM RANKINE CYCLE

11.1 INTRODUCTION

Cycle heat balances were performed for a 300 KWe (nominal) non-reheat steam Rankine cycle employing two stages of regenerative feedwater heating. The turbine exhausts to a condenser at 2.5 inches HgA (8.46×10^3 Pa).

The cycle was analyzed using an in-house computer program (D135A), which performs a mass and energy balance around the specified cycle. The shape of the turbine expansion curve is as specified in Figure 25 of Reference 1. The program accesses subroutines to calculate the fluid state conditions around the cycle using the relationships specified in the ASME steam tables.

11.2 CYCLE DESCRIPTION

The throttle conditions used for the reference cycle are 1250 psia (8.62×10^6 Pa) and 950°F (510°C). The final feedwater temperature is 400°F (204.4°C), which allows a reasonable pressure ratio from the throttle to the first extraction point. A 10 percent pressure loss is assumed across the steam generator.

For this study, the total heat input to the cycle was held constant while the throttle conditions were varied. The throttle temperature was varied over the range of 700°F to 1100°F (371.1°C to 593.3°C), and the throttle pressure was varied over the range of 850 psia to 1800 psia (5.86×10^6 Pa to 12.41×10^6 Pa). The pressures selected are those which are normally used in the power industry. The upper temperature limit was selected as the limit of existing steam turbine technology. The lower limit of temperature varies with the throttle pressure, as it is desirable to maintain the turbine exhaust steam quality at a value greater than 84 percent. For the 1800 psia (12.41×10^6 Pa) pressure, the minimum temperature is 825°F (440.6°C); for the 1450 psia (10.0×10^6 Pa) pressure, the minimum temperature is 725°F (385°C); and for the 850 psia (5.86×10^6 Pa) pressure, the minimum temperature is 650°F (343.3°C). A representative turbine efficiency was calculated using the method specified in Reference 1. The turbine efficiency was based on the throttle conditions of the reference cycle. The turbine efficiency was held constant over the range of throttle conditions in order to prevent distortion of the effect of throttle conditions on overall cycle efficiency. In reality, the efficiency of the turbine will increase slightly as the amount of superheat of the throttle steam increases. Generator output was allowed to vary with the throttle conditions.

Turbine extractions provide steam to one high-pressure closed feedwater heater and one open deaerating feedwater heater. The heater performance characteristics are derived from standard design values and are held constant over the range of throttle conditions. Turbine steam leakages are not accounted for in the cycle, as these leakages are small when compared to other cycle flows. These leakages include shaft leakage from the high pressure end of the turbine, seal steam flow to the low pressure end of the turbine, and throttle stop and control valve packing leakage. The total of these leakages is typically less than one percent of the throttle flow in current commercial units.

A cycle schematic showing component efficiencies and cycle flow data is shown in Figure 11–1 for the base cycle. Plots of cycle efficiency versus throttle temperature are shown in Figure 11–2 and Figure 11–3. A normalized representation of cycle efficiency with respect to the base cycle efficiency versus throttle temperature is shown in Figure 11–4. A plot of generator output versus throttle temperature is shown in Figure 11–5. Gross cycle efficiency is defined as the total cycle input energy divided into the generator output. Net cycle efficiency is defined as the total cycle input energy divided into the generator output less the plant auxiliary power requirement (which is assumed to be 8 percent of the generator output). Plant auxiliary power includes that power used by the receiver feed pump, condensate pump, circulating water pump, cooling tower fans, controls, plant lighting, plant HVAC, solar collector field usage, etc.

REFERENCE

1. General Electric Company, "A Method for Predicting the Performance of Steam Turbine Generators . . . 16,500 KW and Larger," (GER-2007C) Revised July 1974.

