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SOLAR THERMAL ELECTRIC POWER SYSTEMS

Progress Report
for the period
1 May 1973 to 30 June 1973

Prepared by

Colorado State University
Fort Collins, Colorado

and

Westinghouse Electric Corporation
Georesearch Laboratory
Boulder, Colorado

Prepared for

The National Science Foundation
Research Applied to National Needs
Washington, D. C.

Grant GI-37815

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COLORADO STATE UNIVERSITY
Solar Energy Applications Laboratory
Engineering Research Center

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PREFACE

Colorado State University, in association with the Westinghouse Electric Corporation, have undertaken a program of research in the generation of electric power from solar energy. The objective of the research is a rigorous appraisal of methods for converting solar energy to electricity and a determination of promising method(s) for future development. There are many possible approaches for generating electrical power from solar energy; the CSU-Westinghouse team will concentrate on thermal-electrical systems.

A summary of the first two months of this 18-month program is presented in this report. During this period, organization and other technical meetings were held to establish common framework and base for the CSU-Westinghouse team. This progress report is presented in two parts; (1) the CSU effort and (2) the Westinghouse effort. Scheduled progress for the total of the grant period is indicated in the respective segments of the report.

SOLAR THERMAL ELECTRIC POWER SYSTEMS

CONTENTS

PART I

COLORADO STATE UNIVERSITY

1.0 INTRODUCTION

The first phase in this study of Solar Thermal Electric Power Systems (STEPS) consisted of data collection for materials and components which may be needed during subsequent phases of the study when a variety of possible systems are considered. The task was divided between the Colorado State University and Westinghouse teams; Colorado State University was responsible for data collection of previous and contemporary solar power systems, collectors and heat storage; Westinghouse was responsible for data collection on materials, heat engines, heat transport, heat transfer, and energy storage. No data collection effort, however intensive, can claim to have <u>all</u> relevant data preparatory to designing a wide variety of solar power systems. There will be need to make additional search for relevant information as this study progresses. Nevertheless, a substantial amount of information has been assembled; mostly from published literature and some from private correspondence, which is considered sufficient for the second and succeeding phases of the study to proceed.

The CSU study tasks for STEPS and scheduled progress are shown in Figure 1-1. A corresponding Westinghouse task schedule is included in the Westinghouse section of this report (refer to Figure 1-2 of the Westinghouse report).

A great number of theoretical studies of solar-thermal power generating systems have been published in the literature and some experimental results of performance characteristics of small systems are available. Solar-thermal systems operating small pumps for producing useful mechanical work have been built and are in operation today, but there is no system in existence which is generating useful power to a utility network. Thus, the available

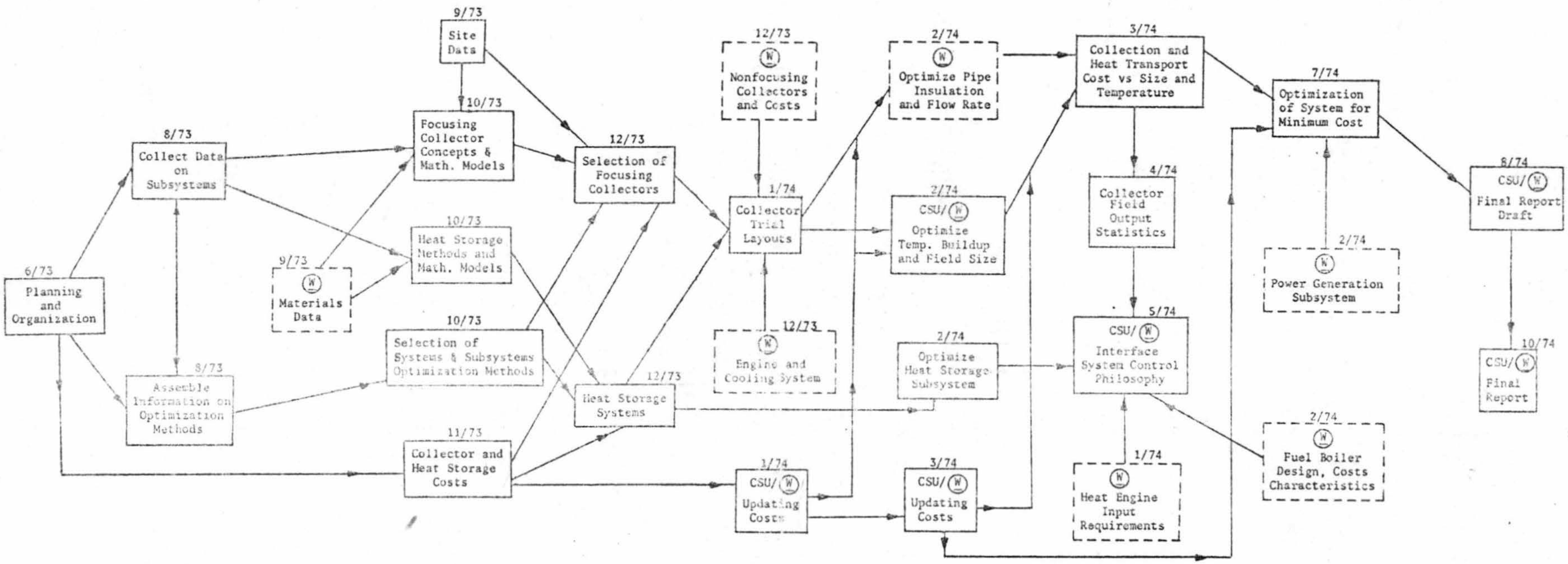

Figure 1-1 Flow Chart of CSU Effort on Solar Thermal Electric Power Systems Optimization Study
(Date on blocks show month of scheduled completion)

data on performance characteristics and costs of systems and components are either theoretical or applies to small systems. Discussion of types of collectors, and heat storage subsystems are included in this progress report.

Closely related to these tasks has been the establishment of a reference library on solar energy applications. The organization and cataloging of books, reports, journals, papers, proceedings, technical correspondence, manufacturer's literature, and other references has proceeded to the stage of a useful and comprehensive collection of most of the world literature in this field. Several hundred items are available, containing more than a thousand technical articles. Assembly of this library is still in progress and is expected to be completed by the end of the next quarterly report period.

Solar equipment cost data are scarce, and the limited information is not usually applicable to sizable production levels. But approximately 25 references have been obtained, of varying quality and utility. These range from reported costs of complete solar power plants to estimates and quotations on the major components in volume manufacture.

Cost data and estimates so far procured include (a) complete manufactured systems, primarily solar water heaters, (b) house heating system estimates, (c) prices of materials in quantity, such as glass, heat exchanger plates, storage tanks, etc., (d) actual costs of experimental solar equipment, (e) projected costs of solar power systems. These figures are of highly variable quality and utility, but used judiciously, they will provide values with which other data (to be obtained) can be compared.

Additional cost data are being collected. Discussion of cost estimate and cost analysis methodology are included in sections 5 and 6 of this report.

2.0 SOLAR COLLECTORS

2.1 General

Solar energy collectors may be grouped into two categories; (1) flat plate or nonfocusing collectors and (2) concentrating or focusing collectors. Because the sun's radiation is not ordinarily intense enough to obtain high temperatures of the working fluid, effort is made to concentrate radiation from a large area onto a small heat absorber. The focusing collectors thus achieve higher temperatures than nonfocusing collectors, but require direct sunlight and usually some mechanical means to track the sun. Nonfocusing collectors on the other hand, are not usually equipped with tracking devices and can make use of both direct and diffuse sunlight. However, because a horizontal surface receives less solar radiation than a surface which is normal to the direction of radiation from the sun, the flat plate, nonfocusing collectors are usually provided with some adjustability for tilt which may be varied seasonally.

Principal efforts related to solar collectors of the concentrating type have been the identification of possible candidate systems and the procurement and cataloging of information on their design and performance. This report summarizes progress in these areas.

2.2 Classification of Concentrating Solar Collectors

The examination of focusing solar collectors for potential use in electric power generation systems may follow a number of paths. A catalog of principal types, based on theoretical concepts, published designs, and experimental units actually built and tested may first be prepared. The list provided in Table 2-1 is the output of one such effort in which all of these sources were used and combined.

TABLE 2-1

TYPES OF FOCUSING COLLECTORS
(Concentrators)

A. Multiple Flat Reflectors

   1. Stationary
   2. Individually sun following; one axis
   3. Individually sun following; two axes
   4. Array, sun following, one axis
   5. Array, sun following, two axes

B. Single CurvatureReflectors

   1. Parabolic cylinder

      a. North-South sun following, one axis
         (1) horizontal
         (2) fixed tilt
         (3) adjustable tile
         (4) segmented reflectors
      b. East-West sun following, one axis, horizontal
         (1) horizontal
         (2) segmented reflectors
      c. Normal position, sun following, two axes

   2. Circular cylinder

      a. North-South sun following, one axis
         (1) horizontal
         (2) fixed tilt
         (3) adjustable tilt
      b. East-West sun following, one axis, horizontal
      c. East-West sun following, periodic adjustment
      d. Normal, sun following, two axes

C. Double Curvature Reflectors

   1. Paraboloids, two axes
   2. Spheres, two axes
   3. Catenaries of revolution, two axes
   4. Arrays focused on one receiver, two axes
   5. Arrays focused on individual receivers, two axes

D. Other Shapes and Composite Reflectors

   1. Fresnel reflectors, cylindrical
   2. Fresnel reflectors, circular
   3. Fresnel reflectors, strips
   4. Truncated cones
   5. Sun following flat reflector and fixed paraboloid reflector
   6. Multiple sun following flat reflectors and large fixed paraboloid

E. Lenses

   1. Conventional cylindrical
   2. Conventional circular
   3. Fresnel cylindrical
   4. Fresnel circular
   5. Fresnel strips

Several other arrangements might be used in classification of types. Degree of concentration, type of tracking, reflecting surface material, or other criterion could be selected. Selection based on general shape of concentrator is used here with orientation system as sub-categories.

The list in Table 2-1 is believed to contain representatives of all types of concentrating solar collectors previously suggested or tested and some additional types known to be possible. A few others, especially of multiple and composite types, could be included, but their characteristics are not sufficiently different to justify extending the list.

2.3 Examples of Principal Collector Types

A. Multiple Flat Reflectors

1. *Multiple Flat Concentric Reflectors.* This system involves individually tracked reflectors which are tracked on 2 axes and involve a stationary or rotating central boiler on a tower. The type is represented by a USSR design which is discussed by Baum, et.al. (1). This type of reflector system is also the subject of the University of Houston/ McDonnell Douglas study (2).

2. *Multiple Flat Reflectors in Fixed Horizontal Cylindrical Contour.* This type of collector system involves an adjustable receiver position to maximize concentration of reflected energy. Representative concepts are illustrated by the USSR design (3) and suggestion by the Gulf General Atomic Company (4).

3. *Array of Flat Mirrors.* This system with a receiver on a fully tracking frame is represented by the USSR unit (concentrator for photovoltaic cells) (5) and University of Wisconsin disk-shaped mosaic mirror assembly (6).

B. Single Curvature Reflectors

1. *Horizontal Parabolic Cylinder With N-S Orientation.* The Shuman-Boys solar power plant (7) at Meadi Egypt (1913) is an historically important system. This system consisted of parabolic cylinder collectors 205 feet long with a total of 13,000 square feet of solar interception area. Steam was generated and piped to a 100-Hp engine to provide irrigation from the Nile.

2. *Fixed Tilt Parabolic Cylinder.* An experimental system with a N-S orientation, with one axis tracking, was constructed by the University of Wisconsin and tested in Denver by Löf, Fester and Duffie (8). Experimental data for concentration ratios of 10 to 22.5 and surface temperatures from 88 to 353 degrees F were obtained.

3. *Horizontal Parabolic Cylinder With E-W Orientation.* A recent report by the University of Minnesota/Honeywell (9) involves a study of this type of system. Together with heat pipes and heat storage, each modular unit of the system is calculated to produce 2 Kw of electrical power. The Aerospace Corporation has also studied a similar system to provide base power at an average of approximately 1000 Kw (10).

4. *Segmented Parabolic Cylinder, Horizontal, E-W Orientation-One Axis Tracking.* An experimental unit has been constructed at the University of Provence, Marseille, France. The unit consists of segments of plane mirrors arranged to form a parabolic cylinder. The chief argument for considering this type of reflector surface is that a continuous parabolic surface is difficult to construct. Some experimental results from a segment of such a collector is reported by Sakr and Helwa (11).

5. *Horizontal Circular Cylinder, E-W Orientation.* A focusing collector consisting of an inflated plastic film cylinder 12 meters long and 1.5 meters in diameter was tested by the Israel National Physical Laboratory (12). This particular unit is representative of this type.

6. *Assymetric Parabolic Cylinder, N-S Orientation, One Axis Tracking.* This type is represented by the Israel National Physical Laboratory unit in Beersheba. The optics of the system is discussed by Tabor (13).

C. Double Curvature Reflectors

1. *Paraboloids of Various Sizes, Shapes, Materials.* Numerous units have been constructed and operated in the USA, USSR, France, Japan, Israel and elsewhere. This particular type has been studied extensively and operating characteristics are available in the literature, (14) (15) (16) (17) (18) (19) (20) (21).

2. *Spheres, Two-Axis Tracking*. A segmented spherical reflector of approximately 12 ft diameter was tested at the University of Wisconsin (6).

3. *Catenary of Revolution, Two-Axis Tracking*. A plastic film reflector of this type has operated in Tashkent, USSR (5).

4. *Array of Long-Focus Small Paraboloids*. This unit which individually tracks the sun on two axes reflects to a single receiver. Such units have been constructed in Italy (22).

D. Composite Reflectors

1. *Single Flat Mirror (or of smaller segments), Two-Axis Tracking, Reflecting to Fixed Paraboloid (or of smaller segments) and to Receiver*. A solar furnace of this type is in operation at Montlouis, France (23) and small solar furnaces are at Odeillo, France.

2. *Multiple Flat Mirrors (each segmented), Each Tracked on Two Axes, Reflecting to Fixed Paraboloid (segmented) and to Receiver*. A 1000 Kw solar furnace of this type has been constructed at Odeillo, France (24).

E. Lenses -- Fresnel lenses of circular or strip form are capable of concentrating solar heat on a focus or linear target. Solar furnaces using circular modules have been constructed (25). Performance of strip concentrators have also been reported (26).

2.4 Plans for Analysis of Concentrating Collectors

The considerable number of concentrating solar collectors that have been built, and the much larger number that could be designed, make it necessary to devise a strategy for evaluation which can reduce the number of alternatives without jeopardizing the objectives of the study. Toward this end, it now appears that a de-coupling of components of the concentrating collector and the separate analysis of each component will be highly advantageous.

A concentrating solar collector comprises two main elements and two or more subordinate components. The primary elements are a concentrator (reflector or lens type) and an absorber heat exchanger. Subordinate parts are the supporting structure, and the tracking mechanism. For purpose of performance analysis and cost estimation, these components can all be examined separately.

The principal advantage in this de-coupling is the separation of concentrator and absorber. Mathematical models of the concentrator and estimates of cost can be made independent of the design and performance of the absorber. Only the optical properties of the concentrator need be involved, and once determined, can be used with any appropriate absorber.

Similarly, the absorber-heat exchanger can be separately modeled, the energy input being a suitable representation of a solar radiation rate delivered over a chosen absorber area. A non-concentrating (flat-plate) collector model can be applied to this absorber, simply by using the higher radiation rates characteristic of concentrator outputs. Suitable consideration of "spilled" radiation beyond the absorber area, energy variability across the focal zone, and differences in absorptivity of radiation received at various angles of course need to be considered. But once a particular type of absorber-heat exchanger has been mathematically described, its performance in a concentrating system with various types of concentrators, or with no concentrator at all, can be determined. There is essentially no feed-back to the concentrator from the absorber.

To some degree, at least, supporting structures and solar tracking and guidance equipment can be similarly separated. Interchangeability between various types of concentrators can be expected of these units, so design and cost can be considered apart from specific collector concepts. The size, design temperature, and operating efficiency of concentrator-absorber assembly are factors which will set the requirements for stability and tracking precision in the support and guidance components.

The schematic diagram depicts these concepts and shows in a general way the mutual dependencies as well as the separability of components. The simulation study on a non-concentrating collector by Löf and Tybout followed this sort of pattern. By such procedure, it is believed that a concentrator can be optimized to deliver, for example, 5 Kw thermal in a 6-inch circle, with 5 percent spillage beyond that circle, under a particular set of conditions. Similarly, the absorber-heat exchanger for delivering least cost heat at 250^{o}C from the 6-inch diameter 5 Kw radiation supply can be designed. Further details of analytical methods and procedures will be established during the next report interval.

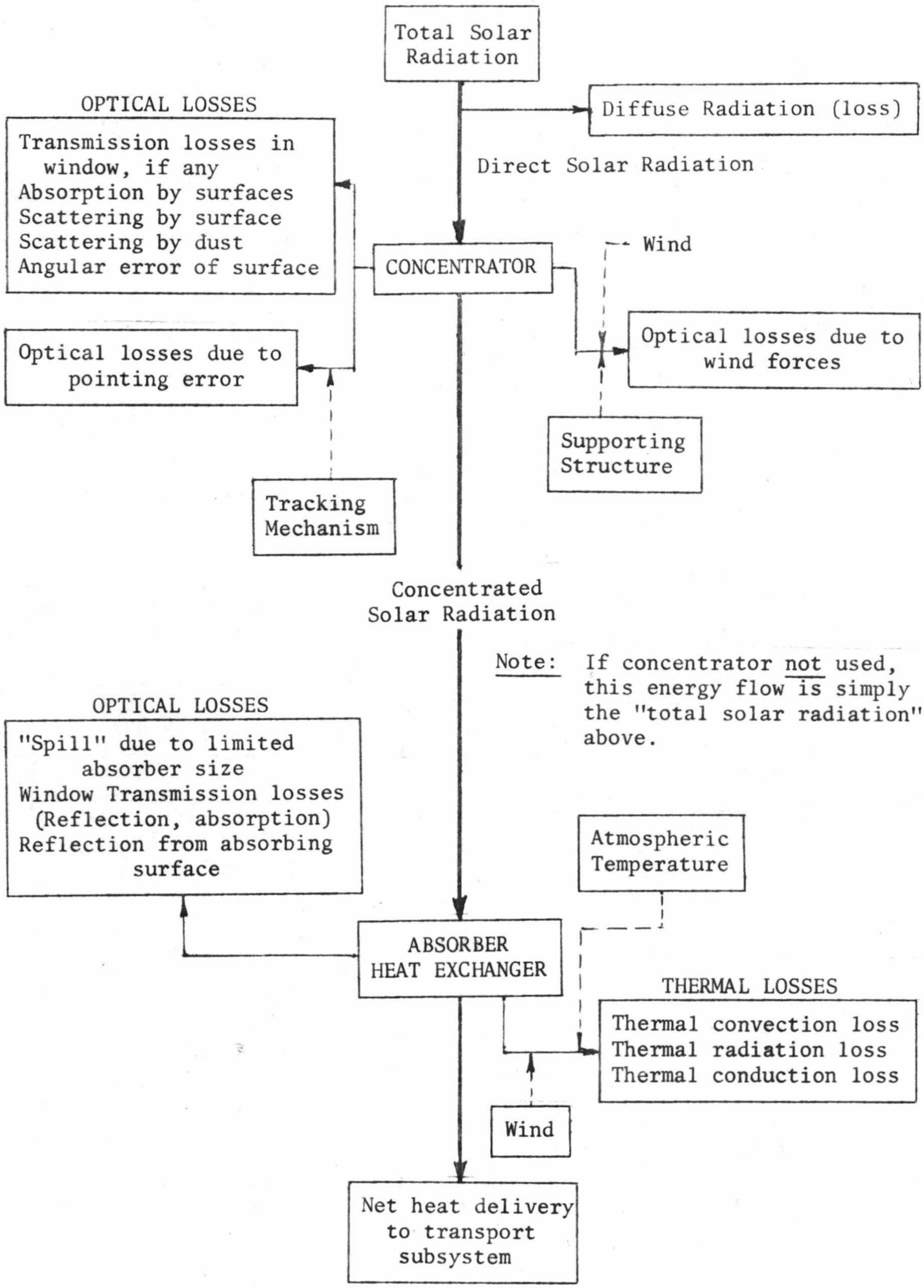

Figure 2-1 Schematic Representation
Energy Flows in Concentrating Solar Collectors
(Transient Losses Excluded)

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3.0 HEAT STORAGE

3.1 Introduction

The need for some kind of heat storage in solar thermal power generation systems is reasonably well established. This point is easily appreciated by consideration of the uncontrollably interruptible nature of sunlight due to clouds. Because a utility company's production is almost entirely consumer controlled, a solar plant with no heat storage probably would not fit easily or profitably into the production capability of the utility. Temporary storage immediately becomes important, then, as a means of smoothing out the minute to minute fluctuations of insolation and also as a means of providing needed time to start up alternate power sources when solar power is going to become unavailable for longer periods.

