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HYDROGEN TURBINE POWER CONVERSION SYSTEM ASSESSMENT

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16. Abstract

A three-part technical study was conducted whereby parametric technical and economic feasibility data were developed on several power conversion systems suitable for the generation of central station electric power through the combustion of hydrogen and the use of the resulting heat energy in turbogenerator equipment.

The study assessed potential applications of hydrogen-fueled power conversion systems and identified the three most promising candidates: (1) Ericsson Cycle, (2) gas turbine, and (3) direct steam injection system for fossil fuel as well as nuclear powerplants. A technical and economic evaluation was performed on the three systems from which the direct injection system (fossil fuel only) was selected for a preliminary conceptual design of an integrated hydrogen-fired power conversion system.

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FOREWORD

The work herein was conducted from August 1973 to April 1976 by personnel from the Advanced and Propulsion Engineering and Engineering Test units at Rocketdyne, a division of Rockwell International, under Contract NAS3-20388. Mr. Larry H. Gordon, Lewis Research Center, was NASA project manager. At Rocketdyne Mr. David E. Wright, Program Manager and Mr. A. D. Lucci, Project Manager, were responsible for the direction of the program.

Important contributions to the conduct of the program and to the preparation of the report material were made by the following Rocketdyne personnel:

Combustion Devices	Mr. J. Campbell
Analysis	Dr. J. C. Lee

Important contributions were made by the following team members:

- Institute of Gas Technology
- Commonwealth Edison
- Ralph M. Parsons Company
- Atomics International Division of Rockwell International

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SUMMARY

A three-part study was conducted to evaluate the feasibility of using hydrogen and oxygen or air combustors in power conversion systems (PCS) for the generation of electricity. The three tasks in the study were organized to perform a technical and marketing evaluation of several PCS's in Task 1, and to select the most promising three for evaluation in Task 2. The effort in Task 2 was to perform additional technical analysis of the selected PCS, including preliminary combustor design, and to perform an economic evaluation to determine cost of electricity (COE). At the conclusion of Task 2, one of the PCS's was to be selected for the Task 3 effort, which was to perform a plant conceptual design. The study results show that several applications for H_2/O_2 or air combustor are attractive. The supplementary steam generation (SSG) application is a near-term system that has the potential to solve current problems at a low development cost and low capital and operating cost.

TASK 1 - IDENTIFICATION OF POTENTIAL APPLICATIONS AND DEFINITION OF CANDIDATE PCS

The PCS's identified in Task I were as follows:

(1) Retrofit Systems (Supplementary Steam Generation)

This involved replacing old or worn out boilers, still having serviceable turbogenerating equipment, with a H_2/O_2 combustor. This application was found to be attractive but, in review with our consultants, principally Commonwealth Edison, a modification of this application was developed called the supplementary steam generation (SSG). In this concept, a H_2/O_2 combustor would supply makeup steam directly to a system that has been derated for one of many reasons, including emission control requirements, fuel changes, or boiler loss of efficiency.

(2) Nuclear Superheating Systems

a. A direct inspection of H_2/O_2 combustion steam for pressurized water reactor (PWR) systems

b. An indirect superheating by means of a H_2/O_2 or air combustor for boiling water reactor (BWR) systems

(3) Advanced Steam Systems

a. The Ericsson cycle incorporating several reheats (reburns)

b. A high-temperature steam cycle with turbine temperature range to 2000 F

c. A modified high-temperature cycle with partial condensing

(4) Gas Turbine Systems

a. Simple Brayton with and without regeneration

b. Brayton with regeneration and intercooling

c. Combined cycle (steam bottoming)

All systems examined were found to be technically feasible. The advanced steam cycles and particularly the Ericsson cycle produced the highest efficiency, but were the most complex or required signficiant advances in present state of the art. Both the retrofit and SSG applications improved the efficiency of the system being modified. The simple cycle gas turbine applications produced the lowest efficiency, but this was improved significantly when combined with a steam bottoming cycle.

The marketing effort of this task indicated no limitations to using a H_2/O_2 combustor.

The environmental assessment revealed possible problems with the disposal of excess and potential radioactive water in the nuclear applications. Formulation of NOX in the gas turbine application was also identified as a potential problem. No other polution problems were identified. No operational problems beyond the present state of the art in handling and using hydrogen were uncovered.

Output

The analyses of Task 1 eliminated all PCS's except the Ericsson cycle, the gas turbine and direct steam injection systems, which include both PWR (as required by contract) and SSG.

TASK 2 - TECHNICAL ANALYSIS AND ECONOMIC EVALUATION

This effort required a system analysis, preliminary conceptual design of the combustor, and an economic study resulting in an estimate of the COE. Heat and mass balances of the three PCS's were performed. Flow diagrams and schematics were prepared. The sizes of the combustors for each application were determined. Combustor cooling requirements were established. The cost of retrofit or, as in the case of the Ericsson cycle, the cost of the new plant was determined. Installation times were estimated and a final COE determined.

The basic result was that the SSG and gas turbine applications produced the lowest COE. Direct steam injection for a PWR system produced a high COE primarily because the application necessitates either base load operation or a second turbogenerating set with complex valving for peaking application. Either of these limitations add to the COE.

The Ericsson cycle also produced a high COE. The cost of fuel is the major driver in the COE of this application (when compared to a coal-fired plant for baseload and SSG or gas turbines for peaking. The high efficiency of this cycle would make this application a contender should hydrogen costs be reduced or coal costs be increased.

Output

It was recommended and approved that the supplementary steam application be continued in Task 3. This was done on the basis that this application has near-term capability, is a potential solution to several emission control and fuel conversion problems, is cost competitive, and is readily demonstratable at low cost.

TASK 3 - PLANT CONCEPTUAL DESIGN

A preliminary plant conceptual design was performed and an R&D program proposed.

The combustor system was defined, including controls for operation and for minimizing temperature shock. The plan system was described showing possible combustor locations.

The R&D program describes a plan from combustor system development to site preparation and combustor installation and proof of concept testing. This plan can be to the point of testing within 3 years.

Output

No unsolvable problems were uncovered in the conceptual design installation of the SSG in a typical fossil fuel plant.

CONCLUSIONS

- A market exists for all potential uses of the H_2/O_2 combustor.
- The largest market exists for direct steam, i.e., supplementary or steam boiler replacement systems
- Highest efficiency gains result from the Ericsson cycle, advanced steam system.
- Hydrogen costs limit its use in most cases to peaking applications.
- Pressurized water reactor and gas turbine systems using air have problems of potential environmental pollution.
- The lowest COE is obtained from gas turbine and supplementary steam generation.
- Supplementary steam generation offers a solution to plant generating capacity loss due to emission control requirements, fuel conversion, or boiler efficiency losses.
- Supplementary steam generation offers incentive to convert to coal from gas or oil without loss of generating capacity, with early payback for conversion costs.
- A supplementary steam generator demonstration is possible in the least amount of time at the lowest cost.
- The commercial and space program records of safety in the use of hydrogen affirms that minimum operating and handling problems will occur with systems using hydrogen as a fuel.

INTRODUCTION

This report is submitted in accordance with the requirements of Contract NAS3-20388. It describes an 8-month technical study to analyze potential systems which utilize hydrogen as a fuel for the generation of electrical power.

A number of studies by both governmental agencies and independent utilities and study groups have indicated that hydrogen has attractive potential advantages as a fuel for the generation of central station electrical power. The subject is complex in that there are many questions to be resolved in the areas of technical feasibility, economics, safety, and environmental impact, such as the production of the hydrogen, its transmission and storage, and its utilization in the generation of electrical power. The purpose of this report is to provide technical and economic feasibility data on potential power conversion systems (PCS) that utilize turbogenerators for the production of the electrical power. The study is concerned with only a portion of the spectrum of technical and economic factors that will ultimately govern the utilization of hydrogen for central station power generation. In the subject study, the economic and technical feasibility of candidate PCS's is assessed relative to competitive systems on the basis of parametric assumptions on the cost of hydrogen and other consumables, i.e., hydrogen fuel cost is not derived in the study. The study results are intended to be utilized in combination with the results of other studies that will define the economic and technical feasibility of the hydrogen production and transmission/storage systems.

The fundamental objective of this program is to provide parametric economic and technical feasibility data on several PCS's suitable for the generation of central station electrical power through the combustion of hydrogen and the utilization of the resulting heat energy in turbogenerator equipment. In connection with this fudamental objective, one may define specific subobjectives as:

1. The identification of potential applications for the generation of central station power through the combustion of hydrogen, and the definition of candidate power conversion systems

2. A preliminary technical analysis and a preliminary economic evaluation of those applications selected as most promising

3. The conceptual preliminary design of an integrated hydrogen-fired PCS selected from among the systems studies in subobjective 2 above.