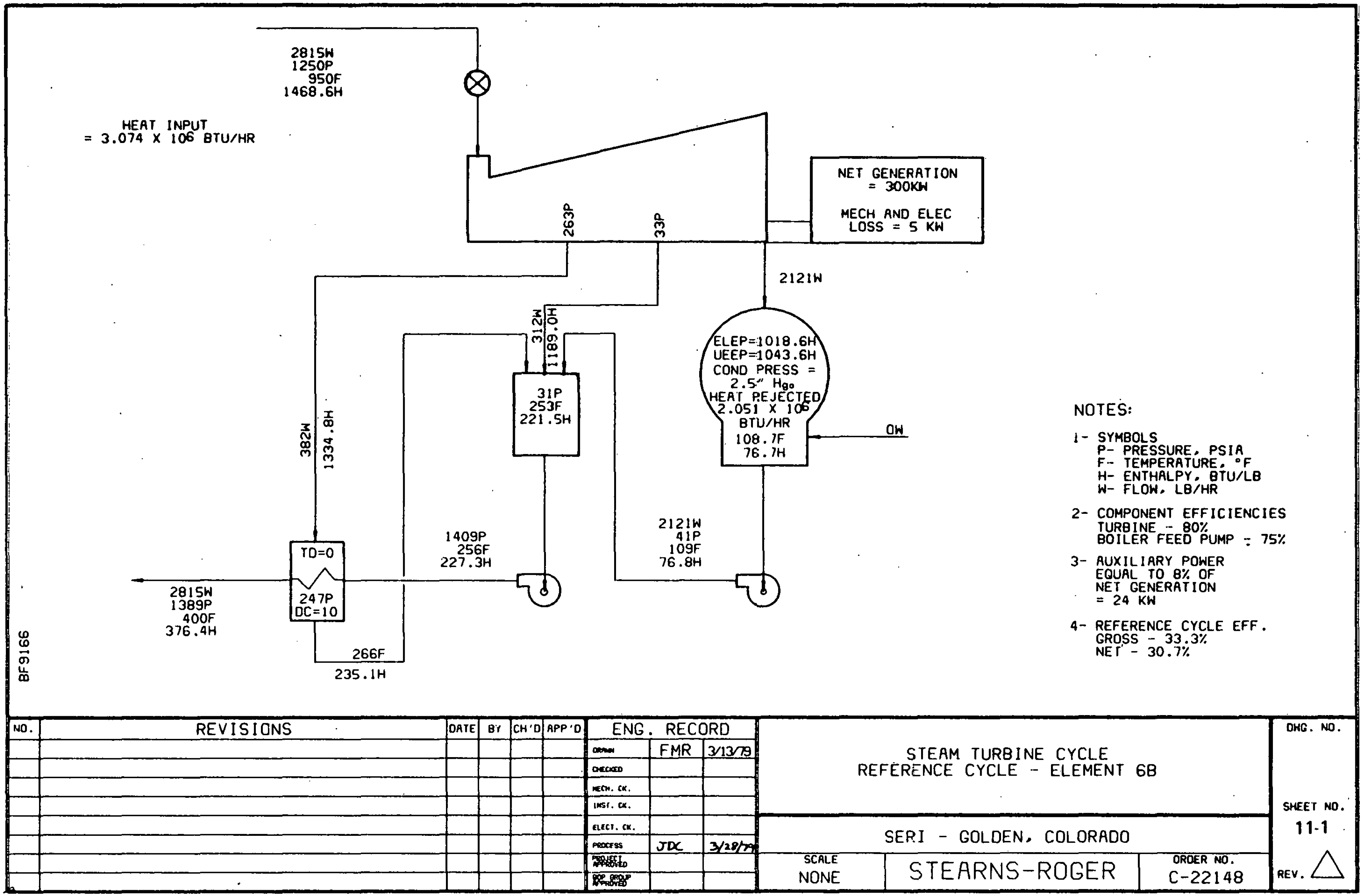

Figure 11—1. ELEMENT 6B - STEAM RANKINE CYCLE, REFERENCE CYCLE

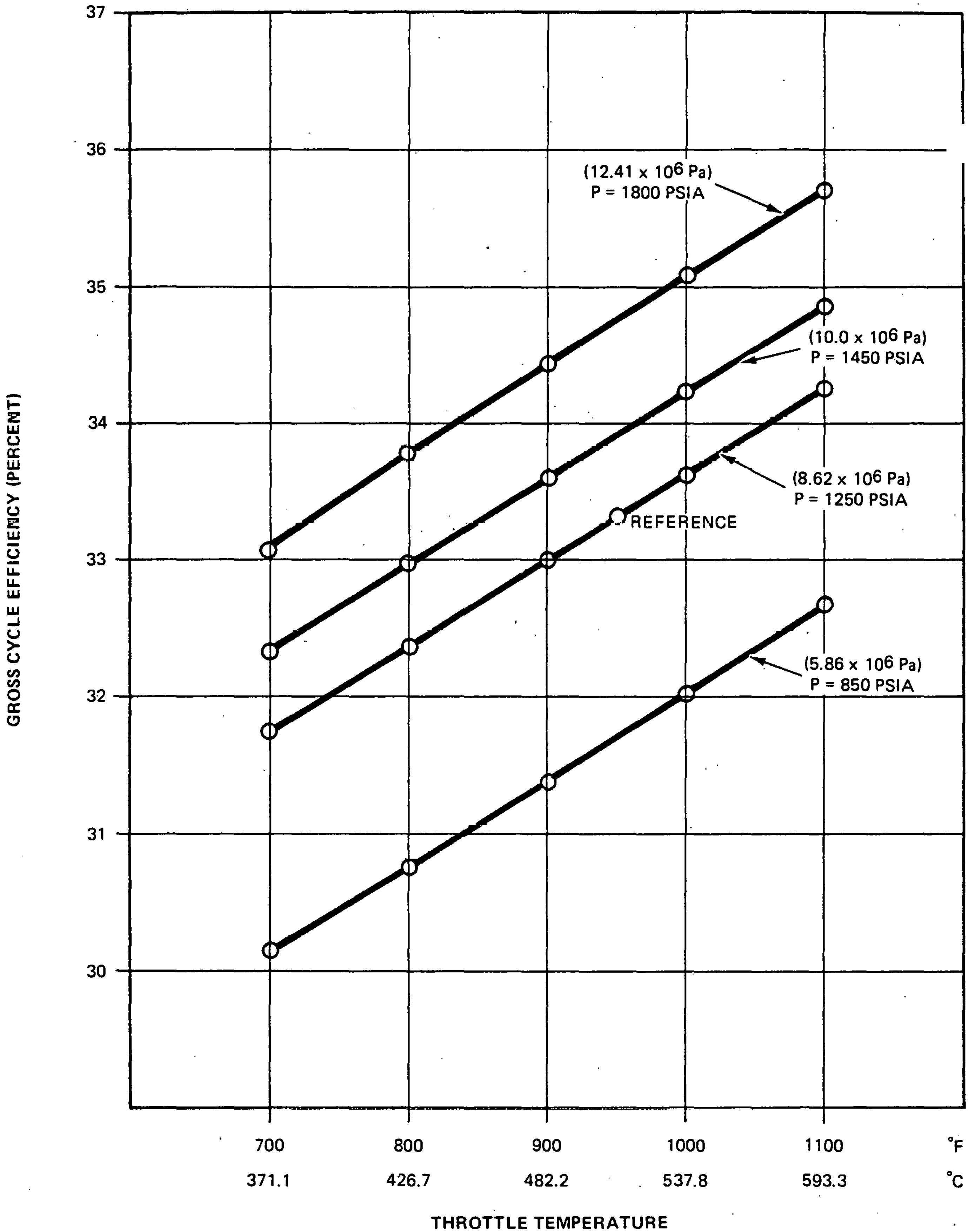

Figure 11–2. ELEMENT 6B - STEAM RANKINE CYCLE, GROSS CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