Even with the basic question of the need for heat storage resolved, still the matter of what type of storage to develop and use remains. The complexity of the material data collection task depends heavily upon the type or types of storage required for a plant. The question of how a solar thermal plant might fit into a utility company's needs must be considered before an intelligent beginning to the data collection work can be made. Two possible schemes, defining a wide spectrum of approaches to the development of solar thermal power, emerge. These are (a) base power plants and (b) peak power plants. Other than differing design optimization criteria, the only major difference between these types is heat storage.

The base power plant concept considered by Aerospace Corp.,(1) is a facility which collects solar energy during insolation, storing some as sensible heat and using the remainder for conversion to electrical energy for immediate use. The storage is designed for a continuous output of some base power load for 18 hours after interruption of sunlight due to either nightfall or extended cloudy periods. The storage facility of such a plant clearly will provide the short-term smoothing function essential in all solar plants, but goes much further in providing enough storage to get through the night time hours after a day of essentially full sun.

The other concept, somewhat less easily defined, is a peak power system, designed to provide power during peak usage periods. This system requires storage for periods of the order of one hour. The power generated by this sytem could supply the added load caused by air conditioning and refrigeration needs which are tied fairly closely to hot sunny weather.

As opposed to the solar thermal base power plant, the solar thermal peaking plant would represent a truly radical departure in the economic sense from the equipment used conventionally to generate peak power. Typically, equipment with low capital investment but high maintenance and fuel costs is used for peak power, whereas the solar plant features high initial capital cost and zero fuel costs.*

The storage capacity question represents an important degree of design complexity in the optimization problem facing the CSU-Westinghouse research team. The immediate problem with respect to heat storage is to recognize that one hour storage may require systems and materials differing substantially from those used for 18-hour storage. Thus, the data collection task must be conducted with as general an approach as possible.

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* Maintenance costs for solar plants are largely unknown at this point.

3.2 Data Availability

The possible use of solar energy in home heating and cooling applications has already prompted serious consideration of heat storage systems and materials. Winter residential heating requirements make storage essential (2). Thus, some fairly complete data on materials have been collected, and some laboratory and field testing of these systems have been performed (3) (4). Unfortunately, the temperature ranges of the storage materials for solar home heating are below the commonly accepted limits for solar thermal power generation systems.

3.3 Data Requirements

The data collection task on materials is necessarily focused on a specialized set of physico-chemical properties. The properties identified as being of principal importance are listed here.

A. Heat storage per volume and per mass

B. Mass density

C. Temperature (or temperature range)

D. Thermal conductivity

E. Volume expansion coefficient

F. Corrosive properties and chemical compatibility

Engineering considerations add other sets of data to the above list. At this point, however, materials cost is a needed essential item.

G. Storage material cost per mass

3.4 Data Collected

While a large number of different conceptual designs for heat storage in solar energy systems has been proposed, very few concepts have been subjected to feasibility studies. Thus, available information is often limited to partially complete physico-chemical data. The major types of storage systems have been identified, and data on a number of materials useful in these systems have been collected. The Westinghouse effort in data collection for heat transfer materials has produced a thick loose-leaf binder containing a large amount of data on both heat transfer and storage materials. We (CSU and Westinghouse) have independently discovered no common format for the data tables found, and so we have taken the approach of placing xerox copies of all relevant data tables in the notebook. Reconciling and integrating all the data found by CSU and Westinghouse remains to be done. The information available on many heat storage materials is incomplete, and continued effort at some level throughout the rest of the grant period will be required to fill in the gaps.

3.5 Heat Storage Systems

Heat storage systems relevant to solar thermal power generation has been studied. These types are briefly reviewed here to provide a background to the heat storage issue. Some basic information is provided for each system. The systems submit to classification as either (a) sensible heat storage, or (b) phase change heat storage.

A. Sensible Heat -- Systems storing sensible heat seem generally to be simpler in design. Liquid sensible heat storage materials offer the added advantage that the heat storage material can be used as the heat transfer medium. Because most of the technology already exists, the costing exercise is more straightforward.

Water is the best known liquid sensible heat storage material. It has high specific heat, good thermal conductivity, excellent handling properties, and it is very cheap. On the negative side, it freezes at $32^{o}F$ and boils at $212^{o}F$ requiring pressurization. A relevant technology exists; specifically an existing conventional fossil fuel plant in Germany (1a) features a sensible heat storage unit using water to accommodate load fluctuations.*

Organic fluids can be considered for sensible heat storage. These fluids are relatively cheap, they are not corrosive, and they generally have higher atmospheric boiling points than water. Yet their specific heats and thermal conductivities are not as favorable as water. Again, all the elements of a heat storage technology using organic liquid exist now.

Iron and rock are examples of solid materials presently in use as heat storage media. Such solids have apparently not been considered for application in steam cycle systems.

B. Phase Change -- All phase change phenomena involve a latent heat quantity either given up or used when the transformation occurs. Such heat storage schemes feature the useful advantage of zero temperature change as heat is stored or given up.

1. *Heat of Fusion.* All materials release a latent heat of fusion upon making a liquid to solid transition, and this heat can be used for heat storage. Three principle types of fusion materials are,pure substances, eutectics, and incongruently melting substances. High heat contents per

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* Note the interesting analogy to load fluctuations to insolation fluctuations.

mass can be achieved with very small temperature changes. Also, at least the eutectics show a considerable range of fusion temperatures. Unfortunately, heat exchanger problems, temperature dependent volume change, material degradation with time, and material cost are significant disadvantages. The existing technology for these systems is very limited.

2. *Heat of Vaporization.* The latent heat of vaporization may be exploited as a heat storage mechanism. A distillation-condensation system concept has been considered, and it appears to have a relatively high storage capacity. Disadvantages include the requirement of two containment vessels (with attendant increased system complexity). Water is an excellent choice of working fluid, but little or no technology exists for such a system.

3. *Others.* Solid-solid transitions and heats of solution are mechanisms of heat storage which have received some attention. The previously mentioned heat of fusion concept appears to offer better promise than either of these. Further, these two concepts suffer from an absolute lack of relevant existing engineering experience.

C. Hybrid Systems -- Such a system might use both latent heat of fusion and sensible heat as storage mechanisms. The resulting system has most of the advantages and disadvantages of both. Eutectics are materials which might be appropriate for such systems (e.g., LiOH-LiF).

3.6 Phase Two Plan of Action

With the heat storage data collection task well underway, effort must now be directed toward the next phase of work, namely the conceptual design of heat storage subsystems. The initial task will now be to consider in some detail heat storage systems showing some promise. Because most information is presently available on sensible heat systems, these will be considered first. The fluids for these systems will be water and various organic liquids. The concept of a combined heat transfer/heat storage liquid function will be considered, as well as various conventional multi-component heat exchangers. The initial goal will be to obtain preliminary cost figures per Btu of storage per unit time at various storage temperatures. In this way, no limit on generality with respect to the specific storage function (short-term or long-term) will be imposed. Likewise, for generality, only very broad limits on storage temperature will be imposed (viz., $0^\circ C < T < 700^\circ C$). It is hoped that general storage cost guidelines will emerge from this effort.

For heat storage concepts which are not yet tested or proven out, effort will be made commensurate with the available knowledge. If possible, some future research priority listing of these more exotic schemes will be made, based upon the extent to which they show engineering promise.

3.7 Closure

The previous discussion is intended to outline the scope and complexity of the heat storage problem in solar thermal power generation. The need for some type of storage has been argued, independent of solar energy conversion strategy. The need for physico-chemical materials data and also engineering data on various storage schemes has been covered, and the important heat storage schemes have been reviewed briefly. Finally, the plan of action for the next phase of the heat storage work has been presented.

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4.0 SYSTEMS OPTIMIZATION METHODS

The major purpose of the work on this task has been to collect background information on optimization methods that can be used for the STEPS study. The information collected in this phase will be used in Phase II, Conceptual Development, to optimally combine subsystems of STEPS on the basis of minimum electric energy costs.

It will be necessary later to match subsystem problem characteristics to the appropriate optimization techniques. This generates a requirement to examine the relevant system and subsystem characteristics in a "cost optimization" context.

4.1 Requirements Imposed by STEPS on the Choice of Optimization Techniques

A. From the point of view of optimization theory the STEPS problem is a large scale systems problem. This means that the system optimization problem is too large to be solved in one step by available techniques. This is due to the large number of feasible combinations of subsystems and to the types of non-linearities that occur. Two alternatives are available in this case, (1) using a suboptimal procedure or (2) finding a way to decompose the system, do some of the optimization on the parts and then recompose it doing the remaining optimization so that the optimal solution to the original problem is found. The major difficulty in the second alternative is in finding a way to decompose the problem so that it can actually be recomposed into the optimal solution to the original problem. Examination of the STEPS problem characteristics will determine the feasibility of this approach. (One procedure in the suboptimal category is to follow a decomposition - recomposition procedure while only achieving a suboptimal solution to the original problem.)

B. The systems optimization problem and most of the subsystems optimization problems involve non-linear relationships. For example, many system performance characteristics enter into the systems equations as products.

C. The uncertainty in much of the cost data and in potential improvements in STEPS technological state-of-the-art require optimization approaches with good sensitivity analysis capabilities.

D. The STEPS literature shows that a variety of modeling procedures are used in the case of each of the STEPS subsystems. There are incompatibilities imposed by the modeling procedure on potential optimization approaches and vice-versa.

E. The stochastic nature of solar insolation may limit some of the optimization to approaches having good stochastic capabilities or to simulation approaches.

F. There are existing computer programs of system and subsystem models from previous or concurrent studies. Their use in this study may be preferable to developing some of our own computer programs. The model on which the computer program is based may limit the choice of optimization techniques that can be used. Some programs that will be considered are those used by the Solar Energy Laboratory at the University of Wisconsin and those developed by the University of Minnesota-Honeywell team.

G. The data available will dictate to some extent the optimization procedures used.

H. A mixture of 0-1 integer and continuous variables occur in the STEPS and most subsystem problems.

4.2 Characteristics of Possible Optimization Approaches to the STEP System and Subsystem Problems

It is likely, due to the large scale aspects of the STEPS problem and the different nature of the subsystems that a number of the following optimization techniques will be used in the subsequent phases of this study. It is also possible that more than one approach may be used for each subsystem problem or for the system problem.

A. Linear Programming -- The applicability of this widely used technique may be quite limited in the STEPS context since linear constraints and objective functions are required. However, some non-linear problems may be solved by linear programming. Through separable programming, a linear programming approximation procedure can be developed for many non-linear problems. Generalized programming is a non-linear programming technique that uses a sequence of linear programming problems to effectively solve some non-linear programming problems. Also adaptations of problems into the linear programming format such as fractional programming, where the objective function is a ratio of linear forms, and quadratic programming, where the objective function is quadratic, may be useful. The major desirable features of linear programming are its capability for solving large problems and its extensive sensitivity analysis properties. Unfortunately, linear programming is quite limited in its capability to handle stochastic problems.

B. 0-1 Integer Programming -- 0-1 and mixed integer problems have been solved by a variety of methods. The introduction of 0-1 variables drastically reduces the size of problem that can be solved. One of the most successful approaches from this point of view has been Geoffrion's strongest surrogate constraint modification to the Balas implicit enumeration algorithm. A 100 variable problem can be solved efficiently with this method. Major disadvantages are that sensitivity analysis capabilities are virtually non-existent and that problem relationships must be linear. In theory, the basic Balas approach may be extended to non-linear objective functions. In doing this the problem size that can be solved is further drastically reduced.

C. Dynamic Programming -- An important feature of this technique is that its basic assumptions are different from most other optimization approaches. These are (sufficient conditions) the separability of the problem into parts and a monotonicity relationship among the parts -- assumptions that are satisfied in a wide variety of cases. The assumptions do not require linearity, convexity, differentiability, continuity and so forth. Thus, dynamic programming can often be used where no other technique is applicable. The separability requirement matches with that of large scale systems so that dynamic programming is often used as the basis for optimal recomposition in these problems. Though the basic principles of dynamic programming are straight-forward, its application in a specific context often requires considerable ingenuity in problem formulation to generate an efficient computational scheme. There is no single dynamic programming computational algorithm that one may "plug" the data into as in linear or non-linear programming. Each of the many algorithms in inventory theory, resource allocation, control theory, and so forth is unique to a particular set of conditions and the algorithms themselves take many widely varying forms. Dynamic programming lends itself to problems where there are stochastic considerations, non-linearities, discontinuities, tabular data, integer variables or sequential processes. In addition, quite good sensitivity analysis properties are usually present.

D. Non-Linear Programming -- Non linear programming refers to a number of techniques that are based on interative procedures. These techniques operate by means of choices of direction and step size at each interation. One requirement for these techniques is that the partial derivatives of the objective function be unique, finite and continuous. In this approach reaching a global maximum is only guaranteed under certain restrictive conditions, such as the convexity of the objective function or quasi-convexity of the objective function with no inflection points. These conditions are not always present. In this case the application of the procedure will often lead to a local maximum (where the partial derivatives are identically zero). Some of the specific techniques are the basic gradient method, the Newton-Raphson or modified gradient method, subrelaxation, penalty function methods, linear programming aided methods and the gradient projection method. The capabilities of these methods to handle stochastic conditions and their sensitivity analysis properties are quite limited.

E. Queueing and Inventory Theory -- These techniques (Queueing is predominantly a technique of analysis rather than optimization), have been used in the analysis of reservoirs in hydrology studies. The STEPS inputs to a heat storage device from a solar collector are at various intensities and intermittent as in the case of a reservoir so that the analogy and therefore these techniques may be useful in the STEPS study.

F. Search Techniques -- Search procedures are usually sub-optimal particularly in the multivariable case. They are particularly valuable when no other optimization technique will suffice. They suffer much the same drawbacks as non-linear programming, such as being drawn to local optima and lack of sensitivity analysis capabilities, although they can be used when the non-linear programming assumptions do not hold. These procedures are often used in conjunction with simulation to generate sequences of run parameters.

G. Simulation -- Computer simulation of a system is an extremely powerful and versatile technique. When a system is poorly structured or very complex, analytical optimization techniques may not be applicable and simulation may be the only recourse. Also the stochastic characteristics of the system may preclude analytical optimization techniques. Simulation is usually a sub-optimization approach because the number of feasible combinations is so large (or is infinite) that they cannot all be considered. Problems where simulation is resorted to often have this characteristic. Simulation frequently has the advantage that while a system can only be imperfectly modeled for optimization it can be quite accurately modeled for simulation. Thus, a simulation study of a system may often be run in parallel with an optimization study of the system. Simulation can also be used to verify the accuracy of an optimization study.

4.3 Systems and Subsystems Modeling Approaches

Discussions have been held with personnel at the Solar Energy Laboratory at the University of Wisconsin and with personnel from the University of Minnesota/Honeywell, NSF Solar Power System project team. Some of the system and subsystem modeling approaches used by these teams will be considered for use in the STEPS project. In addition to these two approaches, summarized below, various other approaches will be considered.

A. The major component of the Minnesota/Honeywell collector modeling approach is the Monte Carlo ray trace simulation. Some of the other components of this approach are a tracking program, a collector tilt program, a shadow effects program and a heat transfer program. Preliminary estimates indicate that the adaptation of these programs from the present parabolic cylinder-circular heat pipe to other types of collectors would be possible with the expenditure of a reasonable amount of effort on the part of the Minnesota/Honeywell team.

B. A modeling technique used by the University of Wisconsin Solar Energy Laboratory is based on the numerical solution to equations for a system of the form

$$mC_p \frac{dT}{dt} = Q_{conduction} + Q_{convection} + Q_{radiation} + Q_{generated}.$$

The system is separated into a set of nodes where a node may be an entire component or a segment of a component. The heat transfer mechanisms from each node to every other node are identified and characterized. For an elementary approach to a problem inter-node conductances that are constants may be formulated so that the equations become

$$mC_p \frac{dT_i}{dt} = \sum_{j \neq i} C_{ji} (T_j - T_i) + Q_{i,\ generated}$$

For more refined analyses the C_{ji} are no longer constants but may be functions of other variables such as T_i and T_j.

A major advantage of this type of modeling approach is that it can be used to depict the wide range of possible STEP systems that must be considered in this study. Another advantage is that transient effects can be taken into account. A potential advantage of this approach that will be explored in Phase II of this study is its possible adaptability to stochastic analysis and the decomposition required for the large scale aspects of the STEPS problem.

4.4 Selected Optimization Bibliography

A. Texts

"Principles of Operations Research" by Harvey M. Wagner, Prentice Hall, 1969.

"Introduction to Operations Research" by Fredrick S. Hillier and Gerald J. Lieberman, Holden-Day, 1967.

"Integer Programming" by Harold Greenberg, Academic Press, 1971.

"Finite State Markovian Decision Processes", by Cyrus Derman, Academic Press, 1970.

"Linear Programming and Extensions" by George B. Dantzig, Princeton University Press, 1963.

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"Introduction to Linear and Non-Linear Programming" by David G. Luenberger, 1973.

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"Geometric Programming" by Richard Duffin, Elmor L. Peterson and Clarence Zener, Wiley and Sons, 1967.

"Introduction to Dynamic Programming" by George T. Nemhauser, Wiley and Sons, 1966.

"Dynamic Programming" by Richard Bellman, Princeton University Press, 1957.

"Applied Dynamic Programming" by Richard E. Bellman and Stuart E. Dreyfus, Princeton University Press, 1962.

"Dynamic Programming and Markov Processes" by Ronald A. Howard, M. I. T. Press, 1960.

"Optimum Seeking Methods" by Douglas Wilde, Prentice Hall, 1964.

B. Journals

Operations Research

Management Science

Naval Research Logistics Quarterly

SIAM Journal on Applied Mathematics

Econometrica

5.0 COST ESTIMATION

5.1 Objectives of Cost Estimating Task

The objectives of the cost estimating task group is to estimate the cost of manufacturing solar components. The cost of components will be used as one of the optimization parameters by the optimization task group.

5.2 Methodology

Major accomplishments during the period have been the development of a methodology for cost estimating of solar components and the gathering of basic cost data.

The methodology proposed to estimate costs of solar components is to break components into subparts and use cost estimating procedures developed by the Department of the Navy. The NAVDOCK estimating procedures were originally developed for maintenance operations at naval facilities. The NAVDOCK manuals are available from the Commerce Clearinghouse. The basis of the NACDOCK procedures are a series of man-hour estimating manuals organized by crafts such as sheet metal, welding, etc. The craft manuals are further broken down into a series of task descriptions. These task descriptions describe typical work assignments of that craft and the man-hour required for accomplishment. The tasks descriptions are organized on "spread sheets" in increasing order of time required to accomplish the task.