The approach to achieving these objectives was to conduct an 8-month technical study consisting of a well-defined series of sequential technical tasks, with consultation and review with NASA occurring upon the completion of each task to obtain approval prior to initiation of the subsequent task. The details of the work accomplished are described in this report.

The study deals with the technical, (including environmental effects), and economic aspects of the utilization of hydrogen as a fuel in hydrogen combustor/turbine components for utilization in the central station generation of electricity. While this is only a portion of the spectrum of technical and economic aspects of the "hydrogen economy", a wide range of technical and economic disciplines is involved.

The Rocketdyne Division of Rockwell International led the study effort and provided the major inputs, on the technical and economic feasibility of the new technologies that were required in the area of hydrogen combustion and heat transfer equipment. A group of consultants participated in the study and critiqued the results on the basis of their special expertise. The Ralph M. Parsons Company, an architect and engineers firm, provided expertise on overall powerplant factors and on the effect of hydrogen fuel-burning equipment upon the cost and technical feasibility of other portions of the powerplant. The Institute of Gas Technology provided state-of-the-art and background information on the costs and constraints on the hydrogen fuel likely to be available for combustion in the studied installations. the Commonwealth Edison Company provided the viewpoint of a major utility on the feasibility of the concepts studied. The Atomics International Division of Rockwell International provided the viewpoint of a major developer and manufacturer of nuclear and other power generation equipment. The role of each of the consultants was to provide the study input information required, in addition to the information generated by Rocketdyne on the combustor and heat transfer equipment, to evaluate the technical and economic feasibility of each concept. Additionally, the consultants participated in reviews of the progress of the studies, adding the benefit of their specialized knowledge and background to periodically critique the work accomplished.

The technical feasibility of the concepts was evaluated largely on the basis of the present state of the art in combustion, heat transfer, materials utilization, and environmental effects. As a part of the evaluation of the technical feasibility, it was also intended to identify technical development requirements. The economic feasibility of the concepts examined were a function both of the equipment costs and the costs of the hydrogen and other input streams. The economic evaluations are expressed in many cases in terms of the differential costs, i.e., the difference in cost/kW-hr of electricity of power produced with hydrogen fuel as against the cost of a competitive system with conventional fossil and nuclear fuels. Additionally, electrical power costs are expressed parametrically in terms of various costs of hydrogen and other input streams, so that economic feasibility assessments may be tied in with the results of other studies on the cost of production, transmission, and storage of such fuel streams.

This study was conducted according to the following plan. The first task involved the identification of potential applications of hydrogen combustor/turbine power conversion systems and the definition of their content. The six potential applications identified by NASA, and additional applications identified by the contractor, were studied for technical and market feasibility. The results of the study were reviewed with the contractor team and with NASA with three of the candidate applications selected for a more detailed analysis and evaluation. The second task consisted of a more detailed parametric analysis and evaluation of the hydrogen applications selected under Task 1. The capacity range studied was as appropriate for the type of application, i.e., peaking, intermediate, or baseload operation. On the conclusion of the second task, another review by the contract team and NASA resulted in the selection of a single candidate application for further study. This application, the supplementary steam generation (SSG) system was studied in Task III and resulted in the preparation of a conceptual design of the hydrogen combustor/turbine power conversion system integrated with a complete central station powerplant.

STUDY OBJECTIVES AND CRITERIA

The work conducted on this contract was organized into three tasks, all concerned with power conversion subsystems which convert the energy content of hydrogen (H2) into electric power via combustion, and the routing of the combustion products through turbomachinery.

Task 1 of the program was concerned with the assessment and definition of particular applications of candidate power conversion systems (PCS). For this study, the production and storage of H_2 and/or oxygen (O_2) were assumed to have been satisfactorily resolved. It was recognized however that both H_2 and O_2 may be produced in a wide variety of purities, depending upon the production process and, in each instance, the interrelation between system technology and H_2 and/or O_2 property specifications was established.

Several factors were evaluated in examining the likelihood for successful application of any particular candidate H_2/O_2 PCS. The fundamental economics of the application were a prime consideration, and this involved, as well, the projected size of the market for the H_2 PCS application. The question of the applicability of the particular PCS to the various duty cycles encountered in a central station power supply is also important. Those systems that are capable of rapid and repeated startup and shutdown have the advantage of being applicable to "peaking" installations as well as to intermediate and baseload installations. The economics and competitive position of H_2 PCS's relative to alternative systems are thus a function of the duty cycles available.

The technology sophistication required for each H_2 PCS is also important in evaluating the likelihood that the PCS system can attain maturity and commercial operation. The polluting effects of the H_2 PCS cycle are of interest in this regard as well as the questions of operability and safety of plant operating personnel. Pollution standards and requirements are expected to become stricter with the passage of time and to have a greater impact on station and cycle practice than the very substantial impact which exists today. The influence of cycle thermodynamic efficiency, i.e., KWH produced per unit of energy input, is automatically included in the evaluations in the areas of economic competitiveness and potential marketability. Additionally, cycle efficiency was evaluated in this study as a separate entity on the basis that international political developments over the long term may well place a premium on cycle efficiency attainment beyond its influence in the competitive market place.

The candidate PCS's assessed during Task 1 and listed in Table 1 may be organized into several classes as described below.

The light water reactors, (i.e., ordinary water), which almost entirely monopolize the generation of nuclear power in the United States, are presently limited to relatively low steam pressures and, more importantly, low steam temperatures, by the limitations of the nuclear reactors. One can project significant improvements in overall cycle efficiency by the separate addition of superheat to the steam, leaving a light water reactor. Two concepts involving the provision of such superheat by the combustion of H_2 were examined in this effort, the pressurized water reactor (PWR) and the boiling water reaction (BWR).

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TABLE 1. PCS'S REVIEWED

- PWR Direct Superheater
- BWR Indirect Superheater
- Steam Boiler Retrofit Cycle
  - Conventional Fossil Steam Plant Replacement
  - Supplementary Steam Generation (SSG)*
- High-Temperature Steam Turbine Cycle
  - 24.13×10^6 N/m^2/1366 K/1366 K
  - (3500 psi/2000 F/2000 F)
- Partial-Condensing Rankine-Brayton Cycle*
  - 24.13×10^6 N/m^2/1366 K/1366 K
  - (3500 psi/2000 F/2000 F
- Steam Ericsson Cycles
  - 24.13×10^6 N/m^2/811 K/811 K
  - 3500 psi/1000 F/1000 F/1000 F/1000 F)
  - 24.13×10^6 N/m^2/1366 K/1366 K
  - (3500 psi/2000 F/2000 F/2000 F/2000 F)
- H_2 Gas Turbine Cycles
  - Regenerative High-Temperature 1922 K (3000 F) Open Cycle
  - Regenerative Partial-Closed Cycle 1366 K (2000 F)

*These PCS's were added to the study by Rocketdyne

PWR DIRECT SUPERHEATER

The PWR operates in a mode whereby the steam utilized in the power cycle does not pass through the reactor; the steam is generated by exchange of heat within a completely separate high-pressure water loop that passes from the reactor through the steam boiler and back to the reactor. In this manner, the radiation contamination of the steam in the turbogenerator system is minimized. The hydrogen combustion PCS evaluated for this system superheated the steam by the direct combustion of H_2/O_2 in the steam line, under the pressure existing therein.

BWR INDIRECT SUPERHEATER

The BWR is arranged so that the steam that drives the turbine is produced by boiling water in the reactor itself, and this steam is inevitably contaminated with radioactive material. The excess condensate produced by the direct combustion of the H_2/O_2 in the steam line as a superheat would be expected to contain sufficient radiation activity to constitute an unacceptably expensive disposal problem. One can visualize, however, an appropriate system for the addition of superheat by hydrogen combustion that would be applicable to either the BWR or the PWR, and this is by using H_2/air or H_2/O_2 as fuel for a separately fired tubular superheater. This overcomes the problem of radioactive condensate by keeping the products of combustion separate from the steam produced by the nuclear steam supply system. Such a H_2 PCS system was the second candidate evaluated in Task 1.

In addition to the application of H_2 combustion to superheat nuclear steam, is one in which such combustion would be utilized to replace boilers of conventional fossil fuels, i.e., coal, oil, or natural gas.

RETROFIT (CONVENTIONAL FOSSIL STEAM PLANT)

One system examined in Task 1 uses an H_2 combustor to replace the obsolete and inoperable boiler of a conventional fossil fuel-fired steam powerplant. The obsolete boiler is replaced by a steam supply system in which H_2 and O_2 would be burned directly to produce high-temperature steam, and that steam would be immediately diluted and tempered by the addition of spray water, to produce the quantity, pressure, and temperature of steam required for the turbogenerator of the existing power station. Such a system has the advantage of providing electrical power at a minimum capital cost, in that the turbogenerator system, electrical installation, etc., already exist. Peaking, intermediate, or baseload power supply might be attempted depending upon the cost of the H_2/O_2 fuel supplies and the technical characteristics of the particular station involved.