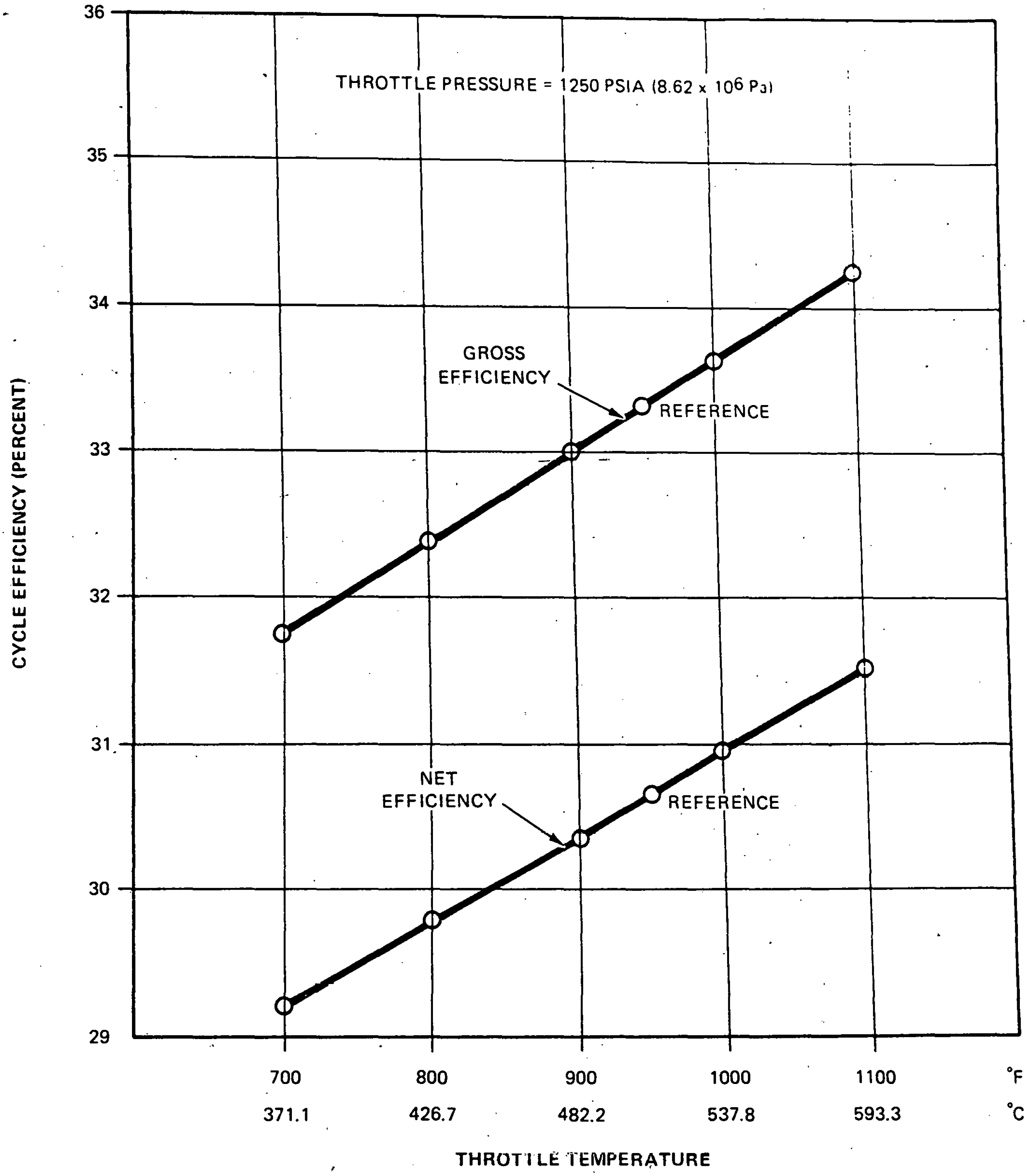

Figure 11–3. ELEMENT 6B - STEAM RANKINE CYCLE, CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