The estimator attempting to estimate the man-hours for a new task mentally compares the task he is evaluating to one of the detailed descriptions in the NAVDOCK manual for the craft involved. He has the opportunity to add or delete subtasks to the NAVDOCK task description to improve the comparability between the task being estimated and the NAVDOCK bench mark. The NAVDOCK manual identifies subtasks and the times associated with those subtasks and it is a relatively simple matter to add or delete subtasks as required. Standard subtasks such as material layout, machine set up, inspection, etc., are tabulated separately and may be added by the estimator when required.

The NAVDOCK manuals are used to estimate man-hour requirements and must be converted to dollar equivalents. The plan for the STEPS Project is to report both man-hour estimates and equivalent dollar costs, thus, as labor rates change, the appropriate adjustment could be made.

The cost estimating procedures outlined above are primarily applicable to first article production and do not reflect the economies of scale which would be possible with mass production. In order to estimate the man-hour requirements for multiple units production, the first article man-hour will be multiplied by appropriate learning curve values. Procedures for this transformation are well documented in such references as: Gallaher, Cost Estimating by Engineering Methods.

Cost estimates to manufacture any item including solar components are, in addition to manhours, the material costs and indirect manufacturing costs (overhead, burden). Added to the total cost of manufacturing are an allowance for general and administrative expense, contingencies and profit.

In addition to estimating solar component costs based on the NAVDOCK manuals and learning curves, it is anticipated that two or three alternate methods will be used for comparative purposes. One of the alternate methods are industry "rules of thumb" such as $1.50 per pound of fabricated metal products. Another method proposed for component cost estimating is to develop an array which would describe components similar to those in the STEPS program and the cost of those components as a function of measures of complexity, weight, accuracy, etc. A comparison would be made similar to that described in the NAVDOCK manuals whereby the estimator would compare the cost of the STEP component to costs of other known components, some similar such as flat mirrors, others more different such as tracking radar antennas.

5.3 Short Course

Pursuent to development of the cost methodology and gathering of basic cost data a Short Course on "Estimating Manufacturing Costs" was held at CSU on 14 and 15 June. The purposes of this short course conducted at no cost to the STEPS project were to become acquainted with cost estimators from metal fabricating companies in order to test out the concepts of cost estimating proposed for the STEPS Project, and to identify knowledgeable cost estimators in industry who would be cooperative in the review of cost estimates for the STEPS Project and who might release standard manufacturing time data.

The course was conducted by Dr. S. B. Thayer of the STEPS Project assisted by Mr. Ray Kincheloe, Chief Estimator, Fabrication Division, Collins Radio Corp., and Mr. M. Stanley Merrill, Industrial Engineer from International Engineering Company. The course attracted 20 cost estimators mostly from job shop, metal fabricating companies. Selected members of the CSU/Westinghouse cost estimating team participated in the course.

Data obtained during the short course which will be useful to the STEPS Project included percentage rates used to cover indirect manufacturing expenses, general and administrative expenses and expected profit margins. The concept of learning curves applied to first article production costs was confirmed. Also, data was obtained on average learning curve experiences.

5.4 Literature Review

Review of literature was also accomplished during the period. Particular attention was devoted to Solar reports which had some estimates of the cost of components.

6.0 COST ANALYSIS

The cost analysis requirements of the solar energy study comprises two parts: (1) cost estimation for the various components of alternative solar energy systems, and (2) the use of these data for comparing comprehensive costs of such alternatives. This portion of the report covers progress to date on the second part of the cost analysis. Progress on the first part is covered in section 5.0 of this report.

The economic analysis of alternative solar energy systems is still in the formative stages. W. Shaner, the CSU member assigned to this task, is scheduled to become more fully involved with this task in early August. In preparation for this work, the following approach involving six phases is tentatively planned.

6.1 Characteristics of Solar Energy as a Source of Electrical Power

Identify the unique features of potential solar energy supply to the electrical utilities. For example, what are the practicable temperature levels, insolation characteristics, reliability of supply, and possible attractive locations in the USA? This part of the investigation would be part of the literature search to acquaint the investigator with the state-of-the-art and as preparation for the second phase of this task.

6.2 Characteristics of the Electric Utility Industry

The electric utility industry would be studied to the extent necessary for understanding how it might make use of solar energy as an alternative source of power. Some areas of concentration would be: present and predicted load curves of selected utility companies and/or market areas; methods of operation whereby different electrical units are used to meet variable demand requirements; investment and operating costs of these facilities; standby facilities and requirements; and managements' attitudes towards the incorporation of solar energy sources into their systems. Published data and interviews with the staffs of utility companies will both be relied upon, as will discussions with the staff of Westinghouse and other relevant sources.

6.3 Ranking of Alternatives for Solar Energy Supply to the Utilities

Compare the characteristics of solar energy capabilities with electric utility requirements to identify the most promising possibilities open for solar energy use. Consideration would be given to the use of solar energy for peaking, intermediate and base load requirements, or any other such possibilities that might emerge during the investigation. Requirements for storage and standby capacity would be included in the above. Once the relative advantages of the different opportunities are identified, greater effort could be directed towards the more attractive possibilities.

6.4 Data Requirements for the Relevant Alternatives

The foregoing effort assumes that the type of data and form of analysis will, at least to some degree, be dependent upon the area of concentration (peaking, intermediate, or base-load power). Even with such concentration, broad opportunities exist for comparing different types of solar facilities (collectors, heat pipes, turbines, etc.). The types of data to be collected or developed include investment costs, operation costs and procedures, variances of costs according to the size of installations, service lives of various components, etc. (Progress in developing estimating procedures for this part are reported by the Cost

Estimating Task Group.) Broader based data are also needed on price trends for labor, materials, imported and domestic fuels, costs of financing, etc.

6.5 Adjustments in the Computer Model

The foregoing steps would be used to adapt the computer model, now being developed in more general terms, to the specific characteristics of the solar energy alternatives to be tested. By assuming that the development of solar energy will have negligible influence on the demand for electrical energy, the basis for the economic analyses becomes one of cost minimization, i.e., cost effectiveness. Thus, output from the model will probably be in terms of $/kwhr or $/kw for alternative solar systems and for the relevant conventional systems. These cost estimates will no doubt require adjustment for the differences in reliability of the alternatives. Adding in the costs of standby facilities is one way to adjust the basic cost data in a way that alternative systems provide energy at comparable levels of certainty. The computer model will also allow for comparisons of alternative systems using both current and projected costs.

6.6 Interpretation and Extension of the Results

By projecting future costs and changes in technology, some estimate can be made as to when (if ever) solar energy will be competitive, in the narrower business sense, with alternative sources of energy for electrical power production. Estimates can also be made of the conditions under which solar energy will be most competitive. For example, which areas of the country are, most favorable for its introduction and what percentage of the energy market might it capture? Should it become apparent that solar energy will not be competitive for many years to come, some preliminary judgments might be offered concerning the relative advantages of solar energy in terms of improved balance of payments for the country, lesser pollution of the environment, etc. To the extent that this subject is found to be important, some attempts at quantification might be made. For instance, the city of Los Angeles has given an indication of the value of pollution abatement when it requires the electric utility in that area to burn natural gas during the summer months instead of fuel oil, which was less expensive. The value to be attached to abatement is at least as great as the difference in the costs of fuel consumed during the summer.

PART II

WESTINGHOUSE ELECTRIC CORPORATION

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SOLAR-THERMAL ELECTRIC POWER SYSTEMS STUDY

Westinghouse Technical Progress Report

CONTENTS

SOLAR-THERMAL ELECTRIC POWER SYSTEMS STUDY

Westinghouse Technical Progress Report

ABSTRACT

The solar-thermal power generation study under NSF Grant GI 37815 has been in progress for two months. Planning and organization of the study have occupied much of the first two months. Analytical effort has been started on non-focusing collector performance and heat transport fluid characteristics. In addition, heat engine and cooling subsystem design problems are being investigated.

1.0 INTRODUCTION

The Solar Thermal Electric Power Systems (STEPS) Study has been in progress since 1 May 1973 under NSF grant GI-37815. The effort during the first two months has been primarily that of organizing the tasks, scheduling the work, and collecting information on past efforts in the utilization of solar energy and the production of electric power.

Organization of effort and collection of information have largely been completed, and some progress has been made in deriving new results. This report describes some of the important results of the data collection task and some of the preliminary efforts made toward constructing analytical models and developing a methodology for the subsystem and system optimization which is the ultimate goal of the project.

The Westinghouse team is responsible for data collection on materials, heat engines, heat transport, heat transfer, and energy storage as they apply to practicable solar-thermal power generating facilities. In the second phase of the work, conceptual development, the Westinghouse team has responsibility in the same topic areas, as well as in non-focusing or flat-plate collectors. Conceptual development work has already been started in non-focusing systems and heat transport.

A solar-thermal power generation system block diagram is shown in Figure 1-1. In the upper left, the collectors are shown grouped with the heat transport subsystem to indicate the inter-dependence between them in design choices and optimization. In some system concepts the heat transport subsystem may be relatively unimportant, such as in the mirrors and tower concept being studied by the University of Houston.

Another grouping is made of the turbine-generator with the condenser and cooling subsystem. This is done because choices made in their designs are also inter-related and complicated by the necessity for

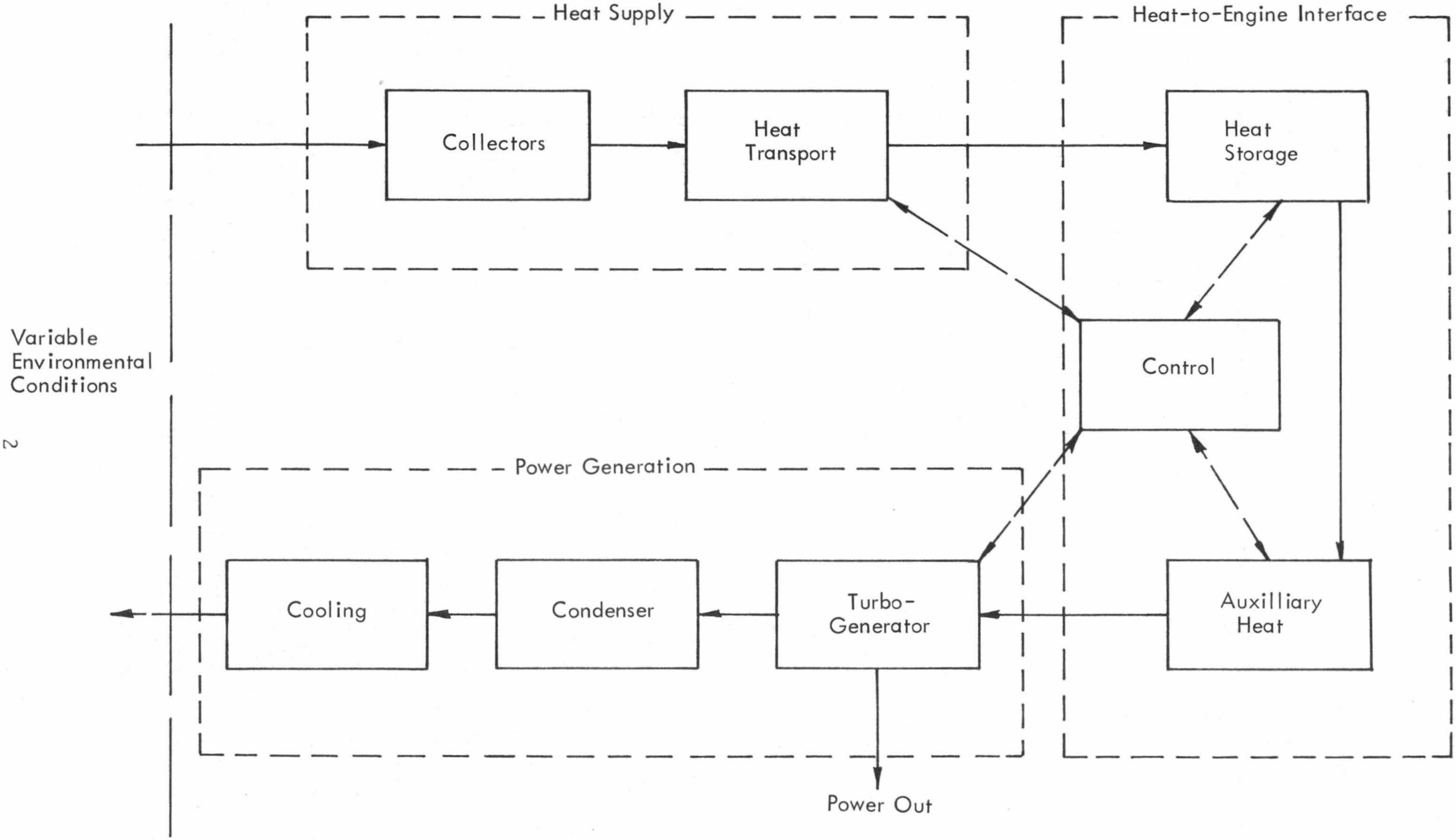

Figure 1-1. Block Diagram of Solar Thermal Electric Power System, Showing Subsystems and Logical Subsystems Groups. Some Systems may not Contain Heat Storage or Auxilliary Heat Supply.

heat rejection to variable ambient conditions.

The third group represents the interface between variable collector field (heat supply) output conditions and the turbine input requirements. As an example of the problem, temporary cloudiness may cause steam at the collector field output to change from dry to wet, and the turbine may be damaged if it receives wet steam. The system design could require this steam to be rejected in some manner, or it could cause heat from a stored heat supply or an auxilliary fuel-fired boiler to be added to bring the steam up to suitable conditions.

System interface problems are some of the most difficult to handle and may present the greatest barriers to the usefulness of solar heat for power generation.

The Westinghouse STEPS study tasks and their scheduled progress are shown in Figure 1-2. At various points inputs are shown from CSU efforts. Dates on each block represent the latest month in which the task should be finished to allow completion of the project. Obviously, early completion of the tasks toward the left of the chart will permit extended intervals between later tasks.

In the discussion below, the beginnings of several of the tasks are described, specifically, the tasks marked 2a, 2b, 2d, 3a, and 3b on the Flow Chart are in progress.

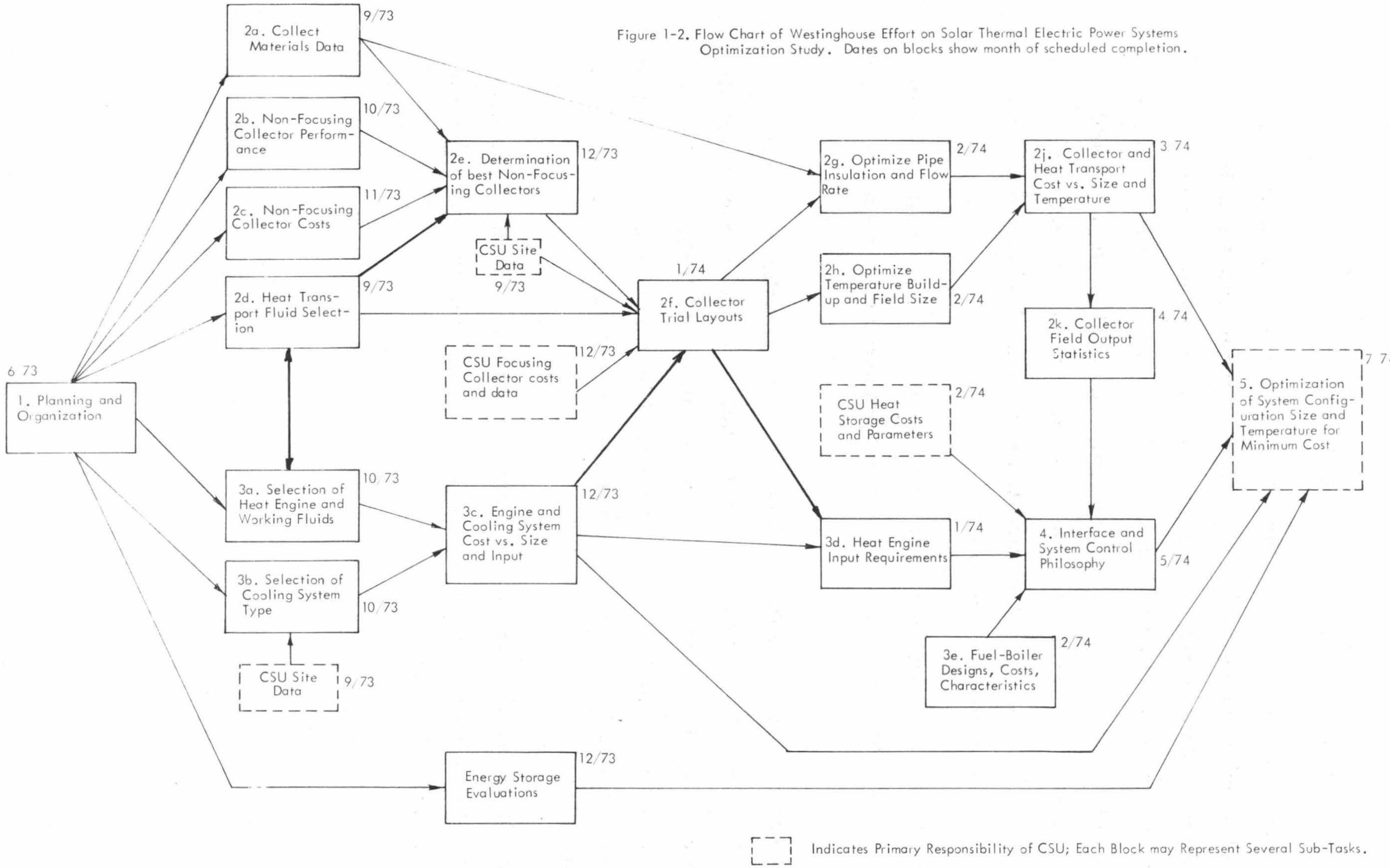

Figure 1-2. Flow Chart of Westinghouse Effort on Solar Thermal Electric Power Systems Optimization Study. Dates on blocks show month of scheduled completion.

2.0 MATERIALS DATA COLLECTION

The objective of the data collection effort for STEPS materials is to assemble U.S. and foreign information on optical, thermal, and mechanical properties of materials suitable for use in solar energy systems. Such materials (and their properties) will include a) opaque selective surfaces (absorbance and emittance), b) transparent selective surfaces (reflectance and transmittance), c) insulating materials (thermal conductivity and effective heat transfer), d) optical focusing materials (reflectivity and surface smoothness and surface stability), e) heat storage and heat transfer materials (specific heats and other characteristics), and f) the variation of these properties with typical operating conditions and exposure.

The materials data collection effort is essentially complete with respect to the more important characteristics listed above. This data has been assembled at the Westinghouse Georesearch Laboratory (WGL) in a loose-leaf ring binder, and a copy has been sent to CSU. Data organization, results, and plans are described below.

2.1 Organization

Following the guidelines in the Statement of Work, information was assembled and collected according to the following outline:

CATEGORIES OF MATERIALS

1.0 COLLECTORS
- 1.1 Opaque and Selective Surfaces
- 1.2 Transparent Surfaces
- 1.3 Optical Reflecting and Focusing Materials

2.0 HEAT TRANSFER AND STORAGE MATERIALS
- 2.1 Liquids, nonmetallic

2.2 Salts and Eutectics

2.3 Metals, Solid and Liquid

2.4 Gases

3.0 INSULATING MATERIALS

4.0 REFERENCES

4.1 Collector Materials

4.2 Heat Transfer and Storage Materials

4.3 Insulating Materials

The section numbers shown in the above categories are those used in the original data notebook. Information sought in the above categories generally agrees with the objectives; however, cost information, being more germain to individual task optimization efforts, will be acquired in the course of various task studies.