RETROFIT (SUPPLEMENTARY STEAM GENERATION, SSG)

A variation on complete replacement of an existing fossil-fired boiler by the direct generation of steam from H_2/O_2 is the concept of supplying supplementary steam by this method. Such supplementary steam finds application during periods of peak loads at stations whose steam generating capacity has been restricted by environmental considerations to some value less than the capacity of the turbogenerator set. Restricted boiler capacity also may arise in cases where a conversion has been made to coal from oil or gas firing, and the capacity on coal firing is less than that available on oil or gas. The application of the H_2 PCS to these situations is attractive in terms of supplying peaking power at a minimal capital investment. The application can also be visualized in which new coal-fired stations would be provided with additional turbogenerator and electrical station capacity over and above that required for the coal-fired boiler, along with a H_2/O_2 direct-fired steam generator. The incremental cost of the additional turbogenerator capacity and H_2/fired PCS might then be low enough to make peaking power generation via H_2 an attractive economic situation.

The operating steam pressure and temperature conditions in current central station technology are determined at least in part by limitations imposed by boiler and steam piping material. The single reheat 811 K/811 K (1000 F/ 1000 F) steam temperature condition has up to now been the standard of the industry for many years. When one considers direct combustion of H_2 and O_2 to produce steam, the boiler and piping limitations no longer apply. The way is opened to examine power cycles providing substantial improvements in cycle efficiency. Several such cycles were identified and examined during Task 1 of this contract.

ADVANCED STEAM CYCLES

These cycles included advanced high-temperature steam turbine cycles operating at pressures up to 24.13×10^6 N/m^2 (3500 psi) and steam temperatures up to 1366 K (2000 F). Additionally, "partial condensing" steam Rankine/Brayton cycles were examined which are hybrid cycles combining aspects of the Brayton and Rankine cycle with steam as the working fluid. Finally, Ericsson steam cycles with multiple reheats were evaluated at several steam pressures and temperatures.

GAS TURBINE CYCLES

All cycles so far discussed have been concerned largely with variations on the conventional Rankine steam cycle. It seems evident, however, that H_2 will also be an interesting fuel for application to gas turbine, i.e., to Brayton cycle power generation power systems. A number of such possibilities were identified and evaluated during Task 1, including open-cycle gas turbines with H_2/air firing, combined cycles, and high-temperature fluid working cycles.

The objectives of Task 2 of this effort were to analyze and evaluate parametrically the three H_2 applications that had been selected out of the larger number of concepts evaluated in Task 1. This technical-economic assessment was planned initially to cover power ranges from 10 to 100 MWe and to include the following parameters:

1. System thermal cycle efficiency as a function of (a) component performance, (b) turbine inlet temperature, and (c) reburn applications
2. Estimates of physical size and cost of all components of the PCS
3. Cost perturbations as a function of (a) plant use factor, and (b) modular equipment installation
4. Environmental impact assessment
5. Qualitative and quantitative benefits
6. Estimates of capital, operating, and maintenance costs with appropriate contingencies for developmental processes and uncertainties
7. R&D requirements to bring the concept to commercial readiness, including cost and time through evaluation via a demonstration plant

The three power conversion systems which were selected out of the Task 1 array included:

1. A representative advanced steam cycle; in this case, the Ericsson cycle, providing both high steam pressure and temperature and multiple reheats
2. An open-cycle gas turbine combined with steam bottoming for improved cycle efficiency
3. A supplementary cycle, in which H_2 and O_2 are burned to generate additional steam or to provide additional superheat for an already established steam flow

The No. 3 system represents not a single H_2 PCS but a class of PCS's including superheaters for pressurized water reactors and the provision of direct-fired steam generators or superheaters to supplement fossil fuel-fired boilers.

In the initial evaluation of these cycles, it quickly became apparent that it was not reasonable to limit the power range to 10 to 100 MWe, and this limitation was subsequently ignored in favor of selecting the PCS size applicable to the system.

For this study, the production, distribution, and storage of H_2 and/or O_2 was assumed to have been resolved satisfactorily. The H_2 PCS was assumed to be supplied with H_2 (and, when necessary, with O_2), in gaseous form via pipeline to the plant boundary. In evaluating the relevant economics of the various concepts considered, however, it was necessary to have some relative values for the costs of H_2 and O_2 as compared to alternative fuels. Information of this nature was obtained from the open literature and from the Institute of Gas Technology (IGT), one of the consultants on this effort. The H_2 and O_2 costs utilized in this study are presented in Fig. 1, 2, and 3.

The cost of H_2 via electrolysis is presented as a function of the cost of electricity. Many present-day estimates of the cost of electricity produced from newly erected nuclear or coal-fired stations, utilizing coal at roughly $1.00/ per million Btu, or nuclear fuel at 25¢/million Btu, result in estimates of the cost of electricity in the range of 3 to 4¢/kW-hr. As can been seen from Fig. 1 , the major cost of energy in the form of H_2 when electrical costs are in this range is the cost of the electricity itself. Note also that these estimates are based on utilizing an advanced system for electrolysis and that the energy efficiency of the electrolyzer is relatively high, i.e., the higher heating value of the H_2 produced represents about 85% of the energy contained in electricity that went to produce the H_2.

The cost of H_2 produced by chemical processing from a coal feed stock is presented in Fig. 2 . These data are based on information supplied by IGT, and show performance expected from current processes as well as the improved performance expected from advanced processes. Note that the energy efficiency of these processes is on the order of 55%, i.e., the higher heating value of the H_2 produced is about 55% of the higher heating value of the coal that was needed to produce the H_2. Notwithstanding the lower energy efficiency of coal gasification as compared to electrolysis, the cost of H_2 produced at a coal cost of $1.00/million Btu are expected to be substantially less than the cost of electrolytically produced H_2 with the same coal costs. The explanation lies in the relatively low efficiency of conversion-of-coal-contained energy to electricity, which is on the order of 35%, which must then be converted to H_2 by electrolysis at an energy efficiency of 85%. The combined generating station/electrolysis energy efficiency (coal to H_2 via electrolysis) is then on the order of 30% as compared to the 55% for gasification. Figure 2 also presents for some typical compositions of H_2 produced via coal gasification. There are substantial volumes of noncondensible gases that will be produced by the combustion of H_2 derived from coal gasification, and it is necessary to give attention to these noncondensible gases in the economic and technical analyses of H_2 PCS's utilizing such H_2.