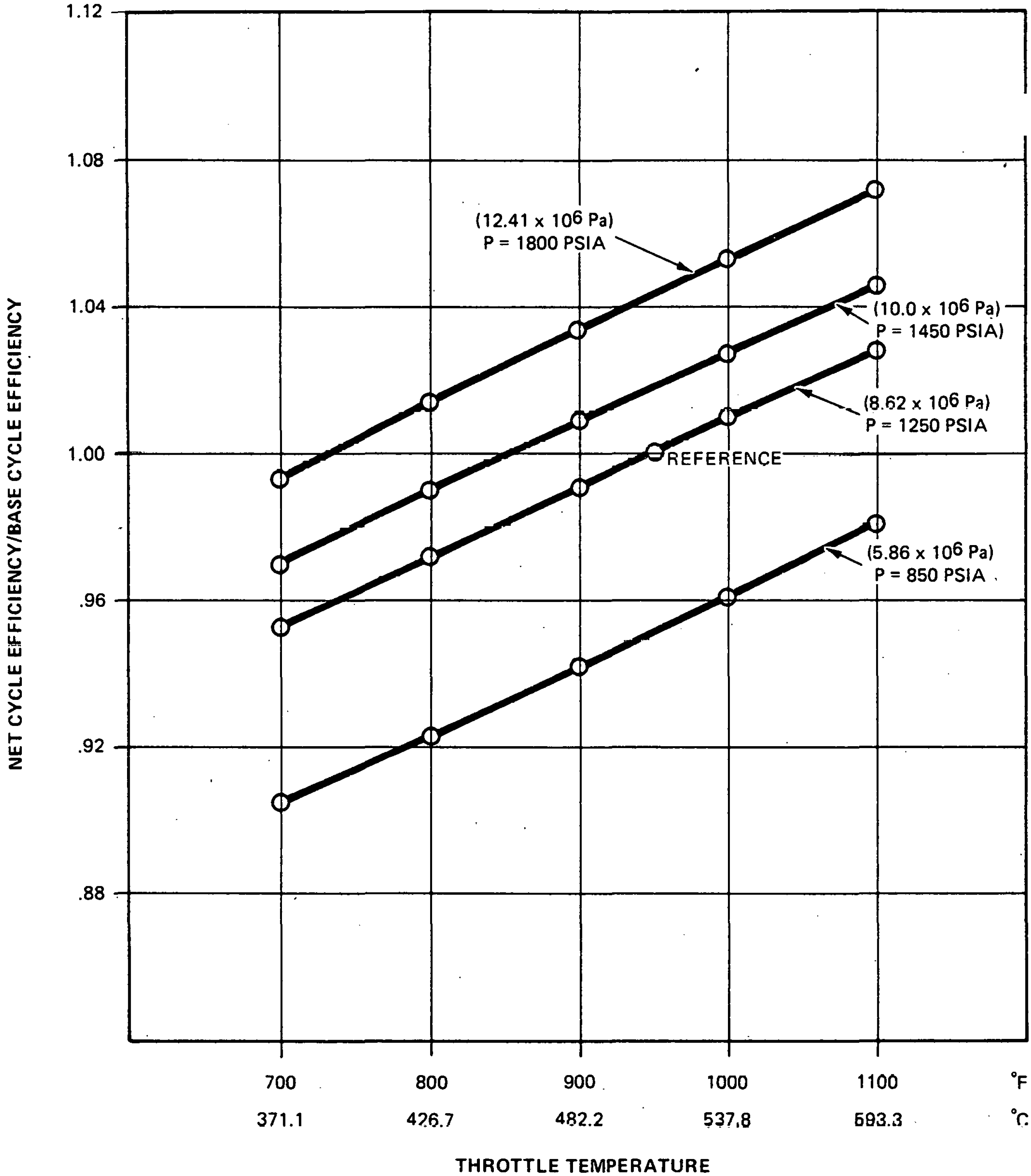

Figure 11–4. ELEMENT 6B - STEAM RANKINE CYCLE, RELATIVE CYCLE EFFICIENCY VS. THROTTLE TEMPERATURE

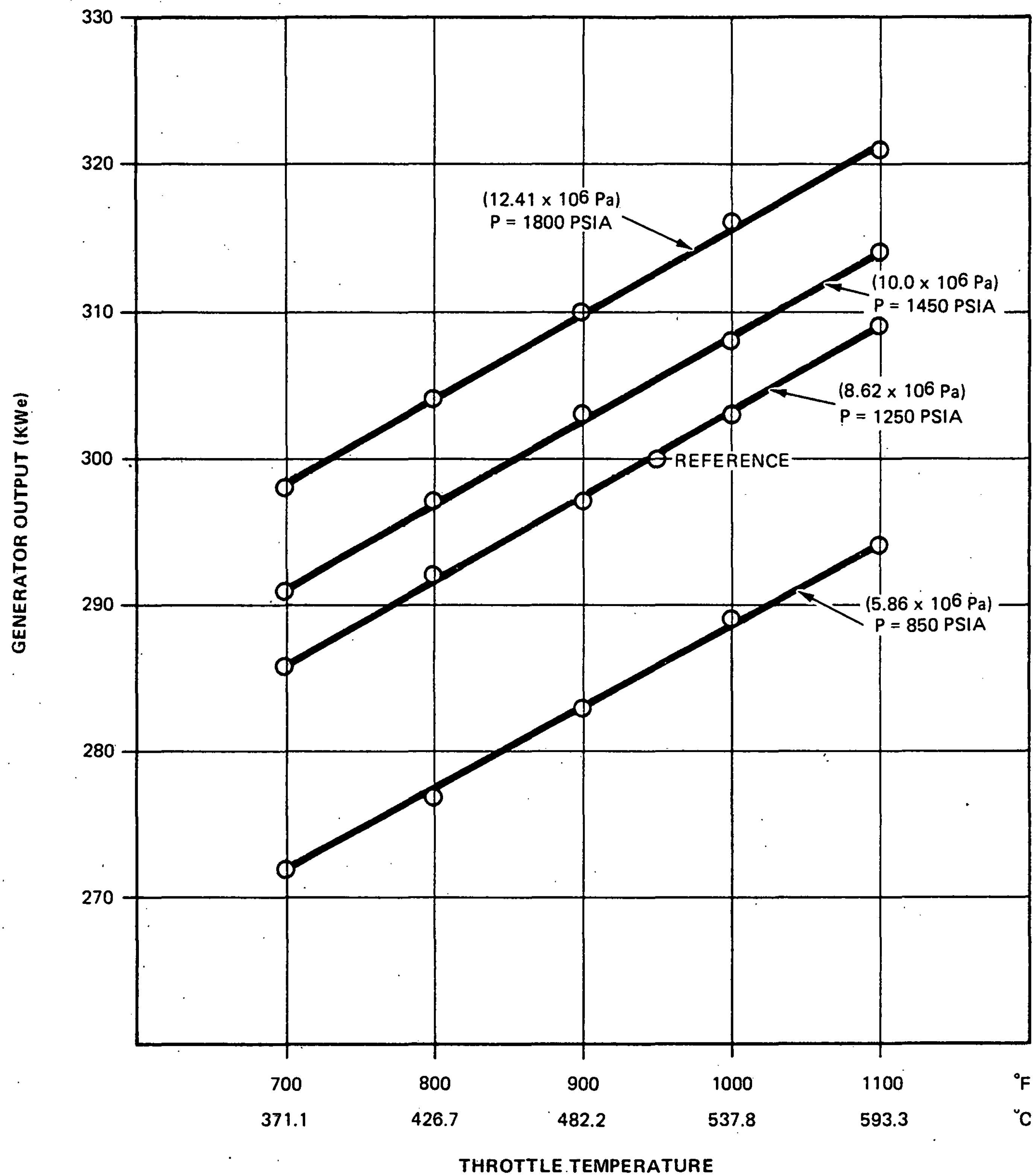

Figure 11–5. ELEMENT 6B - STEAM RANKINE CYCLE, GENERATOR OUTPUT VS. THROTTLE TEMPERATURE

SECTION 12
APPENDIX

CALCULATION PROCEDURE FOR STEAM RANKINE CYCLES

Several computer programs were used to generate the heat balances for the various steam Rankine cycles evaluated in this study. Where possible, existing in-house programs were used to reduce the total manhour requirement. With cycles that could not be evaluated using existing programs, additional programs were generated to meet the requirements of the study. The input values, basic calculation procedure, and output results of all programs were similar with only minor changes in the physical configuration of the cycle.

The input generally consisted of desired throttle conditions, turbine efficiency, condenser pressure, performance criteria for the feedwater heaters, final feedwater temperature, pump efficiency, system mechanical and electrical losses, and required electrical output or specified throttle flow. Additional input would depend on the specific cycle.

The calculation procedure usually began with an input or assumed throttle flow value and calculated a value for the power generated. If a specific generator output was required the value of the throttle flow was modified and the required output value was obtained using a convergence technique. The procedure used to arrive at the power output for a given throttle flow was as follows:

1. A final feed flow was calculated from the throttle flow and any other boiler flows.

2. From the performance characteristics of the top heater the saturation pressure in the heater shell was determined. Using a specified pressure loss for the extraction piping, the pressure at the turbine extraction port was found. Knowing the shape and orientation of the turbine expansion line, based on the turbine efficiency, an enthalpy for the steam at that extraction point was calculated. Finally, knowing the feedwater flow, the feedwater heater performance, and the enthalpy of the extraction steam, the flow of extraction steam was calculated.