2.2 Results To Date

A substantial amount of information has been collected in each category, and only minor additions are expected in the future. At present, the data appears as Xerox copies of published graphs, tables, and pertinent text material. Owing to the diverse formats and units, no effort has been expended at this time toward tabulating and plotting the most important information in each category.

Some appreciation for the scope of information acquired so far can be gained from the list of references from which information was taken. These are presented at the end of Section 2.

2.3 Future Plans

Additional data will be assembled, collated according to the above categories, and inserted into the WGL and CSU data notebooks as the information is acquired. Information is still needed in the following areas:

1) Variation of collector material properties with exposure to weather and typical operating conditions. Some information has been acquired from the University of Minnesota and Honeywell, and they are conducting more tests.

2) Degradation of heat transfer, storage, and insulation materials with use, particularly when used under extreme conditions.

3) Variation of the thermal conductivity of insulations when exposed to weather such as moisture, snow, and hail.

4) Optical properties of various plastics suitable for glazing.

5) Surface accuracy and stability for reflector materials.

It is expected that as subsystem concepts are developed, much of the information on materials will be tabulated with consistent coordinates and units. This information will also be contributed to the materials library.

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Heat Transfer and Storage Materials - References

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8. Jones, T. B., The Feasibility of Electrohydrodynamic Heat Pipes, NASA, CR114392, Dept. of Electrical Engineering, Colorado State University, October 1971.

9. Monsanto Company, Organic Chemicals Division, 2710 Lafayette Street, Santa Clara, California, 95052, (408) 243-0414, Clifford Boutin, Repr.

10. 3-M Company, Chemical Division, 6023 So. Garfield Ave., Los Angeles, California, 90040, (213) 726-1511, Edward T. White, Repr.

11. Dow Corning Corp., Midland Mich., 48640, (517) 636-8000, Rod Cook, Repr.

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3.0 NON-FOCUSING COLLECTOR ANALYSIS

Flat-plate solar collectors may form part or all of the collector field for a solar-thermal power plant, depending on other system design decisions. Within the category of flat-plate collectors, however, are a great variety of possible general types and within each type there are many possible configurations. Table 3-1 gives a breakdown of the flat-plate collector configuration choices which will be evaluated to select the design or designs most suitable for power generation systems.

The first type of collector chosen for analytical consideration has tubes imbedded in a coated aluminum plate (for example, by Roll-Bond ®* type of fabrication). Initially, a comparison is being made of performance of the coated aluminum collector with various numbers and types of transparent covers. At present, computer subroutines are written to calculate heat losses, as a function of collector temperature and ambient condition for

a) 1 glass cover
b) 2 glass covers
c) 3 glass covers
d) 4 glass covers
e) 1 plastic cover
f) 2 plastic covers
g) 1 glass, 1 plastic cover
h) 1 glass, 2 plastic covers.

The program also varies collector surface emissivity in comparing basic flat-plate collectors.

Some initial computations have been made showing relative performance under given input conditions; however, similar performance evaluations have been done by others. The goal of our analysis is a comparison of the cost of the heat output as a function of operating conditions.

Parametric comparisons of collector performance against a true cost

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* Roll-Bond ® Products, Olin Corporation, East Alton, Illinois.

TABLE 3-1

TYPES OF SOLAR FLAT-PLATE COLLECTORS

(Non-Focusing)

1. Types of Absorbers
   a. Metal surfaces with internal tubes
   b. Metal surfaces, matrix or porous
   c. Glass surfaces (overlapped)
2. Types of Glazings
   a. Multi-layers of glass
   b. Multi-layers of heavy plastic
   c. Multi-layers of plastic film
   d. Combination glazings
   e. Cellular plastic slices (large bubble "styrofoam")
3. Types of Fluid Receivers
   a. Liquid temperature increase
   b. Water-steam generation
   c. Gas temperature increase
4. Types of Surfaces
   a. Flat black paints, oxides, enamels
   b. Selective black, low emittance coatings
   c. Transparent honeycomb
   d. Reflective honeycomb
   e. Scratched, finned, or roughened surface
5. Types of Deployment
   a. Stationary tilted
   b. Periodic tilt angle adjustment
   c. Continuous tilt angle adjustment
   d. Continuous, single-axis, east-west adjustment, constant tilt angle
   e. Continuous double-axis adjustment
6. Radiation Augmentation
   a. Adjacent non-specular reflective areas
   b. Adjacent specular reflective areas
7. Special Collector Combinations
   a. Speyer collector -- evacuated glass tube with internal flat absorber-liquid or gas heater
   b. Overlapped glass plate air heater
   c. Liquid heater using semi-transparent surfaces (Solar Energy 12, p. 457)
   d. Collector irradiated from both faces

are difficult to achieve. One comparison might be cost per unit of heat delivered over the life of the collector at a fixed temperature. This implies a variable fluid rate through the collectors during normal annual and diurnal variations. This may not be a realistic mode of operation.

Another comparison could be made by assuming a fixed insolation and ambient and sky temperatures at some annual average value. This is an even more artificial operating condition, but takes much less computation time to evaluate.

If the collector is evaluated over the operating life for a fixed flow rate of transport fluid, a cost per unit of heat can be derived, but the fluid temperature varies over a wide range. Changing the flow rate would change the average temperature. An average operating temperature may have no useful meaning, in any case.

We are not yet certain what criteria are the best for obtaining valid comparisons between different flat-plate collectors. Some additional analyses are required before we can feel confident in a single best approach to collector subsystem optimization.

3.1 Present Modeling Technique

Performance calculations are based at present on the techniques presented by Whillier [1]. While the methods are primarily directed toward the basic collector (Roll-Bond ® plate and transparent cover) chosen above for first analytical efforts, most variations can be worked into the model quite readily and evaluated against the same input conditions.

The basic heat balance equation for a collector gives the usable heat energy per unit area and time, q_u, as

$$q_u = f\,I - U_L\,(T_c - T_a), \qquad \text{watts/m}^2 \tag{1}$$

where

f is a combined cover transmissivity-plate absorptivity factor representing the fraction of incident solar energy captured by the plate,

I is the incident solar energy density, watts/hour $\cdot$ m^2,

U_L is the combined radiation, convection, conduction heat loss per hour, per unit area and per degree centigrade.

T_c is the collector temperature,degrees centigrade, averaged over the plate area, and

T_a is the ambient (air) temperature, degrees centigrade.

The factor, f, depends on the design of the collector, that is, the number and types of covers, the absorptivity and the plate orientation of the collector, shading effect of edges, and even dirt on the cover. Insolation, I, is the normal incidence total of direct and diffuse radiation.

The loss factor, U_L, depends on the design parameters such as glazing and plate (infra-red) emissivity, and it also depends on collector temperature, ambient and sky radiation temperatures, and wind conditions.

The model as presented by Whillier and as used by many others employs a plate efficiency factor to allow the heat losses to be expressed in terms of collector fluid inlet temperature, T_i, rather than the average plate temperature, T_c, shown above. We have chosen, for the initial comparisons between collectors, to use T_c rather than T_i to simplify some of the analysis. Later, in more detailed system modeling, the plate efficiency factor approach will probably be used.

Since considerable reference data are available in the British system of units, the computer analysis is carried out in the British system, and outputs are presented in both British and International (MKS) systems.

3.2 Loss Factors

Heat absorbed by the collector is given by the first term, fI, in Equation (1). The heat losses given by the second term consist of radiation, convection, and conduction losses between the collector absorbing surface and the glass and plastic covers, and between these covers and the atmosphere.

The combination of heat losses, using electrical conductance analogs, is illustrated by the following example of a collector with an outer glass cover and a sheet of plastic between the glass and the collector. as shown in Figure 3-1.

The upward heat loss factor, U_g, is

$$U_g = \cfrac{1}{\cfrac{1}{hc_{1a} + \epsilon_1 hr_{1a}} + \cfrac{1}{\tau E_{c1} hr_{c1} + \cfrac{1}{\cfrac{1}{hc_{c2} + E_{c2} hr_{c2}} + \cfrac{1}{hc_{21} + E_2 hr_{21}}}}} \tag{2}$$

where hc_{xy} is the combined conduction-convection loss coefficient between plates x and y at temperatures T_x and T_y (see Figure 3-1).

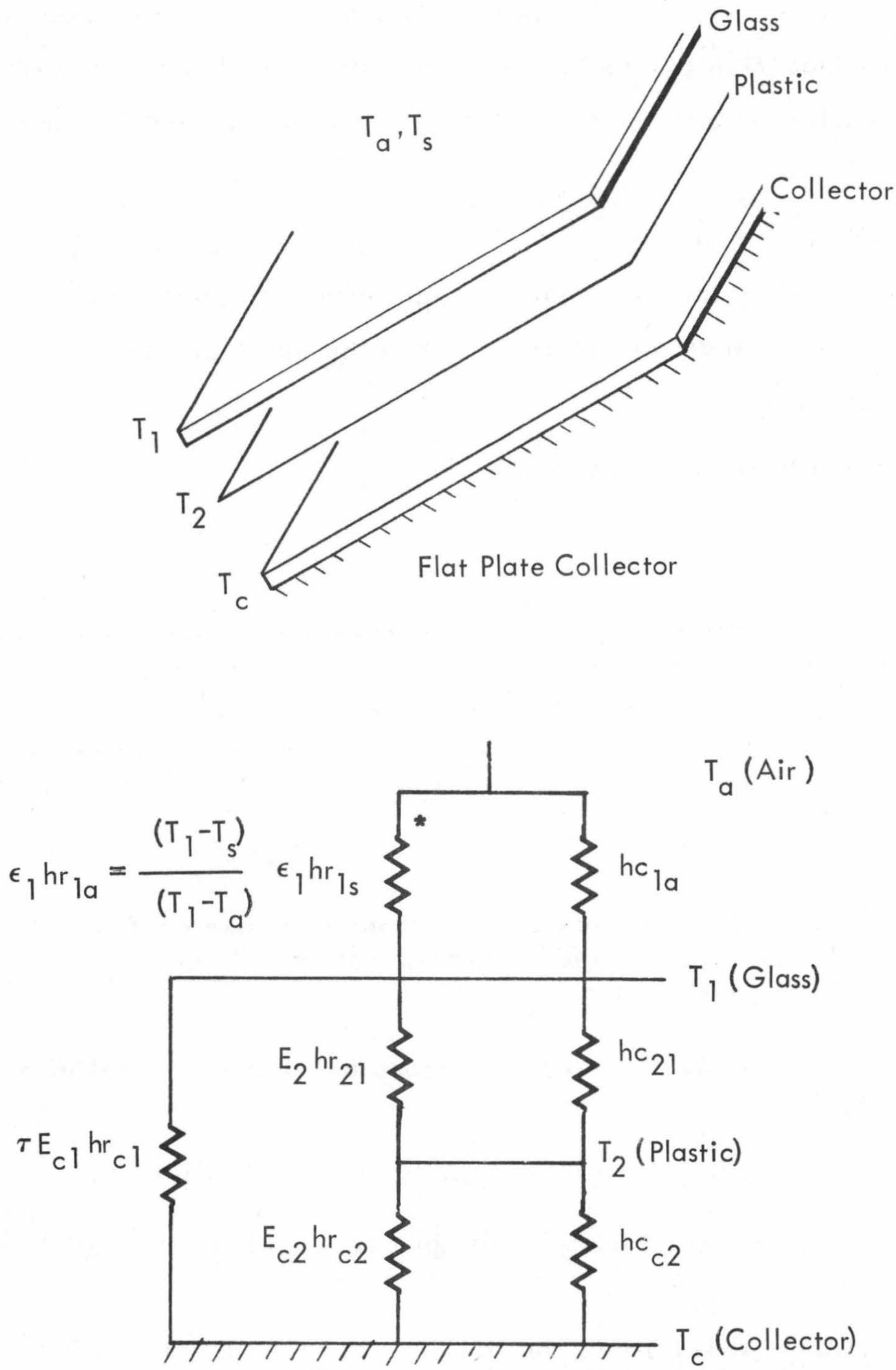
hr_{xy} is the radiation exchange coefficient between parallel plates x and y,

E_{xy} is the effective emissivity between parallel plates,

τ is the transmissivity of the plastic to infra-red radiation, and

ϵ_1 is the emissivity of the top plate (to the sky, which has radiation temperature, T_s).

The conduction-convection coefficient to the air, hc_{1a}, is empirically given by



Heat Conductance Model

* Radiation exchange takes place to a sky temperature which is less than air temperature.

Figure 3-1. Example of Heat Loss Coefficient Derivation from Electrical Analog.

$$hc_{1a} = 1 + 0.3\ V\ \text{Btu/(hr}\cdot\text{ft}^2\cdot{}^\circ\text{F)} \tag{3}$$

where V is the wind velocity in miles per hour. This expression, attributed to McAdams [2], gives approximately twice the loss rate computed from formulas from Holman [3]. The dilemma of whether to use expressions given by McAdams or Holman can be avoided if the outer plate-to-air temperature difference is kept small. This requires several cover plates and a low loss coefficient between them. Where the convection loss to the air is important, the difference will be resolved by further search of the literature.

Conduction-convection loss between plates is described in a 1957 publication of the U.S. Housing and Home Finance Agency [4] regarding research sponsored at the National Bureau of Standards. With a spacing of 2.5 cm (1 in.) between plates, the measured coefficient is closely approximated by

$$hc_{xy} = 0.165 \left[T_x - T_y\right]^{0.3}\ \text{Btu/(hr}\cdot\text{ft}^2\cdot{}^\circ\text{F)} \tag{4}$$

for plates at 45° angle with heat-flow upward. Data is given for an average temperature between plates of 50° F. For 2.5 cm spacing, convection dominates and the conversion to a different average temperature is

$$\left(hc_{xy}\right)_T = \left(hc_{xy}\right)_{50} \left[1 - 0.001\ (T-50)\right]. \tag{5}$$

Thus, the loss coefficient is decreased 10 percent by 100° F greater average temperature.

Between parallel plates, the effective emissivity is given by

$$E_{xy} = \frac{1}{\dfrac{1}{\epsilon_x} + \dfrac{1}{\epsilon_y} - 1} \tag{6}$$

where ϵ_x and ϵ_y are emissivity factors for the individual plates.

The radiation coefficient, hr_{xy}, is given by

$$hr_{xy} = \sigma \left(T_x^4 - T_y^4\right) \Big/ \left(T_x - T_y\right) . \quad (7)$$

where, in the numerator, T_x and T_y must be absolute temperatures. σ is the Stefan-Boltzmann constant and equals $0.173\ (10)^{-8}$ Btu/hr$\cdot$ft$^2 \cdot {}^\circ R^4$ or $5.67\ (10)^{-8}$ watts/m^2 $^\circ K^4$.

In the case of radiation exchange between the top plate and the atmosphere, the coefficient hr_{1s} is formed from the effective sky radiation temperature, T_s, which depends primarily on the moisture content of the air [5] . Specifically,

$$hr_{1a} = \sigma \left(T_1^4 - T_s^4\right) \Big/ \left(T_1 - T_a\right) . \quad (8)$$

T_s is typically 8° C to 25° C (15°F to 45° F) below ambient air temperature. In Figure 3-1 hr_{1s} is adjusted by a temperature ratio to reference it to T_a because the loss factor U_L in Equation (1) is related to air temperature, T_a, rather than T_s.

In the calculation of loss coefficients the temperatures of cover plates must be known, yet the cover plate temperatures are determined by the values of the loss coefficients. The computer programs being written calculate the temperatures and coefficients iteratively. First, the cover temperatures are assumed for calculating the loss coefficients, then these coefficients are used to calculate new temperatures. The process is repeated until the change in results between iterations is insignificant.

The other part of the loss factor, U_L, is the loss through insulation at the back and sides of the collector. This can be maintained at a small proportion of the upward heat loss, U_g. We have chosen to limit back and side losses to 10 percent of U_g. The computer program calculates the insulation thickness required at the back to make the overall loss factor

$$U_L = 1.1\ U_g. \tag{9}$$

Some results are coming from the computer program, but they have not been completely verified and are not in sufficient quantity to be presented with this report. Cost data are not yet assembled on the collectors, but that effort will be started very soon.

3.3 References, Section 3

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4.0 HEAT TRANSPORT

Optimization of the subsystem that transports heat from the collectors to the use point is possibly the most complex requirement in the optimization of the total solar thermal power system. Choices made in heat transport design critically affect the type of collectors used and the configuration of the collector field. For this reason, the cost of alternatives in design choices may be expressed in terms of the cost of collecting the required difference in heat energy. Specific design choices and their equivalent collector cost are discussed briefly below.

4.1 Subsystem Design Choices

Choice of Transport Fluid

An important part of designing the heat transport subsystem is to determine the best choice of heat transport fluid. A single fluid may be used or a combination of fluids as with a phase change during the heating. The choice depends on the heat carrying capacity of the fluid per unit volume, the transfer coefficient from the collector to the fluid, the power required to move the fluid and the fluid cost.

To properly compare working fluids requires an assumption about the end use; that is, whether the fluid is used directly by the heat engine or whether there must be a transfer of heat to the working fluid of the machine. When different fluids are used in the heat transport subsystem and the engine, available heat is lost in the heat exchange. To make up for this, the collectors must operate at higher temperatures, and higher temperature operation means either a higher cost for the collectors or more collectors because of lower fluid flow rates. Section 4.2 gives some examples to illustrate the variation in operating temperatures resulting from the choice of heat transport fluid and the choice of how the collector heat is transferred to the engine working fluid.

Temperature Drops

There are two types of temperature drop in the transfer of heat from the collectors to the heat engine. The first is illustrated above by the heat transfer from one fluid to another or by the transfer of heat from the absorbing surface of the collector to the heat transport fluid. These heat transfers necessarily involve temperature drops in a reasonable design. There is no heat loss involved, but there is a loss in available energy because of the temperature drop.

The other kind of temperature drop is caused by heat losses through the insulation during the transport of the heat carrying fluid. These are true losses of energy from the system, but they can be controlled at the expense of more and higher quality insulation. The heat loss during transmission represents the loss of the highest temperature heat collected, or it is the same as not having the use of the highest temperature output from the most expensive collectors. The cost of the heat loss is the cost of collecting the equivalent amount of heat at the highest temperature.

Pumping Losses

The energy used for pumping the heat transport fluid through the system should be minimized. The cost of this pumping energy loss can be expressed in terms of the cost of producing the additional electricity. In terms of solar-generated electricity, this is primarily an increase in total capital costs.

Subsystems Configuration

Configuration design is the hardest variable to quantify, and there are limitless possibilities for integrating a field of collectors with a heat transport subsystem. We are attempting to design configurations that reduce heat losses by collecting the highest temperature heat near the point of use.

In an example of a good configuration, the heat would be built up in a transport fluid by stages. The fluid returned to the collector field would go first to low cost, low temperature, collectors at the outermost edges of the collector field. In the next stage, medium temperature and higher cost collectors would add heat to the moving fluid, and finally, in concentrating collectors the highest temperature heat would be added to the fluid as it neared the center of the collector field and the point of use.

When such a configuration is analyzed, the division between collector types must be determined. The problem is to find the sequence of collector types and required proportions of each type to minimize the cost of raising the transport fluid temperature from the condenser temperature up to the turbine inlet temperature.

Configuration choices can significantly reduce system losses and reduce the cost of the required collectors, but much of the configuration design must proceed by trial and error.

Fluid Flow Rate

In the transport of fluid between collectors or to the use-point, the losses per unit of energy transmitted are increased by slower flow. Increasing the fluid flow to reduce heat losses will increase the pumping losses, so there is an optimum flow rate to minimize costs attributed to heat losses and pumping.

As the fluid velocity is increased, the pipe diameter required for a fixed volume/hour is reduced. Consequently, the heat losses and pipe and insulation costs are reduced. Optimization of fluid velocity involves minimizing the cost of pipe and insulation and the cost of collectors required to make up the lost heat and the extra pumping power.