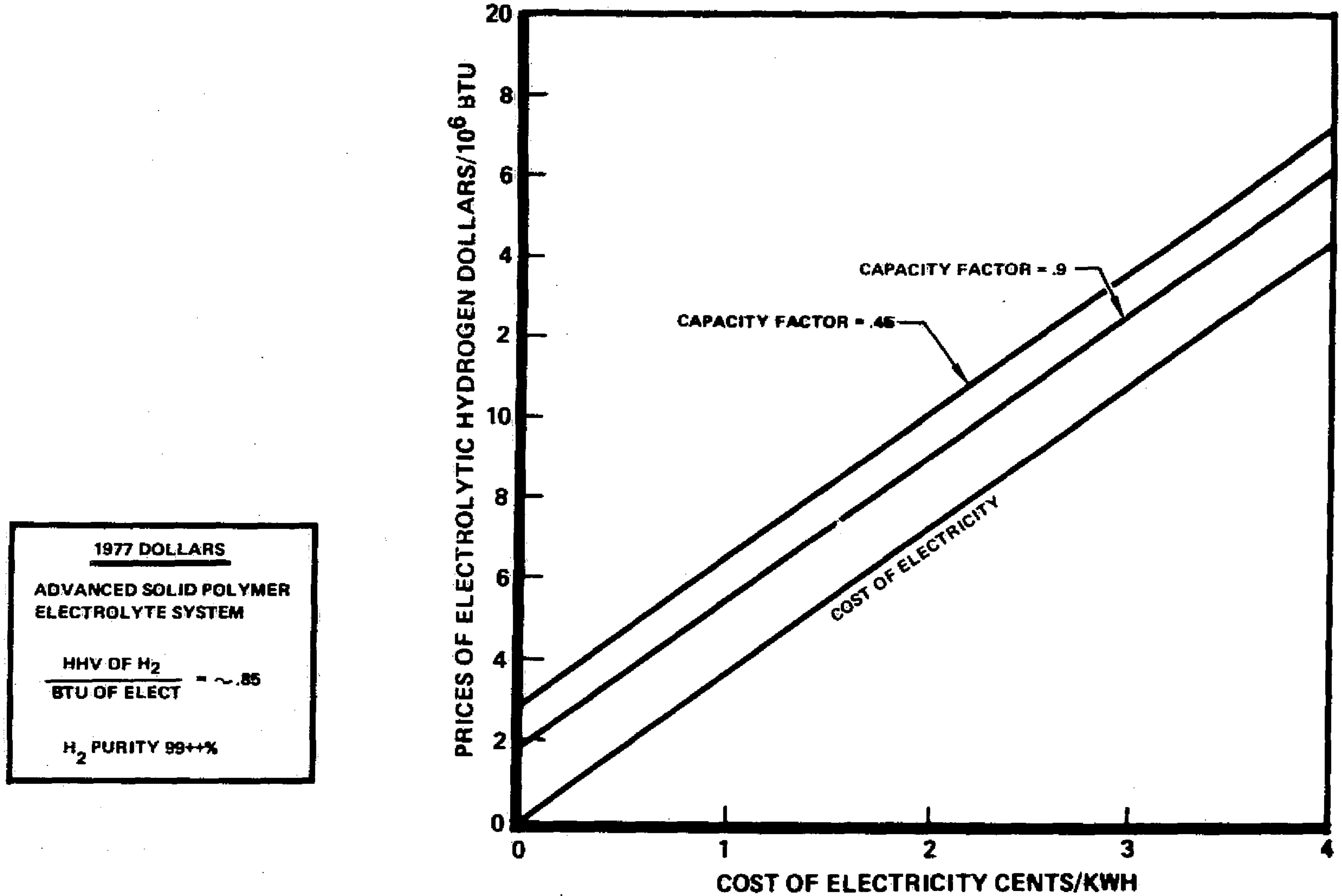
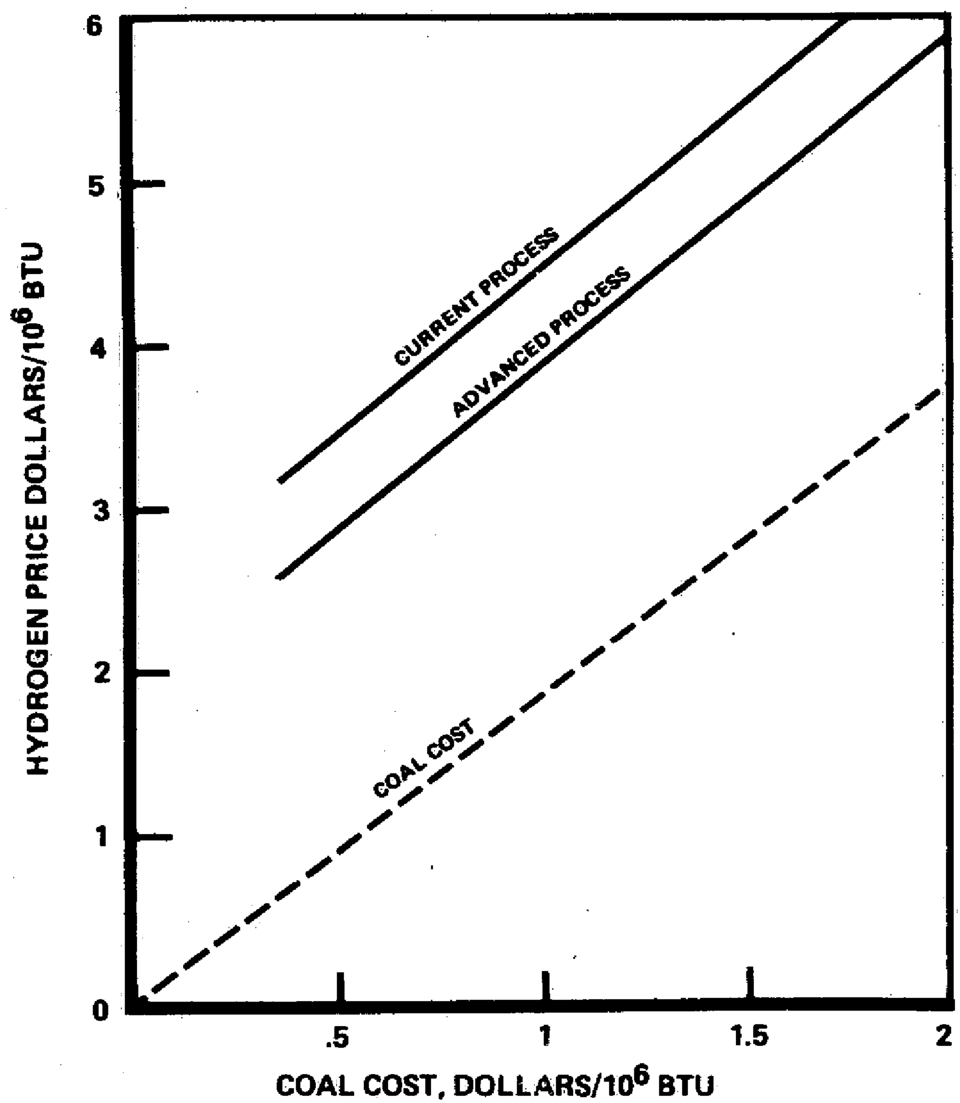


Figure 1. Hydrogen by Electrolysis (Concurrent Oxygen)



1977 DOLLARS

TYPICAL COMPOSITION

	% BY VOLUME	
	K/TOTZEK	U-GAS
CO	0.1	0.1
H_2	93.1	94.3
CH_4	5.5	4.8
N_2 & AR	1.3	0.8

ENERGY EFFICIENCY ~ 55%

Figure 2. Hydrogen From Coal Gasification

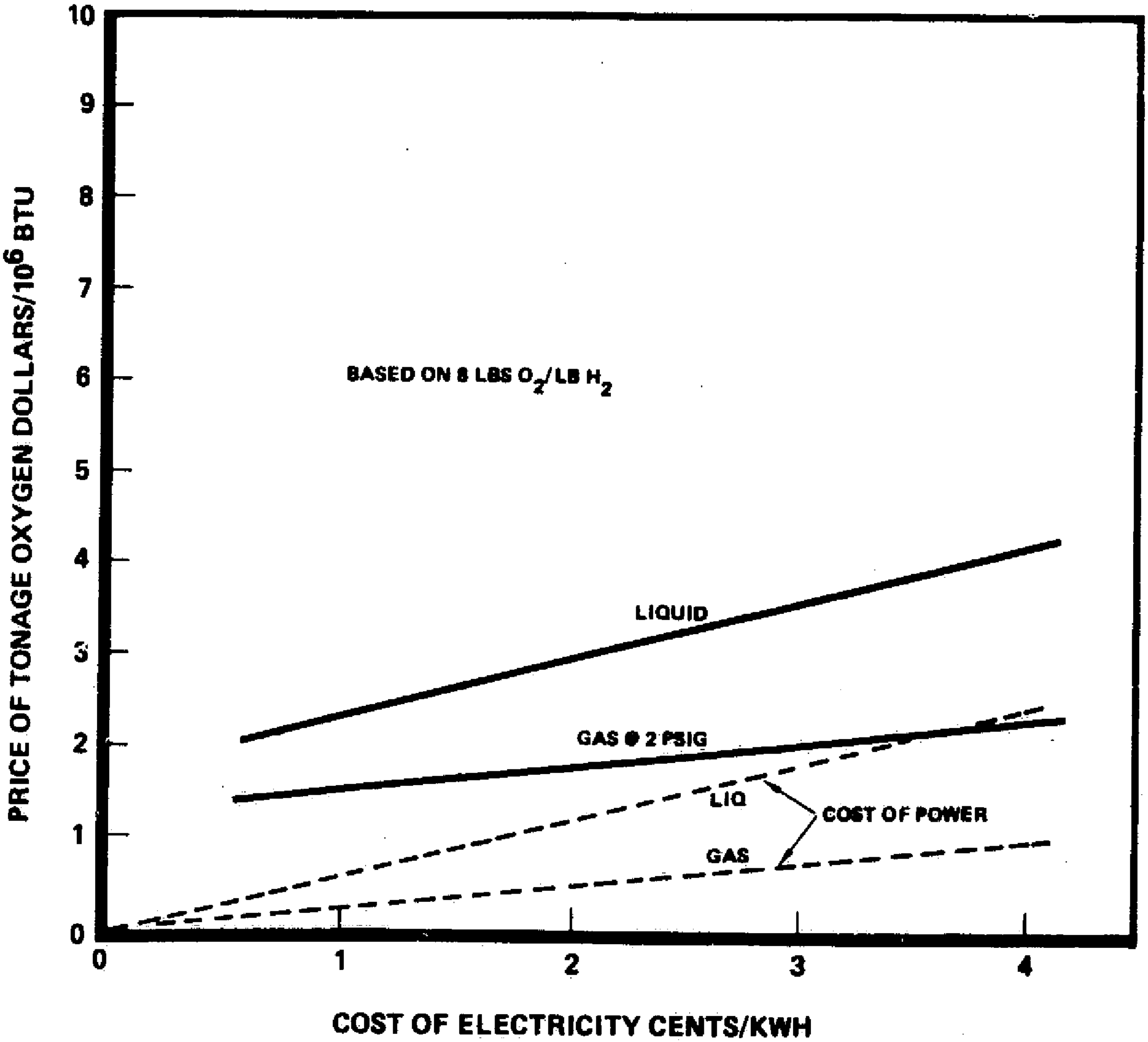


Figure 3. Oxygen by Air Separation

Figure 3 presents some estimates of the price of tonnage O_2 delivered to a central station in either liquid or gaseous form. As can be seen, the cost of such O_2 is expected to vary with the cost of electricity, as the O_2 would be produced by separation from air through processes involving refrigeration and fractional distillation. The costs are presented in terms of O_2 cost/million Btu of H_2, assuming that O_2 is supplied at the stoichiometric ratio, 3.6 kg (8 pounds) of O_2/lb of H_2. This means that the costs presented are the costs of 59.4 kg (131 pounds) of O_2.