3. The above procedure was repeated for each heater.

4. The turbine exhaust conditions were calculated from the turbine efficiency and specified exhaust losses.

5. The turbine shaft power produced was found by completing an energy balance of all flows into and out of the turbine.

The output of the programs included a restatement of input data, state conditions and flows for all major components, power generated, and heat rate or thermal efficiency values.

A sample of the output of one in-house program is included on the following page.

SERI - CYCLE 6A - BASE CASE - 1250 PSIA, 950 F

GENERATCR

GENERATOR OUTPUT = 10000. KW MECH. AND ELEC. LOSS = 150. KW

CYCLE PERFCRMANCE

HEAT RATE = (90243.(1468.6-376.4)/ 10000.) = 9856.2 BTU/KWH
BOILER DUTY = 98563376. BTU/HR BOILER BLOWDOWN = 0.0 PERCENT
CONDENSER DUTY = 64461664. BTU/HR CONDENSER PRESSURE = 2.5 IN HGA

	FLOW LB/HR	PRESS. PSIA	TEMP. DEG F	ENTHALPY BTU/LB	QUALITY
TURBINE THROTTLE	90243.	1250.00	950.00	1468.63	
TURBINE EXHAUST	67313.	1.23	106.70	1018.57	0.91
MAKEUP TO CONDENSER	0.				
HOTWELL	72382.	1.23	108.70	76.69	

TURBINE EFFICIENCY = 80.00 PERCENT
TURBINE FIRST EXTRACTION PRESSURE RATIO = 4.75
BOILER FEEDPUMP EFFICIENCY = 75.00 PERCENT
BOILER FEEDPUMP POWER REQUIREMENT = 155. KW / 208. HP

HEATER NO.	1	2	3	4
TTD	0.	0.	0.	5.
DC	10.	10.	0.	10.
STAGE PRESS	263.04	106.51	33.36	9.07
LINE PRESS LOSS	15.78	6.39	2.00	0.54
EXT FLOW	6650.	5949.	5262.	5069.
EXT ENTHALPY	1334.79	1266.38	1189.00	1115.26
HTR SHL PRESS	247.26	100.12	31.36	8.53
HTR SHL TEMP	400.00	327.91	252.88	185.79
HTR SHL ENTHALPY	375.10	298.63	221.52	153.81
FW IN FLOW	90243.	90243.	72382.	72382.
FW IN PRESS	1408.89	1428.89	41.36	61.36
FW IN TEMP	327.91	255.82	180.79	108.70
FW IN ENTHALPY	300.87	227.38	148.87	76.84
FW OUT FLOW	90243.	90243.	90243.	72382.
FW OUT PRESS	1388.89	1408.89	31.36	41.36
FW OUT TEMP	400.00	327.91	252.88	180.79
FW OUT ENTHALPY	376.44	300.87	221.52	148.87
DR IN FLOW	0.	6650.	12599.	0.
DR IN TEMP	0.0	337.91	265.82	0.0
DR IN ENTHALPY	0.0	309.29	234.81	0.0
DR OUT FLOW	6650.	12599.	0.	5069.
DR OUT TEMP	337.91	265.82	0.0	118.70
DR OUT ENTHALPY	309.29	234.81	0.0	86.68

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15. Supplementary Notes			

16. Abstract (Limit: 200 words)

Methodologies for the analysis of value and comparing thermal storage concepts are presented in this report. Value is a measure of worth and is determined by the cost of conventional fuel systems. Value data for thermal storage in large solar thermal electric power applications are presented. Thermal storage concepts must be compared when all are performing the same mission. A method for doing that analysis, called the ranking index, is derived. Necessary data to use the methodology are included.

17. Document Analysis

a. Descriptors Solar Thermal Power Systems ; Energy Storage ; Heat Storage ; Comparative Evaluations ; Cost ; Selection ; Methodology ; Design ; Data

b. Identifiers/Open-Ended Terms

c. UC Categories

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