Plant Size

A key system variable defined by the heat transport subsystem design is collector field size and therefore maximum plant output. In the configuration described above, the lowest fluid temperatures are supplied farthest from the point of use. As the collector field is enlarged, the amount of heat lost as the fluid is moved from the outer edge of the collector field to the center becomes equivalent to the heat supplied by the outermost collectors. At this point the outer collectors are contributing nothing and the optimum plant size has been exceeded.

This brief consideration of the heat transport design variables illustrates how heat transport subsystem trade-offs may be made in terms of the cost of collectors and the configuration of the collector field. Obviously, the optimization of the collectors and the heat transport subsystem must be closely coordinated.

4.2 Comparison of Three Idealized Heat Transport Loops

The basic purposes for discussing these three systems are to explain important interactions and to develop a methodology for comparing heat transport fluids. Sample calculations are used to show the differences in required collector output temperatures and pumping power losses.

The system assumptions are:

1) Fixed heat engine conditions of 10 MW output with 85 percent turbine efficiency; 400° F (204° C) dry saturated steam at turbine inlet and a condenser pressure of 4 in Hg.

2) Delivery pipe from collector to engine is 100 feet long; the return pipe from engine to collector is 1000 feet long.

3) Pressure drop and heat losses in collectors are neglected.

4) Steady state operating conditions.

For these assumed conditions, the required steam flow rate for a 10 MW turbine is

$$W_s = 1.27 \times 10^5 \text{ lb/hr.}$$

The three systems are shown in Figure 4-1 and are characterized as follows:

1) Pressurized water is circulated in a primary loop through the collector and boiler which produces the steam in a secondary loop (Figure 4-1a).

2) Pressurized water from the collector is flashed to steam in an evaporator (Figure 4-1b).

3) Steam is produced in collectors, which is fed directly to turbine (Figure 4-1c).

A. Pressurized Water-Boiler System

This system is distinguished by the fact that heat is absorbed in the collectors by pressurized water and transferred to the turbine working fluid in a heat exchanger. The water is assumed to have a constant specific heat C_W of 1 Btu/lb/°F and an initial flow rate of $W_W = 5\ W_s$ lb/hr. Also, a zero pinch-point temperature ($T_8 - T_7$ in Figure 4-1a) is assumed. Under these conditions it is found that the collector must deliver water at 565° F under 1180 psia pressure. Then, for a water velocity of 6 ft/sec., the delivery pipe diameter (feeding the boiler) must be about 0.9 feet and the return pipe (to the collector field) must be about 0.8 feet diameter. For reasonable friction losses in pipes of this size, it is found that about 2.6 kW pumping power is required, and the return and delivery pipe thicknesses are 0.6 inches and 0.65 inches, respectively.

B. Pressurized Water-Flash Evaporator

A flash evaporator used to produce the working steam at 400° F, 247.3 psia can operate at an input pressure of 400 psia and 445° F temperature (P_8, T_8 in Figure 4-1b). For an ideal evaporator supplying saturated steam at (4) and returning saturated water at (2) it is found that $W_W / W_s = 16.85$, considerably more water than in the first system. However, the required flow can be reduced by circulating at a higher pressure drop across the evaporator with an attendant increase in pipe thickness. For the assumed conditions, the pumping power is 408 kW, the return and delivery pipes are about 1.6 feet diameter, and the return and delivery pipe wall thickness are 0.23 and 0.37 inches respectively. Thus the diameter is larger but the wall thickness is substantially reduced over the boiler system. The most important difference is the greatly increased pumping power.

C. Circulate Heat Engine Working Fluid Through Collectors

As shown in Figure 4-1c, this system eliminates the boiler and evaporator so that

$$W_s = W_W \quad .$$

Then the delivery conditions (T_4, P_4) are (400° F, 247.3 psia) and the return conditions (T_5, P_5) are (127° F, ≈ 300 psia). Thus, one desirable feature is the reduction of temperature levels in the collector field. Using the same return water velocity of 6 ft/sec, the return and delivery pipe diameters are 0.35 ft and 1.18 ft respectively, and the corresponding pipe thicknesses are 0.06 inches and 0.18 inches. Finally, the required pumping power is 1.68 kW, which is even lower than for the boiler system.

For comparison, the results of the above calculations are shown tabulated in Table 4-1. System optimization for these schemes has not yet been completed, nor have additional system configurations been considered in detail. It can be concluded, so far, that the third system seems to be the most attractive by virtue of fewer required components, lower working temperatures and pressures, and possibly a smaller collector field.

Table 4-1

Comparison of Performance Parameters for
Three Heat Transport Loops

System	W_W/W_s (1)	Collector Output		Pressure (2) Drop, psi		Pipe Diameter, ft.		Pipe Thickness, in.		Pump (3) Power
		T, °F	P, psia	ΔP_R	ΔP_D	D_R	D_D	t_R	t_D	kW
1. Pressurized water-boiler	5	565 (295°C)	1180	3.05	0.214	0.833	0.914	0.59	0.647	2.57
2. Pressurized water-evaporator	16.85	445 (out of throttle) (228°C)	400	152.7 (4)		1.52	1.56	0.226	0.375	408.
3. Steam from collectors	1	400 (204°C)	247.3	11.8	0.212	0.35	1.18	0.063	0.184	1.68

(1) Water flow rate/steam flow rate, $W_s = 1.27 \times 10^5$ lb/hr for 10 MW output.

(2) 1000 foot return line, 100 foot delivery line.

(3) 80% pump efficiency.

(4) Pressure drop, $P_8 - P_4$, across evaporator requires high pumping power.

1 psi = 703.1 kg/m^2 1 ft. = 0.305 m 1 in. = 2.54 cm

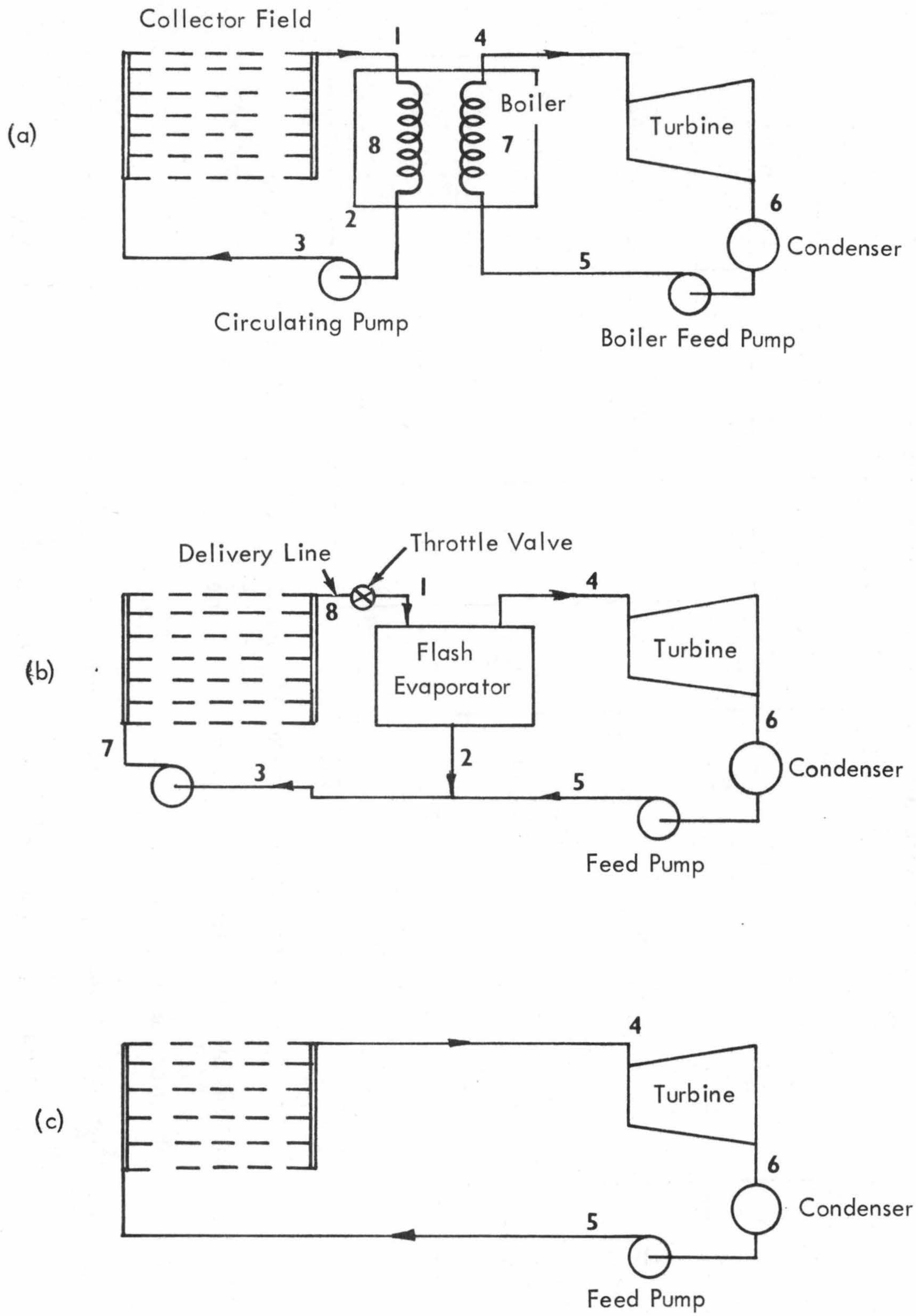

Figure 4-1. Three Schemes for Integrating Solar Heat Collection With the Heat Engine.

5.0 CONVERSION AND STORAGE OF ENERGY

On the subject of the utilization of the heat from solar collectors, Westinghouse has begun the investigation of heat exchanger design, heat engine cycles and working fluid characteristics, optimization of turbine, condenser and cooling system combinations, and electrical energy storage. Detailed discussions are given on the first two of these subjects in appendices to this report, but the salient points of each topic are discussed below.

5.1 Heat Exchangers - (Appendix A)

The design of heat exchangers is largely an empirical process. Data are available on many materials and surface types, but a designer must make choices of shape, flow design, and materials to balance his design and prevent faulty flow distribution, excessive pumping requirements, and scaling or corrosion. Boiler and condenser designs present additional complications because two phases of a fluid must be handled.

It is not meaningful to try to standardize designs, so, rather than try to define heat exchanger cost as a function of some performance parameters, it will be necessary to develop designs as cases require them. In each case the goal will be to minimize cost while providing a balanced design and meeting the design specifications.

5.2 Heat Engine Cycles and Fluids-(Appendix B)

In the conversion of solar heat to mechanical energy and, finally, to electrical energy, some rethinking is possible regarding the choice of heat engines and their working fluids. The cost of collecting high temperature heat from sunlight could make it attractive to use heat engines or working fluids not conventionally employed in power generation.

Probably the most thoroughly studied large heat engine is the steam turbine or Rankine cycle engine. Its widespread acceptance as a prime mover for large generators comes about because of:

1. Relative simplicity of operation and maintenance.
2. Compactness and low flow volume because it uses the heat of phase change rather than just using sensible heat capacity of the working fluid.
3. The cycle is efficient as compared to Carnot cycle (ideal) operation.
4. Heat exchange to the working fluid is external to the engine and thus the heat source is not restricted to a particular fuel.

By contrast, other well known cycles (Brayton, Otto, and Diesel) and less well known cycles (Stirling and Ericsson) use a gas, usually air, as the working fluid and thus require a large volume flow and higher temperatures for comparable efficiency. In the most widely used heat engines using air, the heat is added by the combustion of fuel, either internal or external to the engine itself. Internal combustion air cycles are not amenable to using solar heat, and the external combustion air cycles are less satisfactory than the Rankine cycle because of the larger size and higher temperatures required for comparable efficiency.

Therefore, our effort will be concentrated on evaluating performance of the Rankine engine, and several working fluids including water will be considered. The thermodynamic properties of water are the best known of any fluid, but extensive data are also available for several possible low temperature working fluids such as ammonia and the Freons.

5.3 Turbine, Condenser, Cooling System

The purpose of a cooling system is to lower the condenser temperature at the outlet of the turbine. In effect, the condenser heat is transferred to the environment by heating a body of water, evaporating water or heating the air. A lower condenser temperature permits more efficient engine operation if the engine and condenser are designed to use the lower temperature. However, the operation of the cooling system depends on ambient conditions such as air temperature and relative humidity. Therefore, the design of a condenser and cooling system can be considered an effort to match engine operating conditions to variable atmospheric conditions.

The best design for the engine, condenser, and cooling system must be selected by evaluating the performance and cost of a variety of possible combinations against the ambient conditions at the plant location during an annual cycle. During parts of the year, the optimum system will be operating under conditions for which a different design would be optimum. However, because the input conditions vary, the turbine, condenser, and cooling system design must be chosen which produces the lowest cost energy output during the whole year, not just for one set of operating conditions.

Reference [1] at the end of this section describes a Westinghouse program for optimizing the turbine, condenser, cooling system design, where most of the options are provided in the condenser design. In optimizations of the power generation components for the STEPS study, fewer options can be considered, but a similar technique will be employed.

5.4 Energy Storage

An excess power generation capacity can be matched to the load by storing the energy in some form which can be readily converted to the desired electric power. Energy forms often considered for storage are:

1. Mechanical: Pumped hydro storage is economically used. Compressed gases have also been studied. Kinetic energy (flywheel storage) has been investigated, but has not been shown to be attractive on a large scale.

2. Chemical: Storage batteries are a familiar form of chemical storage. Another is the production of a fuel, for example, hydrogen recovered from hydrolysis of water.

The examples given above do not exhaust the list of possible energy storage techniques, but simply classify some of the more attractive candidates.

Reference [2] of this section is a comparison of the cost of storage by lead acid batteries to pumped hydro storage. The study was done by Westinghouse for a large utility and shows the cost factors that must be considered. Present day batteries require nearly twice the annual cost of pumped hydro in the example case, but technological advances in battery life could close this gap considerably.

A comparison of electric load requirement to the time availability of sunlight suggests that the power generated by a solar plant could be used without any requirement for storage. Therefore, until the day that solar generated power becomes a large contributor to the United States energy supply, storage in this sense will probably not be a part of an economical solar power system.

For completeness, the studies of other energy storage systems will be reviewed in this project to indicate which might be most attractive for future use. However, the choice of a particular type of energy storage system has little to do with the first source of the electricity, that is, whether it comes from a nuclear plant or a solar plant.

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APPENDIX A

HEAT EXCHANGER DESIGN

By

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May 1973

Heat Exchanger Design

The design of a heat exchanger is highly empirical. This is largely because even in this day of greatly advanced analytical capability in all manner of scientific and engineering endeavors, only the simplest geometries lend themselves to adequate fluid flow analysis. Therefore, the designer needs, first of all, empirical data on the heat transfer and friction characteristics of the surfaces he will employ in his design. Fortunately there is an abundance of data in the technical literature on a wide variety of surfaces.

The next concern of the designer is that his surface configuration results in a reasonable structure. Lack of attention can result in a heat exchanger core whose aspect ratios are absurdly unbalanced. This is particularly hazardous in designing heat exchangers where one or both fluids are a gas. The tendency is to have short flow lengths and large frontal areas. This makes the task of the mechanical designer difficult at best and impossible in the extreme.

The matter of aspect ratio is important in another consideration, that of flow distribution. Maldistribution ranks above fouling in reducing the effectiveness of a heat exchanger, in most instances. Not only does an awkward heat exchanger aspect ratio result in clumsy ducting, but it invites flow distribution difficulty. It can also cause poor equipment arrangements in order to accommodate oversize transition ducting or piping. This factor is often overlooked in optimizing heat exchanger design, being outside the design "envelope". Nevertheless, the job is not complete unless this consideration is included.

The matter of materials in a heat exchanger has two important ramifications. First of all is compatibility with the fluids passing through. Sufficient resistance to corrosion and erosion must be obtained to secure adequate life of the unit. From a heat transfer point of view, the resistance of the material to heat transfer is oftimes the major resistance of the unit hence its conductivity plays the major role in determining area requirement. In the same context, fouling due to scaling vs corrosion must be accounted for and minimized.

A balanced design usually requires that the power used to pump the fluids through the exchanger be kept in reasonable proportion. The usual design will keep the pressure loss through an exchanger at about 5% of the head available at the entrance. Obviously, there will be some variation about this value in specific instances, but this is a good value for a first cut.

Closely related to the foregoing considerations is the matter of flow arrangement, i.e., whether the fluids shall flow in opposite directions (counterflow), cross paths (crossflow) in the same direction (parallel flow) or combinations. Thermodynamically, counterflow introduces the least amount of irreversibility and is therefore the most desirable, theoretically. Practical considerations such as headering can over-ride such niceties, however, because of prohibitive cost. A good crossflow design or multiple pass - overall counterflow can provide a very acceptable alternate arrangement and ease the manufacturing task considerably (such as the 1-2 feedwater heater, for example).

In cases where a change of phase is to take place, such as in a condenser, the designer's task is further complicated. Small amounts of non-condensibles (<1%) can cause large increases in surface requirements (>20%). Provision must be made therefore to remove as much of the non-condensibles as is practical or make a careful accounting for their presence in calculating surface requirements.

The design of evaporators is much less taxing than that of condensers. Boiling heat transfer is quite efficient and the process is forgiving of relatively large errors. It should not be assumed however, that the design of a boiler can be accomplished with little care. The evaporator section is only part of the overall design. In boilers where natural circulation is depended upon, a good deal of empiricism is called for.

The references cited below are representative rather than inclusive of the available literature in heat exchanger design. Fluid properties are available in a large number of handbooks and tables. It should not be assumed that heat exchangers design can be undertaken in cook-book fashion. Experience is indispensible in this highly empirical art.

W.H.C.
5/31/73

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APPENDIX B

THE APPLICATION OF SOLAR ENERGY TO THE PRODUCTION OF ELECTRICAL ENERGY FOR LARGE GENERATING SYSTEMS

By

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and
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June 1973

Introduction

The successful application of solar energy to the production of electrical energy for large generating systems represents an enormous and difficult challenge to the United States. In reviewing the problem, it is important to distinguish between using the sun's energy to generate electricity and using solar energy for such purposes as water heating, space heating, and space cooling. The technical feasibility for the latter applications have been clearly demonstrated throughout the world but the conversion of solar energy to electrical energy must await the development of advanced technologies.

About 100 years have elapsed since John Ericsson[1] first began seriously experimenting with novel reciprocating air engines to harness the energy of the sun. More recently some experimenters such as B. S. Leo and S. T. Hsu[2] have actually executed working models of very simple reaction turbines for solar energy applications.

A review of solar energy activities throughout the world suggests that the most ambitious program to date to harness the sun's energy is occurring in the Ararat Valley in Soviet Armenia.[3] The solar power station at Mt. Ararat consists of a single turbo-generator unit of 1200 kW capacity to produce electricity locally to operate pumps for sub-soil

[1] John Ericsson, "Air Engine", U.S. Patent 226,052 (March 30, 1880).

[2] S. T. Hsu and B. S. Leo, "A Simple Reaction Turbine as a Solar Engine", THE JOURNAL OF SOLAR ENERGY SCIENCE AND ENGINEERING, July-October 1958.

[3] A. J. Steiger, "Russia's First Solar Power Station to Have 1200-kW Thermal Capacity", ELECTRIC LIGHT AND POWER, 36(7): pp. 72, 124-125, March 25, 1958.

drainage and irrigation for land reclamation. Such pilot plants could begin to illustrate the development, design, and economic relationships necessary for successful solar power generation systems in the United States.

Although many technical areas can be immediately defined which require intensive investigation in order to develop a solar energy conversion system having the characteristics and capability to operate reliably, and in many instances, in parallel with existing large generating systems; only three technical areas will be identified and reviewed in this limited study.

The first area is the problem of choice of thermodynamic heat cycle. This includes the choice of working fluids and cycle limitations. The second is the power conversion system, particularly the principles and problems of heat engines. The third and final area of study is the heat rejection system of the heat cycle and its characteristics.