The results of the parametric analysis of H_2 and O_2 costs are summarized in Table 2. Many of the economic comparisons made throughout Task 2 were based on the costs listed under the "MID" column.

The ground rules utilized in conducting the economic studies are summarized in Table 3. Fundamental to the ground rules was the specification that the H_2 and O_2 would be delivered "over the fence" as gases at the ambient temperatures and pressure required for utilization in the H_2 PCS. This assumption very much simplifies the economic studies as all questions of storage and compression are omitted.

Estimates of capital equipment costs were based on the 1977 cost of equivalent equipment. Costs of H_2 and O_2 combustor equipment were based on Rocketdyne experience with the fabrication of H_2/O_2 rocket engine and combustor equipment. Costs of turbine generating equipment, electric equipment, valves, piping, etc., were estimated on the basis of current costs for equivalent equipment. It must be recognized that the costs of the more advanced high-pressure/high-temperature equipment have less foundation than the costs of currently produced equipment.

Interest and escalation expense was lumped together as $(1 + \text{interest})^{\text{buildtime}}$ x capital cost. This represents a simplification of interest and escalation expense accounting that is believed to be sufficiently accurate for the end results to be in the right proportion. Interest and escalation expenses are incurred during the period between the time that a commitment is made to construct the station and the time that plant operation begins. During this interval, no power is produced from the plant. At any given moment, that equipment which has been purchased and paid for is assumed to be accumulating interest expenses, while that equipment which has not yet been paid for is being escalated in price due to general inflation. The total of the interest and escalation expense must be added to the actual purchased price of equipment and labor in determining the accumulated capital cost of the installation when power production is initiated.

The fixed charges were assumed to amount to 18% of installed cost each year. Fixed charges are independent of the amount of electricity produced in any year. The 18% value is used frequently in making evaluations of alternative power generating systems, and is reasonably consistent with the assumption of a 30-year useful life of the installation and investor-owned utility economics.

TABLE 2. PARAMETRIC HYDROGEN AND OXYGEN COSTS, $\$/10^6$ BTU

HYDROGEN FROM COAL

	Low	Mid	High
H_2 And Air	3.00	3.75	5.00
H_2 and GO_2	4.50	5.65	7.20
H_2 and LO_2	5.30	7.00	9.10
Coal Cost $\$/10^6$ Btu	0.50	1.00	1.50
Power Cost Mils/kWh	15	25	40

HYDROGEN BY ELECTROLYSIS

	Low	Mid	High
H_2 and Air	5.50	11.00	17.00
H_2 and GO_2	5.50	11.00	17.00
H_2 and LO_2	6.30	12.35	18.90
Power Cost Mils/kWh	10	25	40

TABLE 3. COE GROUND RULES

- H_2 and O_2 Delivered "Over the Fence" as Gases at Required Pressure
- 1977 Capital Equipment Prices
- Interest and Escalation Expense at $(1 + \text{interest})^{\text{buildtime}}$ x Capital Cost
- Fixed Charges = 18% of Installed Cost/Year
- Maintenance Estimate Based on Equipment Type
- Operating Labor at $25/hr
- Development Costs not Included

Both the maintenance and operating costs were estimated individually on the basis of the kind of equipment installed and the duty cycle being examined. As seen in Table 3, the cost of operating labor was assumed to be \$25/hr, which includes all overhead. Thus, the cost of operation was assumed to be the total operating labor hours (including supervision) multiplied by \$25.

In each instance, the technology was assumed to be sufficiently developed so that the system was ready for commercial operation; no development costs other than those normally encountered in commercial powerplant operation were included in the economic balance.

Much of the economic criteria were developed in coordination with the R.M. Parsons Co., which has considerable experience in this area.

The third task of this study was organized to concentrate on the conceptual design and analysis of a plant incorporating the SSG H_2 PCS concept. The objectives were to provide a detailed description of the overall system, including the necessary flow charts, schematics, and heat and mass balances, so that the required operating conditions and performance of the supplementary equipment could be defined. Based on this description, a specification for the major PCS components, i.e., the combustors and their related controls were to be defined in sufficient detail to permit an estimate of cost of these components. As part of these specifications it was planned to provide preliminary design-type drawings of the components and the system, again with the objective of providing sufficient detail to permit an estimate of the cost of the components and the nature of the R&D program required for the development.

The environmental impact of the supplementary steam cycle was to be assessed in sufficient detail to permit an assessment of whether significant environmental problems would need to be resolved in the development of the PCS.

Estimates were to be prepared of the capital costs, including interest and escalation charges during the construction period, as well as operating and maintenance costs of the PCS system. These cost estimates and the technical analyses outlined above then were to be utilized in preparing an analysis of the technical advantages and economic benefits of the integrated supplementary steam PCS/fossil fuel-fired integrated installation. Additionally, the technical and/or environmental and economic problems brought out by the economic and technical analyses were to be identified so that a plan for specific R&D efforts, a schedule of time, and an estimate of the cost required could be prepared on the effort required to develop the integrated H_2 PCS/fossil-fueled concept to the demonstration stage.

MARKET POTENTIAL

TOTAL ELECTRIC CENERATING CAPACITY FORECAST

To determine the market potential for the various candidate PCS's, it was necessary first to project the aggregate growth in electric generating capacity over an extended period of time, then to estimate the probable mix of electric generating technologies during this period. The available market for each of the candidate PCS's, expressed in megawatts of capacity, then could be determined by inspection of curves illustrating this growth in capacity and mix. For one of these candidate systems, i.e., boiler replacmenet, retirement rather than growth in capacity determined the market potential.

Projection of the total installed electric generating capacity to the year 2000 is shown in Fig. 4 . This projection is based on data from the Federal Power Commission (FPC), Edison Electric Institute (EEI), Electric Power Research Institute (EPRI), Energy Research and Development Administration (ERDA), and the publication, Electrical World.

The ERDA high- and low-growth cases bracket most of the projections developed by other sources. The high-growth case assumes intensive electrification, while the low-growth case assumes signifciant conservation efforts and no increased degree of electrification. There is fairly good corroboration of projected growth data from the other sources to the mid-1990's, and this is shown as the solid dark line in Fig. 4 . The Rocketdyne extrapolation from the year 1995 to 2000 assumes growth consistent with that established earlier in that decade, and yields a total United States installed electric generating capacity of 1,620,000 MW by the end of the twentieth century.

The projected mix of electric generating technologies at 5-year anchor points from 1975 to 2000 is shown in Table 4 . Data from three primary sources, EEI, FPC, and Electrical World was used in construction of the matrix, and Rocketdyne extrapolations were used to complete the matrix for the distant years. Applying this mix to the total installed electric generating capacity curve of Fig. 4 yielded the capacity growth curves for generic technologies shown in Fig. 5 .

Although fossil steam electric generating capacity, as a percentage of total capacity, continues to decline through the period to the year 2000, it still represents more than one-half of the total U.S. capacity in the year 2000, or 912,000 MW. Nuclear steam electric capacity, which has healthy growth in the period, primarily at the expense of fossil steam electric capacity, is projected to rise to about 450,000 MW by the year 2000. This is a little less than one-half that of fossil steam in that year.

The other technologies, conventional/pumped hydroelectric and combustion turbines/internal combustion (including combined cycle), show shallow declines in growth to the year 2000, with each expected to represent 130,000 MW of installed capacity in that year.

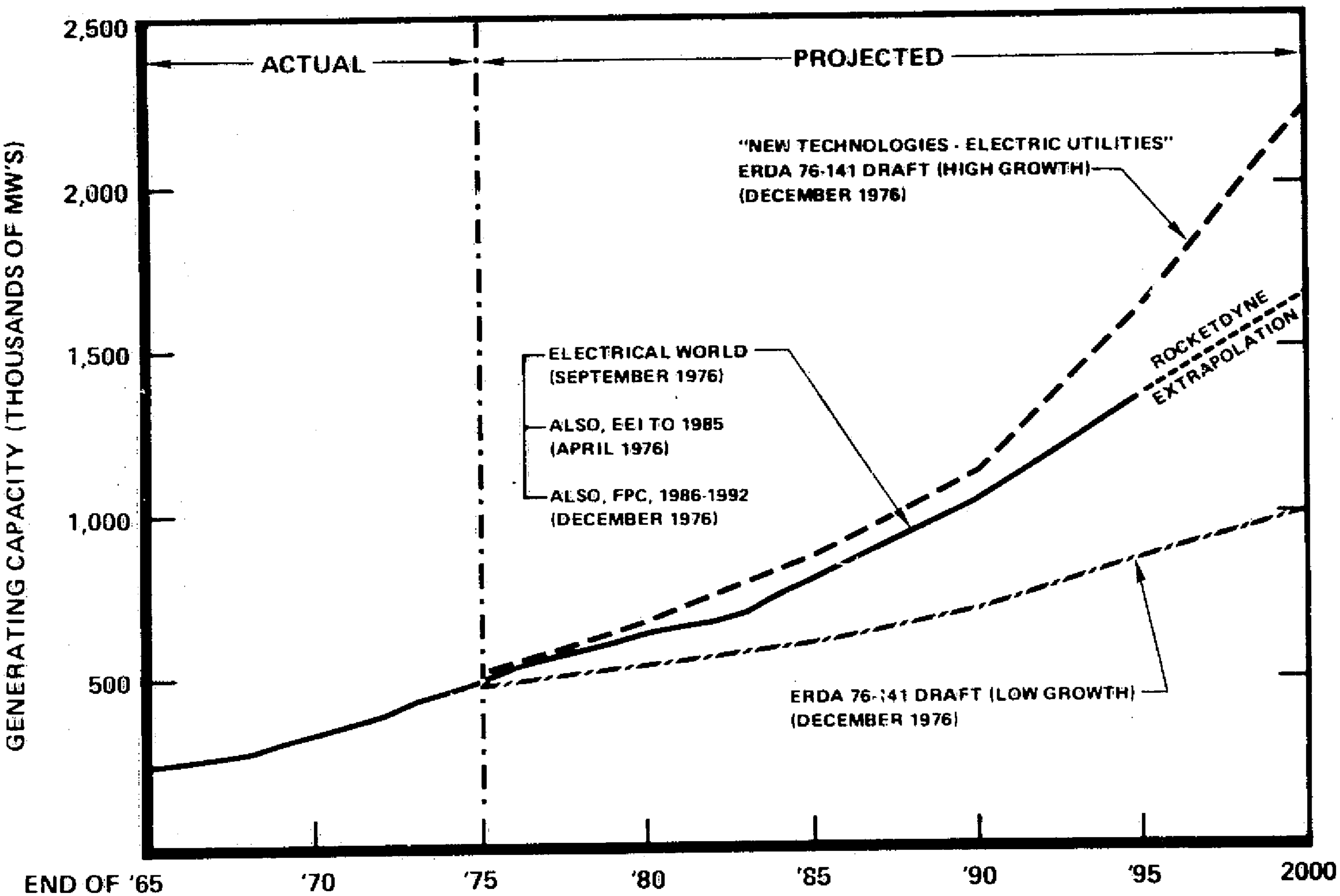


Figure 4. Total Installed Electric Generating Capacity

TABLE 4. PROJECTED MIX* OF NET ELECTRIC GENERATING CAPACITY

	1975**	1980	1985	1990	1995	2000
Fossil Steam	68.7	65.7	60.9	59.5	57.5	56.3
Nuclear Steam	7.2	12.3	19.7	23.1	26.0	27.7
Conventional and Pumped Hydro	13.9	12.5	10.4	9.0	8.5	8.0
Combustion Turbines and Internal Combustion (including combined cycle)	10.2	9.5	9.0	8.4	8.0	8.0
Total	100.0	100.0	100.0	100.0	100.0	100.0

*Mix shown in percentages
**Actuals (approximate)

Sources: EEI, "59th Electric Power Survey" (April 1976)
"Electrical World," 27th Annual Electrical Industry Forecast, (15 September 1976)
FPC News Release No. 22763, "Electric Utility Expansion Plans, 1986-1995" (8 December 1976)

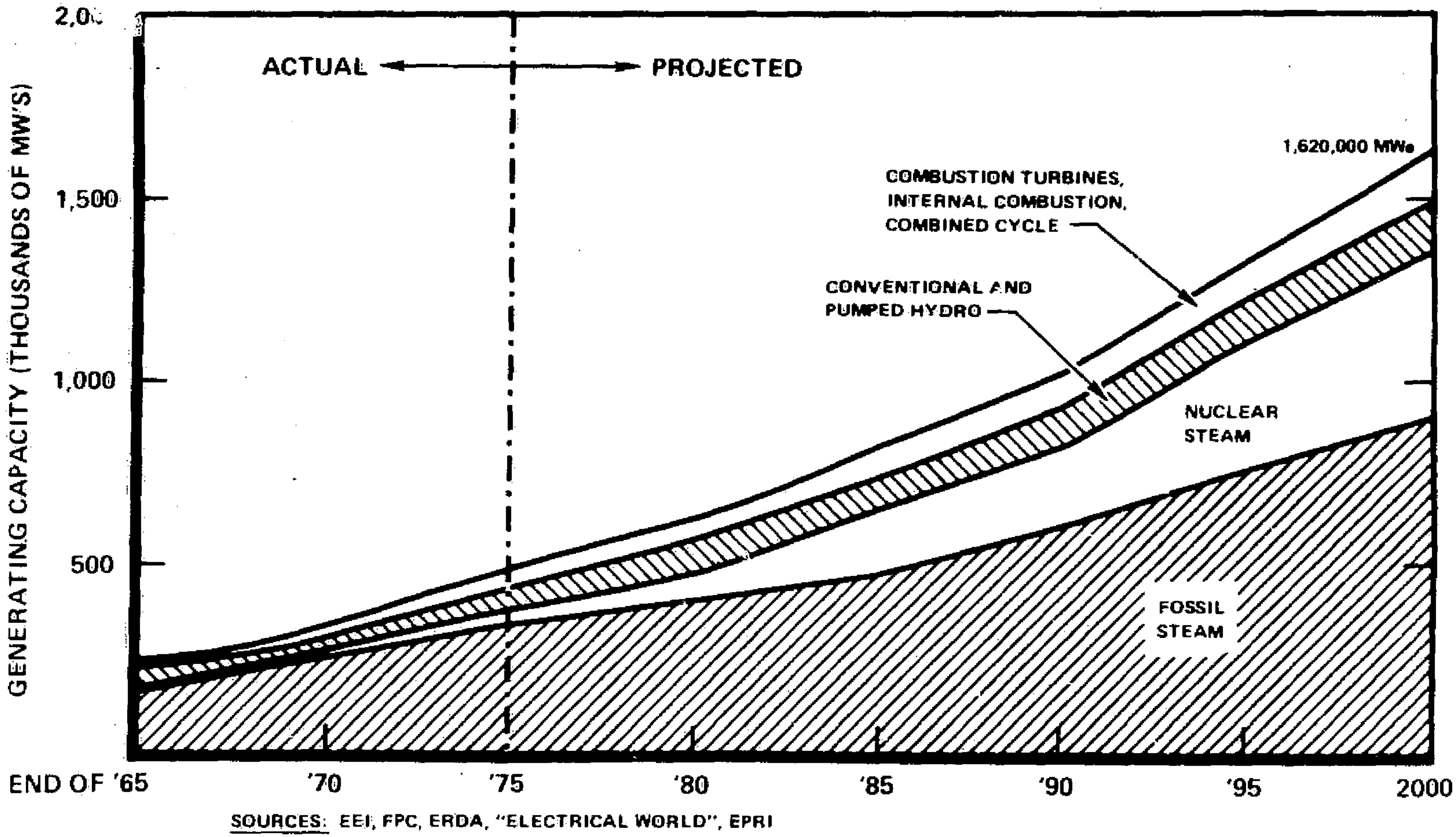


Figure 5. Total Electric Generating Capacity Growth

Second- and third-generation technologies, such as solar thermal electric and magnetohydrodynamics, were purposely excluded from this figure because of uncertainty regarding the timing of their commercial introduction. In any event, the contribution to total electric generating capacity would be a fraction of a percent in the early 1990's, rising to a probable maximum of 5% of total installed capacity by the year 2000, and would displace some of the capacity shown for the generic technologies in Fig. 5. In other words, the projected total capacity would be the same 1,620,000 MWe, but the mix in the out years would be altered slightly.

Table 5 summarizes some of the key growth rates associated with this total electric generating capacity forecast.