In summary it could be noted that an operating fluid in any heat cycle is subjected to the same four sequences of events -- namely, (1) compression, (2) heating, (3) expansion, and (4) cooling. What man has been able to invent are merely ingenious ways of utilizing those fundamental principles to ultimately produce machines that extract more useful work.

Vapor Cycles for Power Generation

The most basic insights into the problem of converting heat into work stem directly from the First and Second Laws of Thermodynamics.

The First Law is essentially a statement of conservation of energy, which indicates equivalence between heat$^{(Q)}$ and work$^{(W)}$. It has been found through experience that when heat and work are applied to a system in a cyclic process, the sum of heat and work transport across the system boundary must be zero for any complete cycle. In a heat engine cycle, we must therefore expect to convert some fraction of the input heat energy to output in the form of work, while the remainder must be returned to the surroundings as heat.

The Second Law of Thermodynamics places a maximum value on the fraction of input heat which may be converted to work. This value is dependent only on the temperatures of heat inputs and heat outputs. This ratio of temperatures is generally referred to as the Carnot Efficiency in honor of Sadi Carnot who first indicated how it might theoretically be achieved in a heat engine operating with a Carnot Cycle.

The Second Law of Thermodynamics, with its wide range of corollaries has resulted in definition of the absolute temperature (T), an intensive thermodynamic property which is defined by the Kelvin and Rankine temperature scales; and entropy (S), an extensive thermodynamic property which is defined by the equation

$$dS = \frac{dQ_{rev}}{T} \; *$$

---

*The rev subscript indicates that dQ is the differential heat input which would be required by a reversible process which is free of friction, temperature gradients and other sources of irreversibility.

The description of heat engines is greatly aided by the study of the cyclic fluid states on a graph of fluid absolute temperature versus entropy (the T-S diagram).

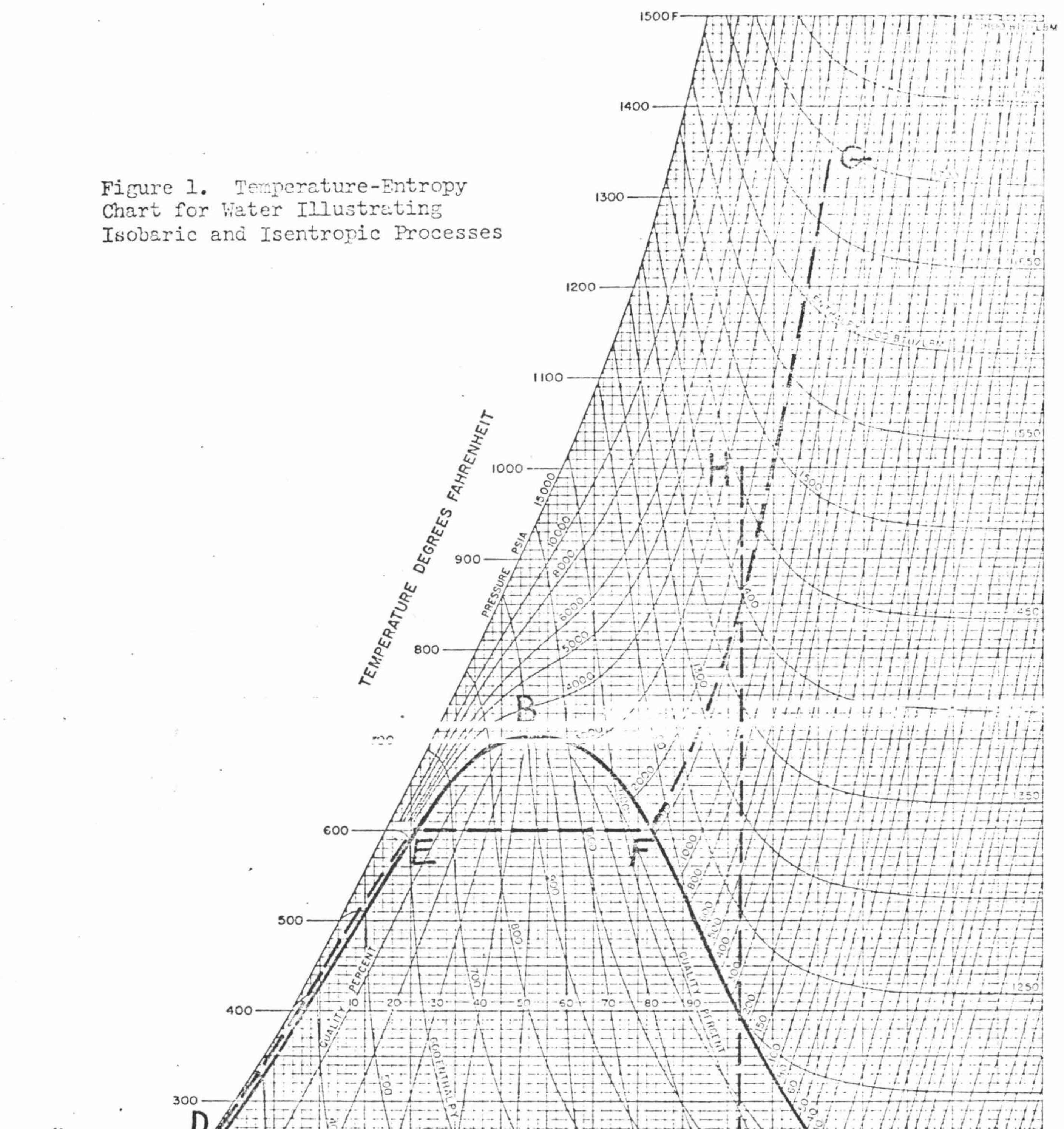
Fluid Properties and Processes on the T-S Diagram Using Steam and Water

Figure 1 contains a temperature-entropy diagram for water which is the working fluid most used in power generation cycles. The diagram defines three distinct regions of fluid properties. These regions are separated by the dome shaped saturation line ABC. To the left of the dome is the liquid phase, and to the right is the vapor phase, while the area under the dome describes the properties of liquid-vapor mixtures.

The familiar process of heating water at atmospheric pressure is indicated by the constant pressure line (isobar) DEFG. The segment DE indicates heating of the compressed liquid to its saturation temperature. The slope of this line is $dT/dS = T \frac{dT}{dQ} = \frac{1}{C_p} T$. It is therefore recognized that a high value of the slope (dt/ds) indicates a low value of specific heat (C_p) for the fluid. For water and many other fluids, the liquid phase isobars are nearly parallel to the saturation line, so that its slope is a useful indication of specific heat.

The segment EF describes the process of evaporation. Since $dS = \frac{dQ}{T}$ and the process EF occurs at constant pressure, it is recognized that $S_f - S_e = \frac{H_f - H_e}{T_f}$ where H is the enthalpy of the fluid. Using the conventional subscript fg for the evaporation process,

Figure 1. Temperature-Entropy Chart for Water Illustrating Isobaric and Isentropic Processes



$S_{fg} = \frac{H_{fg}}{T_f}$, indicating that the latent heat is represented by the width of the steam dome times the absolute temperature.

As the process extends beyond F to G, the temperature again increases with the slope of the path indicating the specific heat of the vapor phase. This region is referred to as the superheat region.

If this same process is performed at higher pressures, the path remains similar until one reaches the "critical pressure" of the fluid which is 3206.2 psia. At this pressure the path passes tangent to the steam dome at point B with a "critical temperature" of 705.4°F. At this point the latent heat of vaporization is zero. Indeed, there is no distinction between liquid and vapor phases and the material is referred to only as a fluid. Processes which occur under these circumstances are called supercritical.

Another important process which is well described on the T-S diagram is the adiabatic expansion or compression. If these processes occur ideally without internal losses, which are evidenced by internal heat generation, they must also be isentropic. Adiabatic expansion of a superheated vapor is indicated by the line HI. This is the ideal path of a fluid as it expands through a heat engine and clearly indicates how in the example of steam, moisture must form as the expansion enters the steam dome. In practice, such an expansion will result in an increase in entropy due to internal losses in a real engine.

The Ideal Cycle

The ideal cycle proposed by Sadi Carnot for conversion of heat into work is shown on the T-S diagram of Figure 2. The Carnot Cycle consists of four processes which are most easily described when occurring entirely within the liquid-vapor region. Beginning at a point A the fluid is compressed adiabatically and isentropically to a higher pressure at point B. It is then heated at constant temperature to produce a higher quality steam at point C. This fluid is then expanded adiabatically, doing work in a heat engine, until it reaches the initial pressure and temperature at point D. Isothermal cooling of the fluid returns it to point A to complete the cycle.

Thermodynamic analysis of this cycle indicates an efficiency of

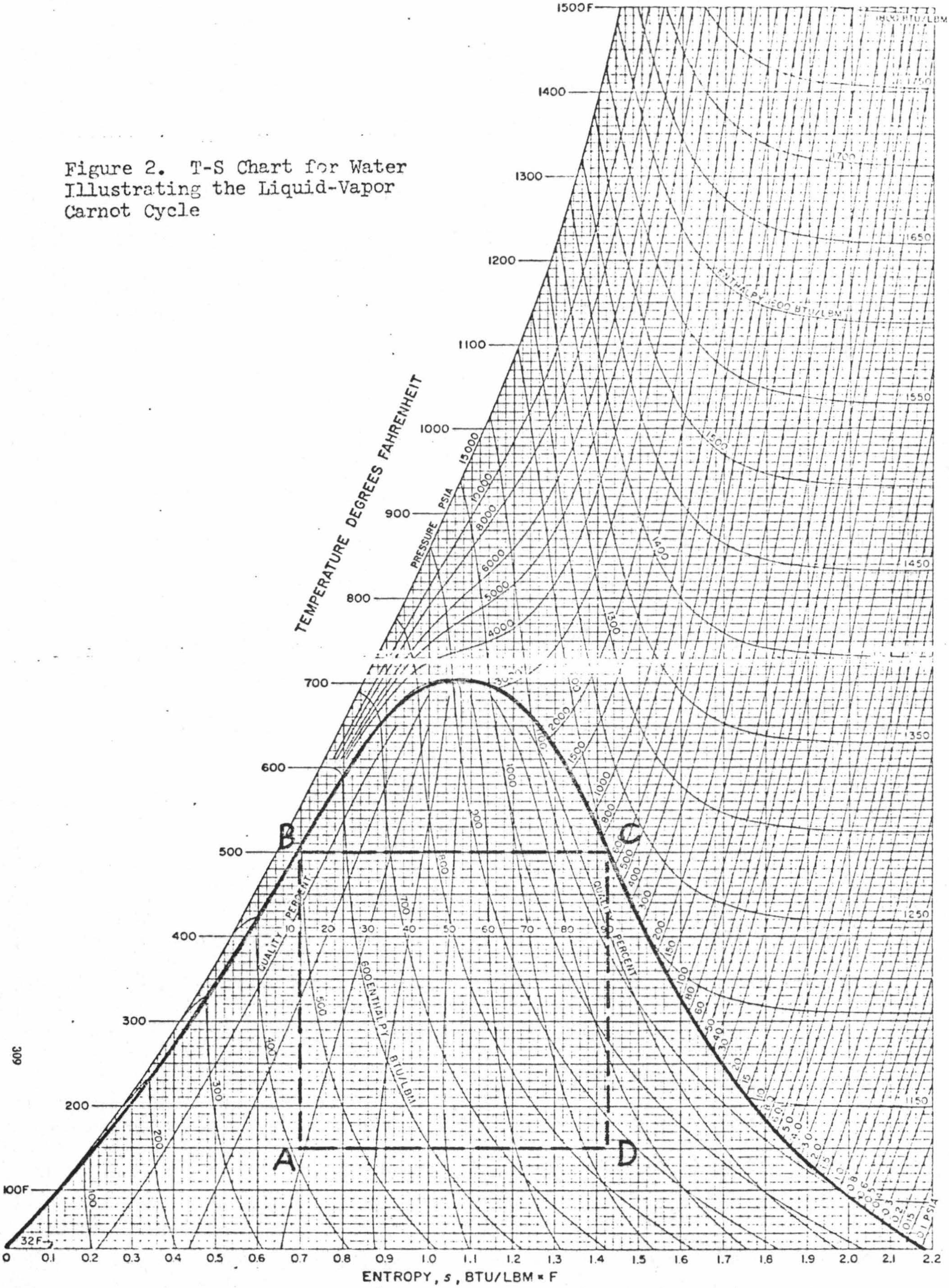
$$\eta_{Carnot} = 1 - \frac{T_{cold}}{T_{hot}}$$

as is shown in any basic thermodynamics text. Values of Carnot efficiency are shown in Figure 3 as a function of heat input temperature for heat rejection temperatures of 80°F, 120°F, and 160°F.

Practical Heat Engine Power Cycles

Few areas of modern technology have benefited from more consistent and fruitful development than the design of cycles and equipment for the conversion of heat into work. Clearly, there is no unique choice of cycle or equipment which provides the best source of power

Figure 2. T-S Chart for Water
Illustrating the Liquid-Vapor
Carnot Cycle



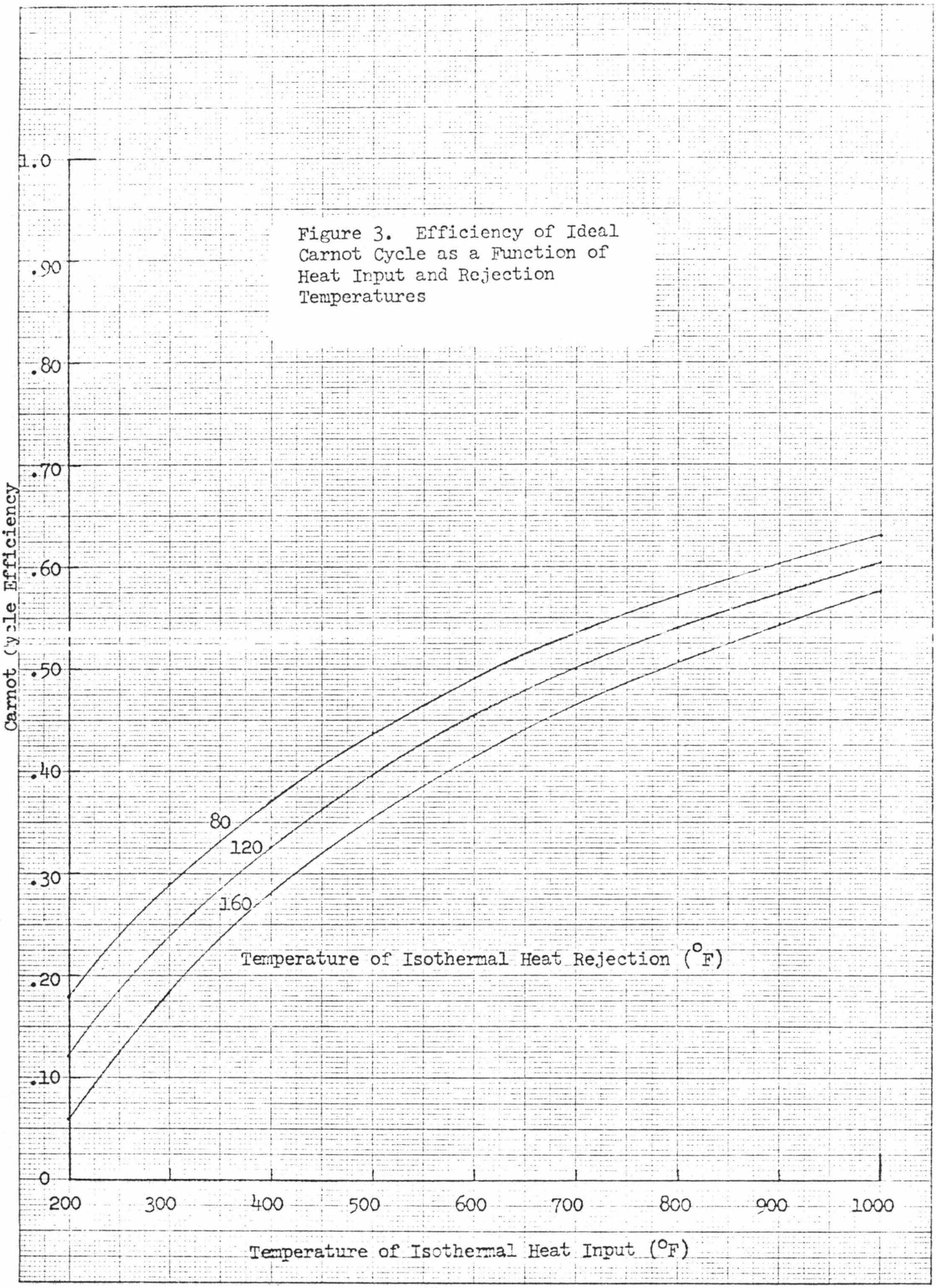


Figure 3. Efficiency of Ideal Carnot Cycle as a Function of Heat Input and Rejection Temperatures
Carnot Cycle Efficiency
1.0
.90
.80
.70
.60
.50
.40
.30
.20
.10
0
80
120
160
Temperature of Isothermal Heat Rejection (°F)
200
300
400
500
600
700
800
900
1000
Temperature of Isothermal Heat Input (°F)

for all applications. Rather, several very different classes of prime movers have enjoyed parallel development because of their marked advantages in dissimilar applications.

The reciprocating Otto and Diesel Cycles use air as a working fluid and find wide application as inexpensive, small scale power sources, but by no means do they approach Carnot efficiency. They are inherently limited in power output by large size requirements for a given fluid mass flow rate.

The gas turbine is finding greater and greater use as a power source for moderate to high output and moderate efficiency. The design is well adapted to handling very high working fluid mass flow rates, and indeed its use at low output levels requires small, highly toleranced, and expensive engine components. Attainment of reasonable efficiency has been historically tied to attainment of very high turbine inlet temperatures, which are generally available only with direct internal combustion. Many enhancements to the simple gas turbine cycle are possible which can significantly improve efficiency. Their application, however, requires extensive heat transfer apparatus which is capable of transferring heat with low losses to very large volumes of working fluid per unit of power output.

For large central station power generation, a modified Rankine cycle has dominated the market due to its unexcelled efficiency, reliability, and adaptability to large output designs. Because practical embodiments of the Rankine cycle have most closely approached the ideal

Carnot efficiency prescribed by the Second Law, this cycle must be very attractive for solar power applications, where collection of input heat, especially at high temperatures, is very difficult. The discussions which follow are most specifically directed toward Rankine cycle engines; not to the exclusion of other alternatives, but to provide a logical background to the state of the art in central station cycle development.

An interesting candidate for solar power applications is the Stirling cycle, which promises theoretical efficiencies approaching the ideal. For this reason alone, it merits considerable attention. But since practical embodiments of the cycle involve reciprocating machinery similar to that of the Otto and Diesel Cycles, application seems best suited to small power outputs. The fact that heat input requires convective heat transfer to large volumes of low density gas reinforces the same conclusion.

The Rankine Cycle as Applied to Conventional Steam Power Plants

Although the Carnot Cycle provides a theoretical means of obtaining maximum conversion efficiency, its practical application leaves much to be desired. For instance, application of the cycle described above could involve compression and expansion of excessively moist fluids, which lead to erosion problems in engines, as well as high aerodynamic losses. In addition, the work required to compress the fluid is a rather high fraction of that which is extracted through expansion. This not only involves large compressors, but reduces the net work output from the cycle.

A more practical cycle is achieved by modifying the Carnot Cycle to accomplish the compression process in the liquid phase. In this way the work of compression is minimized by the low fluid volume. Such a cycle is shown in Figure 4 and is known as an ideal Rankine Cycle. The isentropic compression is replaced by three processes which include condensation of the vapor at constant pressure along the path AA', isentropic compression of the fluid along the path A'B', and isobaric heating of the fluid to the point B. State B' is drawn off scale simply to illustrate the compression process which would not otherwise be visible.