TABLE 5. AVERAGE ANNUAL COMPOUND GROWTH RATES

(1975-2000)

Real GNP	⟶	3.5%
Population	⟶	0.8%
Total Installed Electric Generating Capacity	⟶	4.7
Nuclear Steam Electric Generating Capacity	⟶	10.5%
Fossil Steam Electric Generating Capacity	⟶	3.8%

NUCLEAR STEAM ELECTRIC GENERATING CAPACITY FORECAST

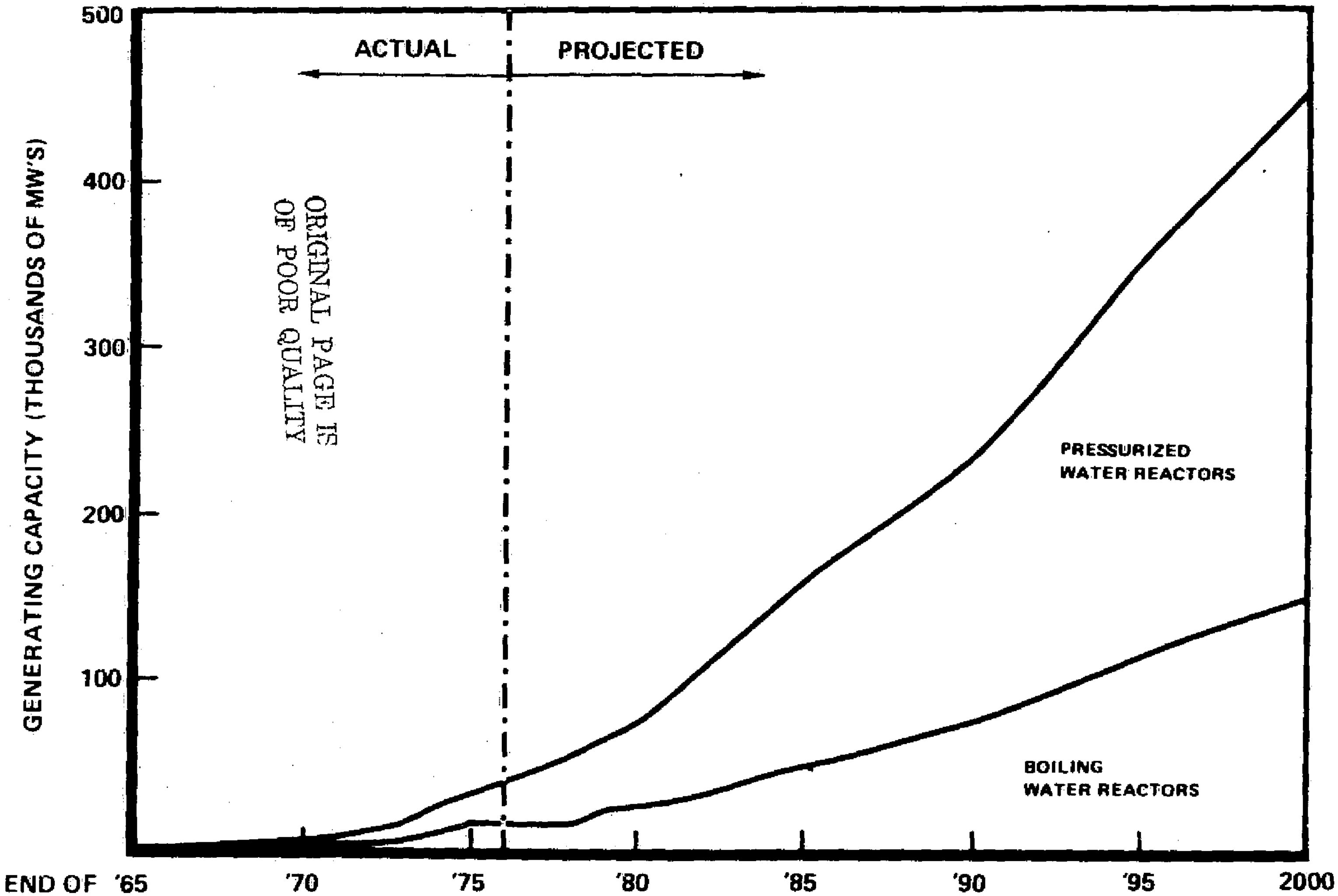
A tabulation of nuclear plants and capacity by year and reactor type (pressurized water or boiling water), for all plants either operational, under construction, or planned, is shown in Table 6. Electrical World's 1977 Nuclear Plant Survey (15 January 1977) was particularly helpful in the preparation of this table. The planned capacity additions beyond 1985 diminish rapidly because of the utilities' uncertainty regarding national energy policy, and because it is not necessary now to make a commitment to build a plant 10 to 20 years into the future.

Figure 6 shows the projected growth in nuclear steam electric generating capacity to the year 2000. This was constructed using the information from Tables 4 and 6, and Fig. 5. The historical 2:1 ratio of pressurized water reactor (PWR) capacity to boiling water reactor (BWR) was used in this projection.

TABLE 6. NUCLEAR PLANT INSTALLATIONS

Year	PWR		BWR	
	No. of Plants	Capacity, MW	No. of Plants	Capacity, MW
1960	-	-	1	200
1961	1	175	-	-
1962	-	-	-	-
1963	-	-	-	-
1964	-	-	-	-
1965	-	-	2	137
1966	-	-	-	-
1967	-	-	-	-
1968	2	1,025	-	-
1969	-	-	3	1310
1970	2	987	2	1499
1971	1	700	2	1354
1972	4	2,854	4	2819
1973	7	5,311	-	-
1974	8	5,541	4	3973
1975	5	4,776	5	4031
1976	4	3,717	-	-
1977	7	6,299	2	1886
1978	5	5,118	1	1078
1979	6	6,349	4	3493
1980	6	7,090	3	3246
1981	11	11,862	4	4322
1982	12	12,849	3	2795
1983	10	11,465	7	8097
1984	14	16,044	5	5699
1985	9	10,331	5	5776
1986	10	10,832	4	4664
1987	5	5,639	1	1178
1988	3	3,440	1	1150
1989	2	2,068	-	-
1990	2	2,050	-	-
1991	1	918	-	-
1992	1	1,150	-	-

NOTE: Includes those plants which are either operational, under construction, or planned. Excludes one HTGR plant and eight plants with indefinite planning status.



SOURCES: "ELECTRICAL WORLD", "POWER", EEI, FPC, ERDA

Figure 6 . Nuclear Steam Electric Generating Capacity

FOSSIL STEAM ELECTRIC GENERATING CAPACITY RETIREMENTS

Annual and cumulative utility boiler retirements to the year 2005 are projected in Table 7. The best method for calculating these retirements, using the data available to Rocketdyne, was determined through discussion with the FPC's Bureau of Power. Briefly, it consisted of using the known fossil steam electric plant capacity additions through the year 1975, and applying a 30-year boiler life to these figures to yield projected retirements to the year 2005. Inasmuch as the FPC data for total installed capacity, available to Rocketdyne, was reported in 10-year increments prior to 1970, constant annual growth rates between these anchor points were assumed to arrive at the annual figures shown. Actual annual figures were used after 1970. The cumulative fossil steam electric generating capacity retired by the year 2000, using this methodology, is almost 341,000 MW, as shown in Fig. 7 .