The efficiency of this cycle is lower than that of the Carnot Cycle operating with the same upper and lower temperature limits. This can be recognized by dividing the overall cycle into a large number of arbitrarily narrow Carnot Cycles as shown in Figure 5. The cycle is now approximated by many smaller cycles, some of which operate with the original Carnot efficiency, but many of which convert a portion of the input heat to useful work with efficiency somewhat below that of the parent cycle.

Theoretically, this net degradation of efficiency can be completely eliminated through the use of regenerative feedwater heating. The T-S diagram of a regenerative Rankine cycle is shown in Figure 6. Almost all of the heat is added to the fluid at the maximum cycle temperature along the path AB, before passing into the engine. At some point C during the expansion process, a small fraction of the fluid is

Figure 4. Idealized Rankine Cycle Using Water. The Isentropic compression A'B' is exagerated.

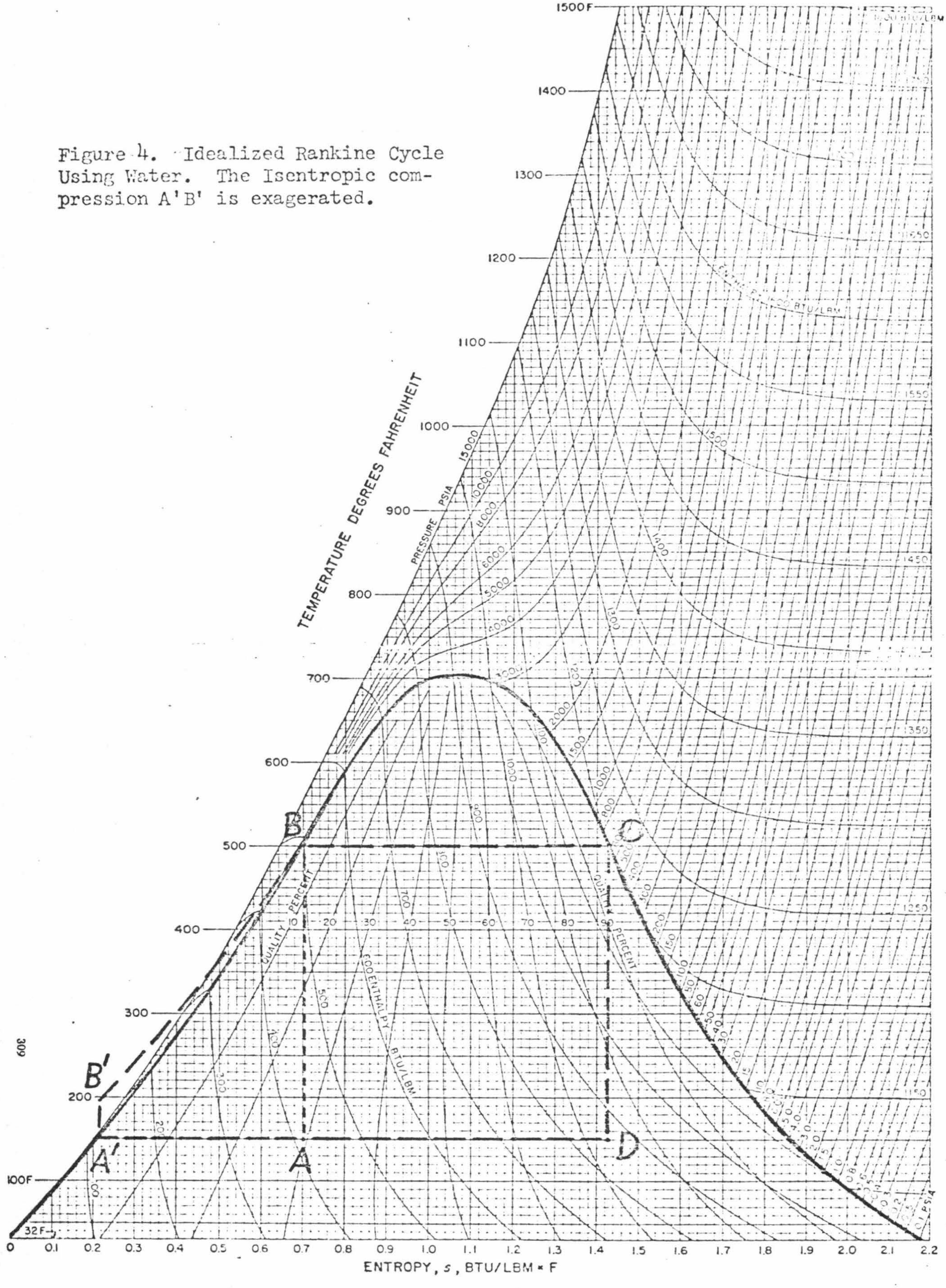


Figure 5. Idealized Rankine Cycle Approximated by Several Carnot Cycles

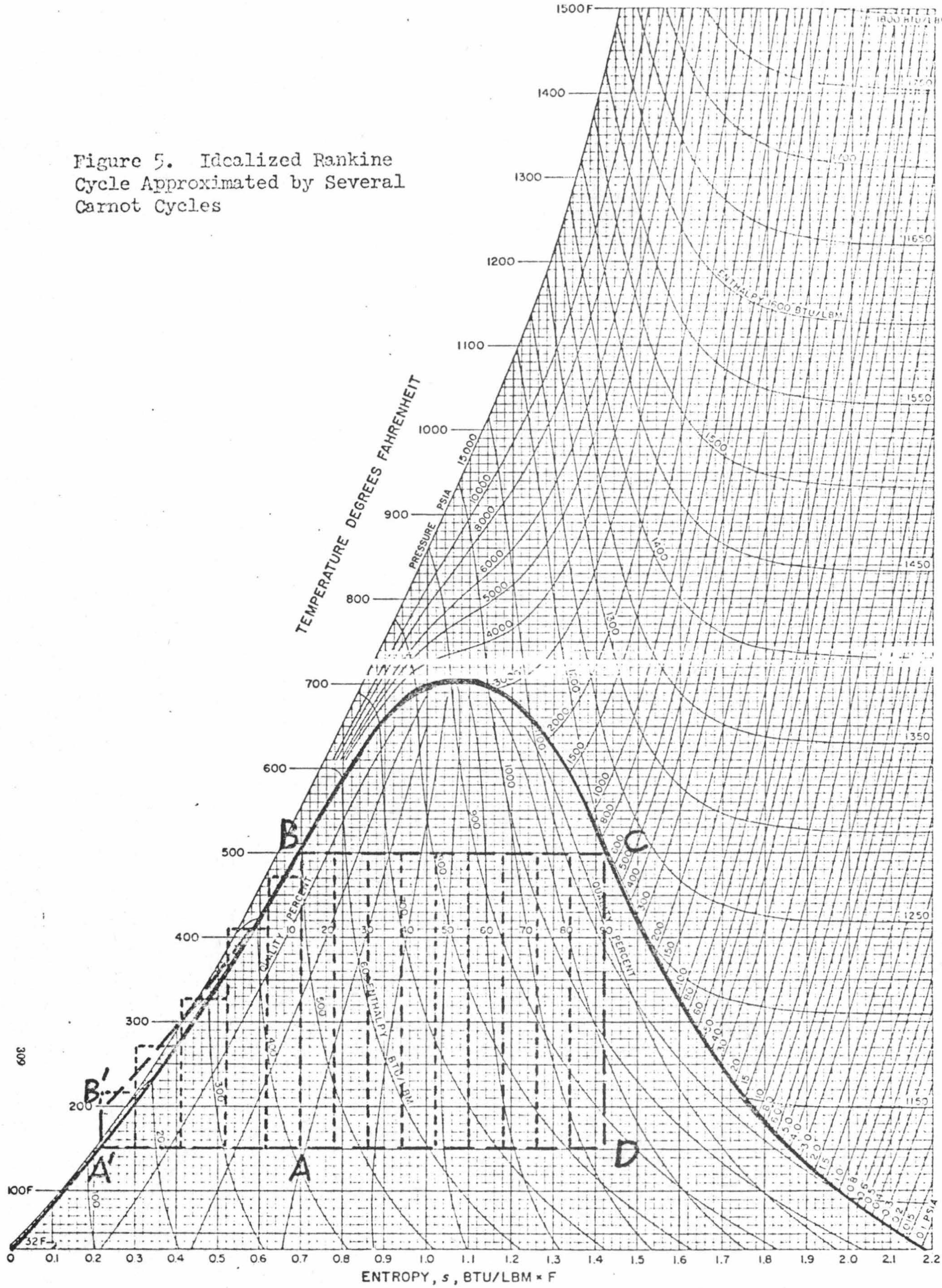
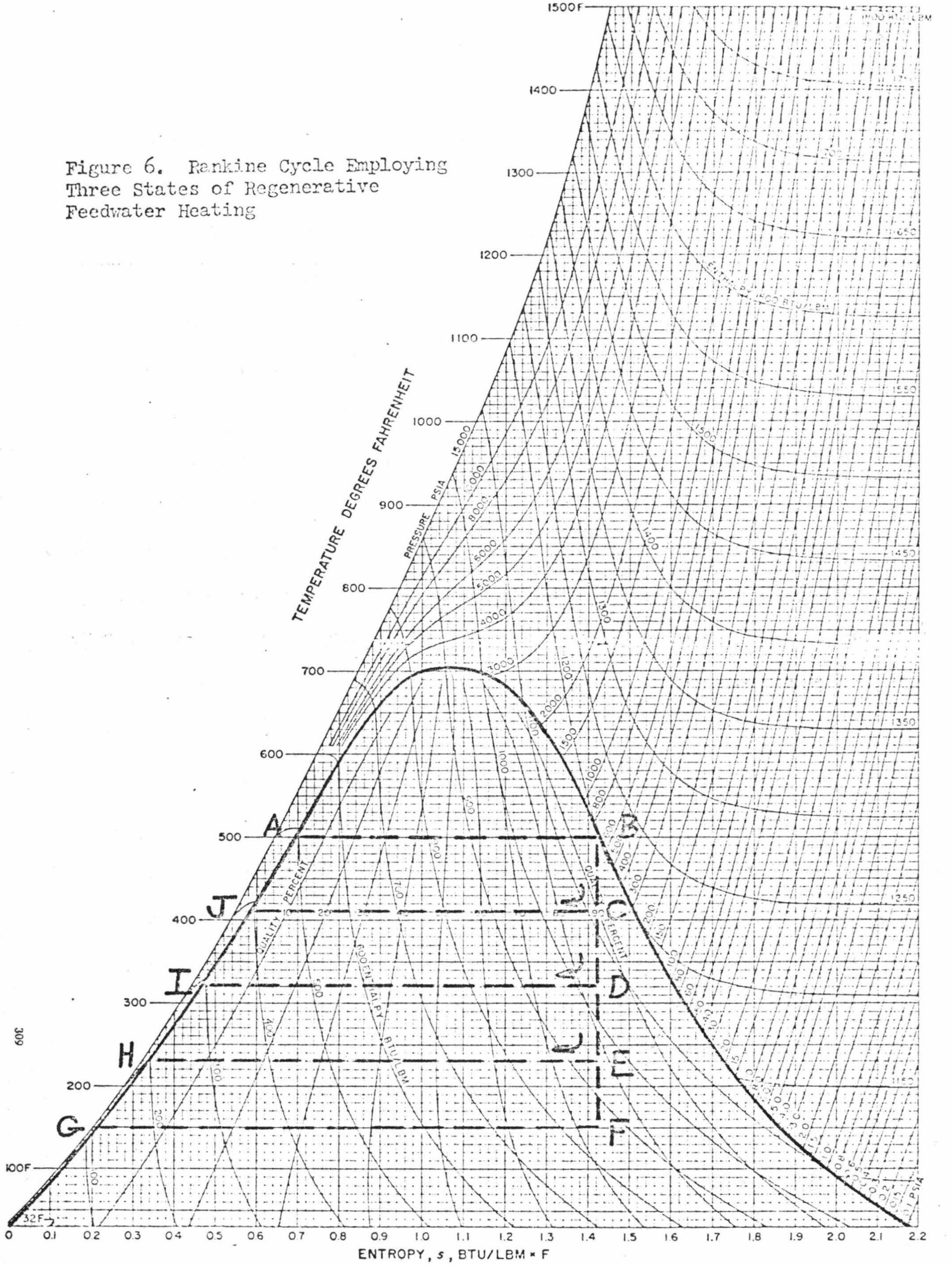


Figure 6. Rankine Cycle Employing Three States of Regenerative Feedwater Heating



extracted from the turbine and condensed at nearly constant pressure. Its latent heat is used to supply the heat requirements of the isobaric feedwater heating process IJ, and the condensate is itself mixed with the feedwater. Only a small quantity of heat is required below the maximum cycle temperature to raise the feedwater temperature from state I to state A. Several similar stages of steam extraction and feedwater heating are encountered during the expansion, and perhaps only 65% of the engine inlet mass flow reaches the condenser.

If a cycle could be constructed with an infinite number of feedwater heaters with infinitesimal resistance to heat transfer, then the cycle could be made entirely reversible and thus equal the ideal Carnot efficiency. Practically speaking, modern steam cycles use five to seven heaters to minimize irreversibilities in the regenerative feedheating process.

Two important benefits accrue from the use of regenerative feedwater heating in the conventional steam Rankine Cycles. First, a larger portion of the cycle work is obtained from steam at the highest cycle pressures. This allows the use of more compact machinery. The engine condensing end, which is large due to the very low exhaust steam density, must be made only large enough to handle the inlet steam flow minus the steam extracted for feedheating.

Secondly, there is potential for moisture removal during extraction. Engines can be designed so that the extracted steam contains a higher than average fraction of moisture, leaving higher quality steam to continue expansion. Such designs, especially applicable to today's nuclear plant low pressure turbines, can significantly reduce blade erosion damage due to moisture.

Modern steam power cycles embody two additional features which surprisingly decrease the theoretical efficiency of the cycle relative to the Carnot Cycle. These are superheat and resuperheating. This anomally requires some explanation.

As noted previously, the critical point for steam occurs at 705.4°F and a pressure of 3206.2 psia. The Rankine Cycles which have been illustrated thus far involve evaporative heat input which is both isothermal and isobaric, so that all heat input occurs at the maximum cycle temperature. This is a sound theoretical objective only insofar as it does not place a limit on the maximum cycle temperature. This indeed is the case with water in conventional steam plants. Metallurgically, heat engines and steam generators can be designed to operate continuously at temperatures exceeding 1100°F, so that more efficient cycles are indeed possible. It is not, however, practical to add all of the heat at such a temperature because this would require a complex process of heating with a well controlled decrease in pressure. But several means are available for approaching a goal of maximum heat input at the highest cycle temperatures.

First, the fluid is superheated to raise the maximum temperature to perhaps 1100°F. Introducing the concept of replacing a cycle by an arbitrarily large number of hypothetical Carnot Cycles, it is evident that the addition of superheat, as illustrated by the cycle ABCDE in Figure 7, will increase the overall cycle efficiency. However a majority of the heat is still added at lower temperatures during the evaporative process.

A greater proportion of heat may be added above the saturation temperature by introducing one stage of resuperheating by the cycle ABCDEFG as shown in Figure 8. By expanding the superheated steam to lower pressures, then resuperheating it to the maximum cycle temperature, and repeating this process for an arbitrarily large number of cycles, one can approach the controlled pressure decrease which is necessary for isothermal heat input from states D to F. Actually the practical limit to such an ideal strategy is in the cost of moving fluids to and from the engine from the heat source, but an ultimate limit results from irreversible pressure losses which degrade the cycle performance. Aside from the benefits of superheat and resuperheating to theoretical cycle efficiency, they are necessary to limit moisture formation in the low pressure end of the engine, thus greatly reducing erosion of critical engine components.

An alternative method of increasing the average temperature of heat input to the engine is to add heat at supercritical pressures. This results in a modest improvement in efficiency but is primarily justified by providing higher heat transfer rates in the heat input portion of the cycle, and thus more economical equipment.

Figure 7. Rankine Cycle with Superheat

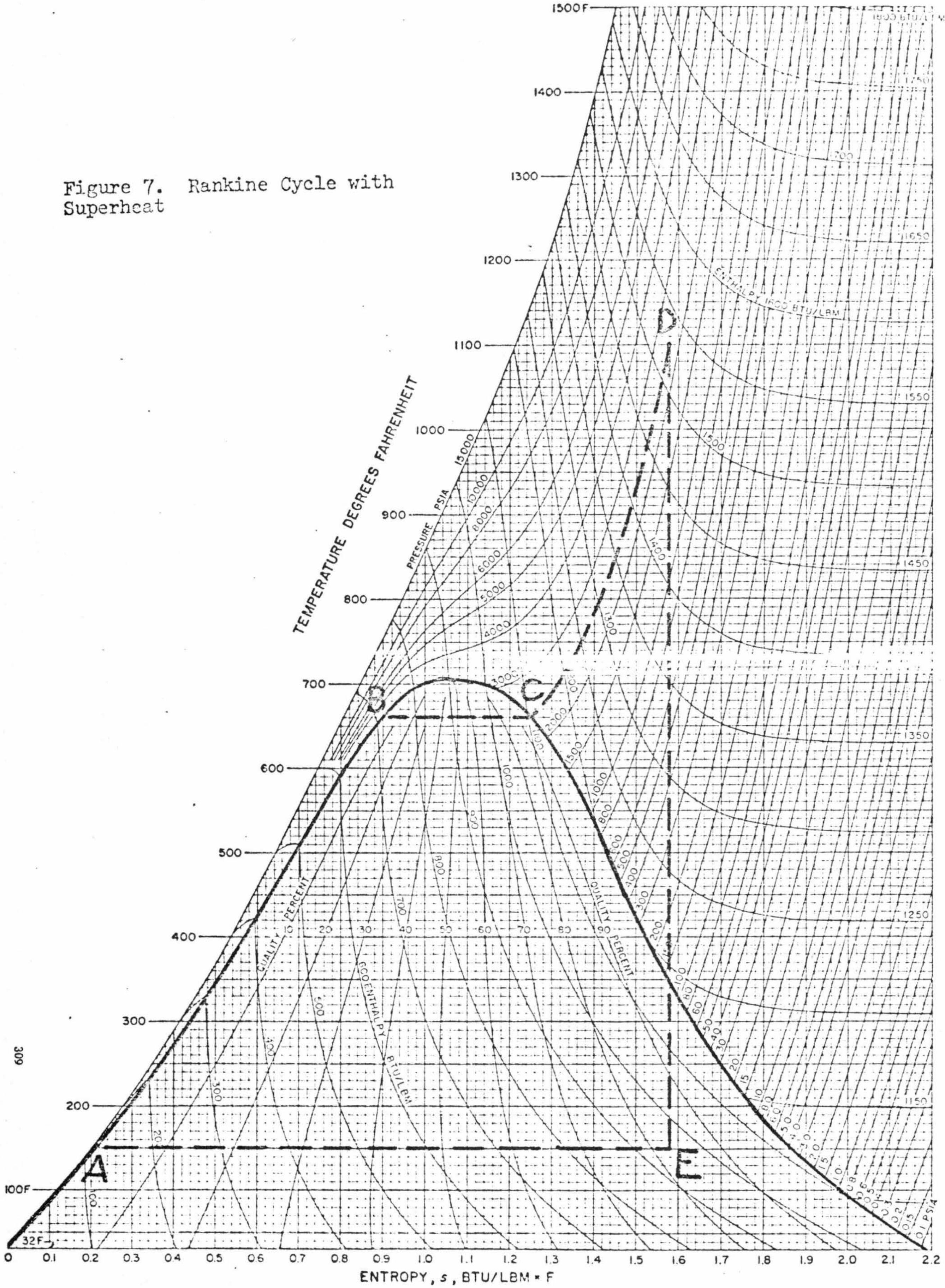


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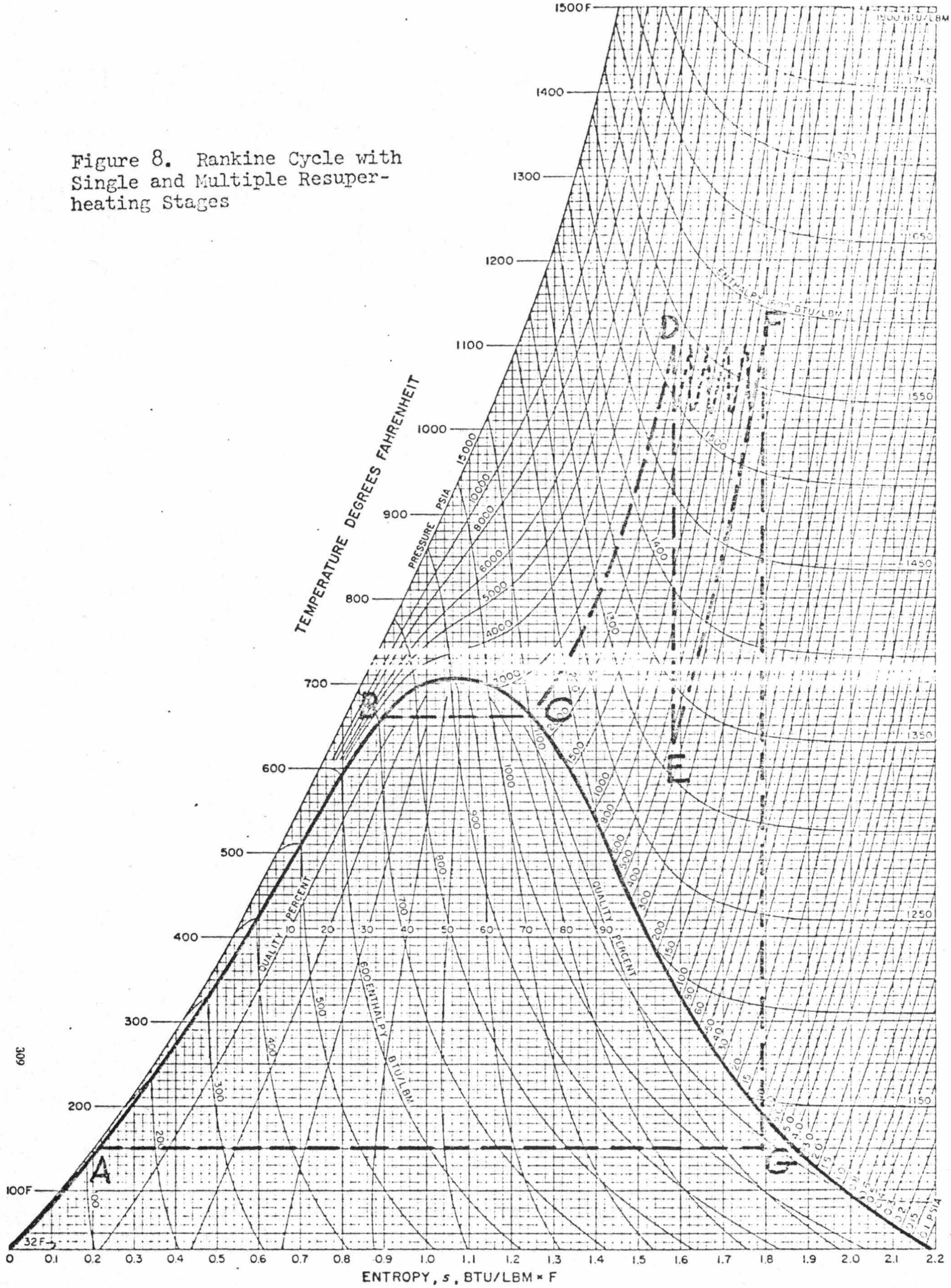
Figure 8. Rankine Cycle with Single and Multiple Resuper-heating Stages



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Non-Conventional Rankine Cycles

From the preceding discussion of conventional Rankine Cycle adaptation to steam power plants, various problems become evident. First, it is seen that regenerative feedwater heating is necessary to minimize the amount of heat addition to the fluid at low temperature. Second, both superheat and one or more resuperheating stages serve to increase the average temperature of heat input, while reducing the fraction of moisture which will develop in the expansion process. It is recognized that all of these additional processes are invented to avoid certain undesirable properties of water; specifically, the specific heat at constant pressure of the liquid phase, the critical temperature and pressure of the fluid, and the slope of the saturation line in the vicinity of the engine expansion line. Knowing the motivations for the various enhancements to the ideal Rankine Cycle, it becomes possible to characterize properties of some hypothetical fluid which might allow a return to a more basic Rankine vapor cycle. The incentives for doing this include a reduction in the cost and complexity of the cycle equipment, and an increase in the efficiency of the cycle.

From strictly thermodynamic considerations, it is recognized that an ideal fluid should have the following properties:

1) The critical temperature should be well in excess of the maximum cycle temperature, so that isothermal heat input may be accomplished at the maximum temperature while at constant pressure.
2) The specific heat at constant pressure of the liquid phase should be low in order to minimize the requirement for costly, and somewhat

irreversible regenerative feedwater heating, or alternatively to minimize direct heat input at temperatures below the maximum.

3) The saturation line in the vicinity of the turbine expansion line should be nearly isentropic. A negative slope, such as that of water, results in excessive moisture. A positive slope, on the other hand, results in superheat of the turbine exhaust. Desuperheating, obviously required prior to condensation, makes heat available at a temperature somewhat higher than that of condensation. If this heat is rejected in the condenser, it represents a loss to the cycle efficiency. Alternatively it may be applied to feedwater heating, but only at considerable expense.

In addition to these strictly thermodynamic considerations, several practical design considerations present themselves as follows:

4) The fluid must be chemically stable at the maximum cycle temperature.
5) The isentropic entalpy drop which is available during expansion should be high, resulting in lower mass requirements and reduced equipment size.
6) The vapor pressure of the fluid at the maximum cycle temperature should be low enough so that material stress levels can be decreased.
7) The vapor pressure of the fluid at the condensing temperature should be high enough to minimize equipment size and problems of ejection of non-condensible gases.
8) The heat transfer properties of the fluid should be good, to minimize the cost of steam generators, condensers, and feedwater heaters.

9) The freezing point of the fluid should be low to eliminate handling difficulties as well as freeze damage to equipment.

10) The cost of the fluid should be low.

The challenge of obtaining useful work from solar energy presents a framework for heat cycle optimization which is quite different from that in which today's steam power cycles have evolved. The maximum temperature limits which are common to conventional steam plants due to metallurgical limitations are undercut by economic considerations which recognize a very high cost for solar heat collection at high temperatures. The well known increase in heat cycle efficiency with temperature must be optimally reconciled with the reverse characteristic for solar collector efficiency. The inevitable decrease in cycle temperature levels, combined with a high cost of heat collection, provide great incentive for the closest possible approach to ideal heat engine efficiency levels. There is no justification for assuming that the techniques developed for present central station energy conversion will be optimum in a solar energy system; therefore very basic investigation of optimum fluids and cycles is warranted.

The Heat Engine -- Principles and Problems

In reviewing the previous discussion it is apparent that present day power cycles are designed to manage the complex task of converting various energy forms by processes that are both economical and socially acceptable. So that conventional power cycles can perform their function they first require a source of heat energy that must be transferred to some motive fluid. Next, and most importantly, the power cycles must be designed to produce useful mechanical work that can be transformed (electrically), transmitted, distributed, and utilized by the end users. Finally, power cycles must reject energy in a controlled manner to cooler environments such as the earth's waters, the earth's atmosphere, or in rarer instances, to outer space. Each of the three elements of a power cycle (heat supply, work output, and heat rejection) possess singular and unique problems in addition to those important problems they share because of their interdependence or relationship within a given power cycle system.

From the point of view of one sub-system within the heat cycle such as an engine that produces useful work ... and in particular prime movers and electric generators .... The concept of useful work is important because it can reveal what design freedoms will exist for the machine builders of the future to harness solar energy systems.

The most direct manner of conceptualizing the problem areas that the future designers of these systems will face is to combine the simple physical definitions of useful work, of mechanical shaft torque, and of electrical torque. These elementary manipulations are developed (see Appendix I) in such a manner that from these physical concepts the basic engineering design parameters for prime movers or heat engines can be delineated.

These important parameters developed in the appendix are expressed in terms of the major prime mover and generator terminal characteristics and appear in Table 1.

Table 1. Prime Mover and Generator System Design Parameters

Prime Mover	Generator
1. Flow Rate (Fluid)	1. Current Flow
2. Inlet State Point (Fluid)	2. Voltage
3. Pressure Ratio (Fluid)	3. *
4. Speed of Shaft	4. Speed of Shaft
5. Efficiency of Conversion	5. Efficiency of Conversion

*The analogy here might be viewed as "ground potential".

The most obvious conclusion drawn from Table 1 is that all prime movers can be completely described in a most elementary manner by means of only five system design parameters. The same conclusion holds for electrical machines. However, in this instance the technology of electrical machines is well developed and may undergo only minor impact. The primary emphasis in this analysis will naturally be directed to the problems of the prime mover, where the state of the art may require the development of corrolary technologies in the application of solar energies.

Since the prime mover will remain the focal point of this study, it is appropriate to define more fully the five system design parameters relating to the prime mover and demonstrate their importance to the engineering design functions.

1. Flow Rate (Fluid) -- γ. The production of prime movers with large power capabilities is critically dependent on large fluid mass flow rates (water, steam, air, ammonia, etc.). Consequently, the primary heat source (input) to the fluid mass must be potentially capable of extrapolation to the large sizes which are necessary for economic power generation.

2. Inlet State Point (Fluid) -- P,T. Traditionally, the thermodynamic conditions of state of a particular fluid (enthalpy, H) determined the energy level (above some reference level) of the fluid. Within the context of this study, the concept has been enlarged to include the totality of physical fluid properties to which a prime mover designer must relate.[4] Pressure (P) and temperature (T), although embodied in the notation (P,T), are intended to include among other properties - the purity of fluid, the chemistry of the fluid and other such factors as fluid radioactivity within which a reference frame can be defined for the fluid entering the prime mover.

3. Pressure Ratio (Fluid) -- ρ. The notion of a pressure ratio relates to the total energy available (H) to the prime mover for conversion to useful work. In an analogous manner, the notion could relate to the condition of the heat sink (i.e.; condenser

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[4] See previous discussion of fluid characteristics, particularly pages 13-14.

pressure) to which level the prime mover must reject the unutilized energy of its motive fluid. It clearly follows that the ideal state is to strive to develop engines with reasonably large pressure ratios (ρ) although this is by no means a limiting consideration to any prime mover design. It is nevertheless an important impediment to most engines if not obtainable.

4. <u>Speed of Shaft</u> -- σ. For commercial power productions in the United States, alternating electricity must be generated at some established frequency, consequently the shaft speeds of both prime movers and the electric generators must be carefully controlled. The discrete acceptable speeds of the electrical machines can be developed according to the following formula.

$$\text{Shaft Rpm} = \frac{120 \times \text{system frequency}}{\text{Number of generator poles}}$$

For the 60 Hz generation systems of the United States this works out to be as shown in Table 2 below for up to ten generator poles.

Table 2. Acceptable Speeds for Prime Mover and Generator Shafts

No. of Poles	Rpm
2	3600
4	1800
6	1200
8	900
10	720

For 50 Hz generation systems, the shaft speeds would be 5/6 of those shown.

5. Efficiency of Conversion --η. The usefulness of any machine, particularly those for power generation, is in large measure determined by the degree that energy inputs can be converted to satisfactory levels of useful work. Consequently, the efficiency of conversion (η) embodies not only the basic efficiency of the heat cycle but the efficiency of the engine (and generator) components through a wide spectrum of loads or operating modes.

In summary, a prime mover viewed as a sub-system of a power network incorporates each of the five system design parameters described above. It must be understood that all prime movers are designed only after making a descrete choice for each of the design parameters (γ, P,T, ρ, σ, η). This does not imply that prime movers cannot be "operated" within different ranges of γ's, ρ's, σ's, etc. -- it implies that only one descrete choice from each parameter is permitted in the "design" process.

Since the "design" process may be viewed as a process that translates the design parameters to physical dimensions (drawings, hardware, etc.) the following relationship must hold:

$$\text{Physical Dimensions of Prime Mover} = f(\gamma, PT, \rho, \sigma, \eta)$$

A somewhat clearer understanding could be developed if the prime mover was restricted to a turbine possessing n stages. Each turbine stage would be designed to its unique set of design parameters so that the equation would be restructured as follows:

$$\text{Physical Dimensions of Turbine} = \sum_{i\,=\,\text{Stage 1}}^{\text{Stage n}} f(\gamma, PT, \rho, \sigma, \eta)$$

The above equation should leave no doubt that turbine stage No. 1 and the last turbine stage, No. n, must be designed for two widely differing environments. But the sum, comprising stage No. 1 to n, will actually control the basic dimensional requirements of the turbine.

The usefulness of the above concept is not restricted to heat engines and holds even if applied to pumps, compressors, or other rotating machine employing fluids. This factor will be true so long as the equations shown allow for both positive and negative values of the design parameters. For example, a turbo-compressor could be viewed as having a "negative" pressure ratio ($-\rho$), flow rates ($-\gamma$), and perhaps efficiency ($-\eta$). Similar analogies could be developed for pumps and hydro-electric turbines with little difficulty.

The major usefulness of the concept embodied in the notation $f(\gamma, PT, \rho, \sigma, \eta)$ is that product life and product costs also fall within this conceptual framework and are clearly related to the choices required by the system.

Since the prime mover will be subjected to a wide range of forces and stresses during its operation, i.e.; centrifugal, thermal, pressure, vibratory, etc., the physical dimension of the equipment will be physically strained to accommodate the new equilibrium state of the forces acting. This change in physical dimensions could be expressed as:

$$\Delta f(\gamma, PT, \rho, \sigma, \eta) = \text{equivalent strain}$$

Since these strains must occur or must exist over some finite time periods ($\Delta \theta$), it is more proper to introduce time (θ) into the above equation so that it can be expressed as:

$$\frac{\Delta f}{\Delta \theta} (\gamma, PT, \rho, \sigma, \eta) = \text{strain/time interval}$$

The concept of equipment lifetimes can be viewed as the frequency (number of cycles) a particular component has been subjected to certain levels of strain. This process is usually referred to as the "duty cycle" of the equipment but poses the most severe problem to the designer. First, he must make an estimate (at best) on how the equipment will be operated and second, after estimating the "duty cycle", make a decision on the economic life-cycle of the particular equipment (should it last 120 seconds as a space rocket or 30 years as a nuclear turbine).

The last decision enumerated is crucial to the entire design process as it will have a most profound effect on material selection. Ultimately, the product cost and the economic viability of the power generation system (energy cost) must be affected.

If for the sake of simplicity, (N) is assumed to be the number of starts and stops the prime mover (neglecting load changes) will be subjected to during its lifetime, then we can view the equipment duty cycle as follows:

$$\frac{N}{\Delta \theta} \Delta f(\gamma, PT, \rho, \sigma \eta) = \text{duty cycle}$$

Finally, and most important, the relative economic costs can be related to any prime mover (for a given duty cycle) by returning to the original equation for the physical dimension of the prime mover, but assuming the cost function of the engine relates to some exponential function of the engineering constraints as follows:

$$\text{Cost} = K \cdot f(\gamma^{a}, PT^{b}, \rho^{c}, \sigma^{d}, \eta^{e})$$

where K is an arbitrary constant

A clear understanding of the above short discussion is crucial to a review of solar energy applications for electrical generation. Although the theoretical principles of mechanics and thermodynamics must be rigorously observed, the above practical philosophical and conceptual framework within which all heat engines or prime movers must ultimately be measured against must be fully appreciated.

Unfortunately it is not enough to describe ideal cycles, ideal fluids, perfect engines, etc. -- real cycles, real fluids and practical engines will eventually be required to perform the work.

The Heat Rejection System -- Its Characteristics

As demonstrated in the previous sections of this report, a simple conceptual framework was established to permit the observer to view the prime mover in terms of five unique engineering parameters or characteristics that influenced the major system dimensions and ultimately its economic costs.

In an analogous manner, the sub-system that must manage the heat rejected from a particular heat engine must be treated similarly. Although the heat rejection sub-system has been already described in preliminary detail by W. H. Comtois[5] in his analysis of heat exchanger designs, it is still useful to observe the differences in heat engines and heat rejection systems when developed by applying the methods and reasoning appearing in the previous section.

The major difference is immediately apparent in that a heat exchanger has no motion ($\sigma = 0$) and therefore does no useful work. But beyond that point, there remain few significant differences.

Consequently, for the heat rejection system, the physical dimensions appear to become a function of only three parameters:

$$\text{Physical Dimensions (surface)} = f(\gamma, PT, \Delta H) \quad *$$

The notions (γ), (PT), and (Δ H) hold essentially the same meaning as in the previous section. However, some modification must be made to the above equation that indicates that heat is flowing from one fluid medium (perhaps hot steam) to another medium (perhaps cold water or air). This factor could be expressed by rewriting the above equation in the following form:

$$\text{Physical Dimensions} = f'(\gamma, PT, \Delta H) + f''(\gamma, PT, \Delta H)$$

Thus it is easily observed that all the usual heat exchanger design data required such as heat flows, friction losses, coolant flows, etc. fall with the notion expressed in the above function. Furthermore, as

*The substitution of (Δ H) for (ρ) represents more accurately the concept of temperature changes in fluids transporting heat.

[5] W. H. Comtois, "Heat Exchanger Design", Appendix A of this report.

in prime movers, the question of "product life" is intimately associated with the question of material selection and ultimately product cost. The <u>conscious</u> choices that equipment designers do make, although recognized as "experience", "empiricism", design optimization", etc. are, as noted by W. H. Comtois, crucial. There is little doubt, however, that the effectiveness of the apparatus is greatly enhanced whenever research or development can substitute for the empiricism or experience so common to this type of equipment.

Summary

This report has attempted to provide insight into certain problem areas that are relevant to the commercial production of electricity using the sun's energy. The specific areas under discussion included the characteristics of thermodynamic heat cycles and fluids and the characteristics of heat engines including their companion heat rejection systems.

An effort was made to develop a general conceptual framework whereby the major engineering parameters affecting equipment configurations and economic costs could be viewed as a part of an organized and logical system of technical design.

In addition, the report implicitly suggests the importance of clearly defining both the inputs and the outputs from all sub-systems so that the final operating system can be successfully synthesized.

Appendix

The elementary analysis below is intended to demonstrate how the basic engineering parameters affecting both the prime mover and generator can be developed.

1 ... Work = Power x Time

2 ... Power = Torque x Speed of Shaft

3 ... Power = Fluid Flow Rate x Energy Available x Efficiency

4 ... Energy Available = f (Pressure Ratio and Inlet State Point of Fluid)

5 ... Power = Fluid Flow Rate x f (Pressure Ratio and Inlet State Point of Fluid) x Efficiency of Conversion

By equating equations 2 and 5 and solving for torque, all the machine characteristics appear in one equation:

6 ... $\text{Mechanical Torque} = \dfrac{\text{Fluid Flow Rate}}{\text{Speed of Shaft}} \times f(\text{Pressure Ratio and Inlet State Point of Fluid}) \times (\text{Efficiency of Conversion})$

Using a similar approach but introducing generator terminal characteristics:

7 ... $\text{Electrical Torque} = \dfrac{\text{Power}}{\text{Speed of Shaft}}$, or

8 ... $\text{Electrical Torque} = \dfrac{\text{Current} \times \text{Voltage} \times \text{Efficiency of Conversion}}{\text{Speed of Shaft}}$