MARKET POTENTIAL ASSESSMENT

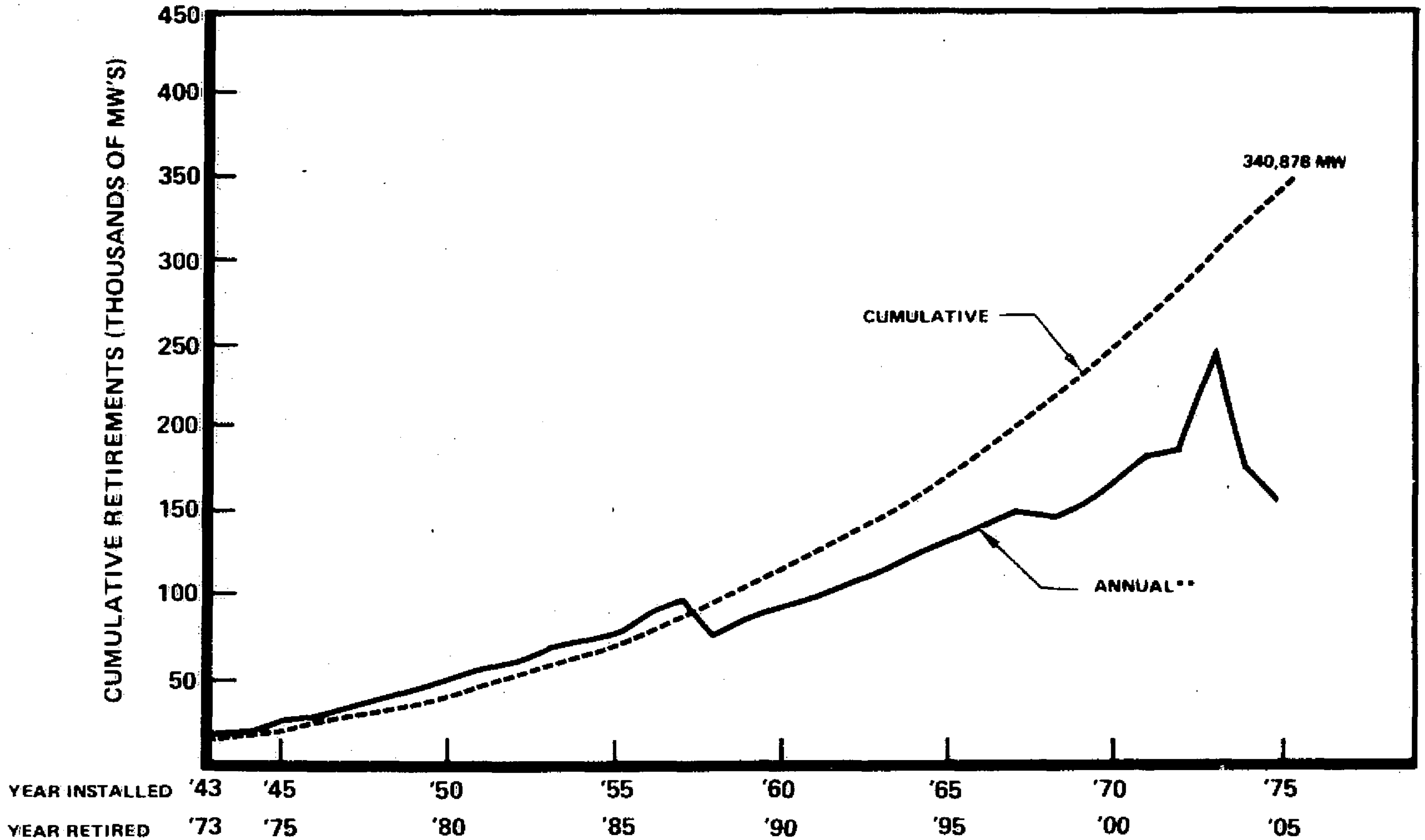
Table 8 summarizes the market assessments completed and illustrates the market potential for H_2 turbine power generation systems. The following conclusions that can be drawn:

1. Market potential, or lack of, is not a deciding factor in evaluating the future mix of H_2 turbine systems.
2. A significant market exists for all the H_2/O_2 combustion cycle system.
3. In fact, the market fraction capable of using the H_2/O_2 combustor actually increases over the time span studied (from 80% in 1980 to 92% after the year 2000).
4. The largest single market exists for steam boiler replacement or supplementary applications.

TABLE 7. FOSSIL STEAM ELECTRIC PLANT RETIREMENTS (ACTUAL AND PROJECTED)

Years	Annual, MW	Cumulative, MW
1973 and Before	-	14,772
1974	1,880	16,652
1975	2,250	18,902
1976	2,694	21,596
1977	3,225	24,821
1978	3,855	28,676
1979	4,269	32,945
1980	4,725	37,670
1981	5,230	42,900
1982	5,790	48,690
1983	6,410	55,100
1984	7,096	62,196
1985	7,855	70,051
1986	8,695	78,746
1987	9,626	88,372
1988	7,731	96,103
1989	8,332	104,435
1990	8,979	113,414
1991	9,677	123,091
1992	10,429	133,520
1993	11,239	144,759
1994	12,112	156,871
1995	13,054	169,925
1996	14,067	183,992
1997	15,161	199,153
1998	14,717	213,870
1999	15,747	229,617
2000	16,849	246,466
2001	18,028	264,494
2002	18,456	282,950
2003	24,308	307,258
2004	17,751	325,009
2005	15,869	340,878

NOTE: Annual figures are calculated on a compounded growth basis using known FPC discrete 10-year net installed capacity anchor points



*BASED ON FEDERAL POWER COMMISSION HISTORICAL DATA, USING AN ASSUMED 30-YEAR BOILER LIFE

**COMPOUNDED TO 1970; ACTUALS THEREAFTER

Figure 7. Projected Utility Boiler Retirements*

TABLE 8. MARKET POTENTIAL SUMMARY

	Years		
	1980	1980 to 2000	After 2000
Total Capacity, MW	550,000	1,000,000	1,600,000
Mix, %	86	90	92
Nuclear Capacity, MW	50,000	130,000	500,000
Predominant Market, MW			
• Retired	50,000	100,000	250,000
• Substitution	310,000	500,000	750,000

TECHNICAL AND ECONOMIC ANALYSIS

This section presents the results of the technical and economic analyses performed on each PCS evaluated. Since the program was structured to delete less promising PCS with each succeding task, the amount of analyses performed reflects this. The Supplementary Steam Generation cycle which was the PCS selected for Task 3 cycle received a complete analyses through plant conceptual design. Plant conceptual design consisted of system analysis, PCS conceptual design, cost estimate and specific evaluations to identify technical and environmental problems plus recommediations, schedule, and cost estimate for an R&D program.

In addition to SSG, the Ericsson cycle and the hydrogen gas turbine cycle received the analyses through Task 2. All of the cycles considered received the analyses of Task 1. As a result of this approach, the BWR, PWR, high-temperature steam turbine cycle and partial condensing Rankine-Brayton cycle received only the least analyses as required by Task 1.

The results of the technical and economic analyses performed for each PCS presented below.

SUPPLEMENTARY STEAM GENERATION

System Description

The hydrogen power conversion system to be evaluated in Task 3 has been described as supplementary steam generation (SSG). The concept provides for the corporation of peaking power by providing additional steam during peaking periods that is over and above the capacity of the fossil-fired boiler installed in the station. One can visualize several scenarios for this condition. There now exist conditions in which steam boilers cannot be operated at full-rated steam output conditions because of environmental restrictions, or because it has been necessary to switch from oil or gas to coal with a resulting reduction in output capacity. The remainder of the station, however, including the turbine generator, transformer, feedwater heating system, etc., are capable of operating at fully rated or even overrated conditions. In this circumstance, one may visualize that the addition of facilities to generate steam by the direct combustion of H_2 or O_2 may be inexpensive enough (in terms of capital costs) so that the use of such facilities during periods of peak load may be justified economically.

In addition to retrofitting existing stations whose boilers are steaming at reduced capacity, it is possible to visualize such supplementary steam generation in new stations. In this concept, additional steam turbine, generator and transformer capacity are provided during construction of a new facility at minimal incremental cost and utilized in conjunction with direct combustion of steam via H_2 and O_2 combustion to furnish economically viable peaking power. In such an installation, for instance, the costs of the coal- and ash-handling equipment, boiler, fuel preparation equipment, flue gas cleanup equipment, etc., would all be based on the base load capacity and only the incremental costs of the H_2 and O_2 combustion system, control system, and turbine/generator/transformer system would be charged to the peaking generating capacity.

To conduct a realistic study of the utilization of H_2 and O_2 for such supplementary steam generation, it was decided to study the integration of the H_2/O_2 steam-generating equipment into a typical coal-fired steam station. The station conditions selected were 12.41×10^6 N/m^2/811 K (1800 psi/1000 F) primary steam conditions and 811 K (1000 F) reheated steam temperature condition. The unit was assumed to have a rated capacity of 160 MWe, but to be limited to 140 MWe by its coal-fired boiler capacity. Thus, 20 MWe of peaking power are to be provied by the generation and superheating of steam with the $H2/O2$ propellant combination.

Having defined a typical system for analysis, the next step was to define the operational and legal requirements that would govern the design of the H_2 PCS and its integration into the larger system. A number of target criteria to be accommodated were identified as discussed below.

1. The nature of the duty cycle, i.e., peaking power supply, implies that both physical arrangement and the control in sequencing of the system should permit the rapid pickup or dropping of load by the H_2/O_2 system. The wasting of H_2/O_2 and/or water must be minimized. It is expected that in typical peaking service, the equipment may well be started 5 to 10 times per week, operating for 2 to 6 hours at a time. The needs of peaking service are best serviced by a capability for rapid acceptance and shedding of load, on the order of 10 minutes between zero- and full-power output from the H_2 PCS.

2. Integration of the H_2 PCS with the remainder of the steam station must be accomplished with a minimum of disruption to the conventional fossil fuel station design and operation. Such minimization is necessary so that the incremental cost of adding the $H2$ PCS is minimized, making it competitive for the peaking application. There are implications here relative to the integration of the H_2 PCS into the station steam piping and the location of the PCS relative to other equipment for both automation and ease of operation. Further implications exist with respect to control and operation of the H_2/O_2 PCS to minimize the quantity of noncondensible gases routed to the steam condenser, where their presence influences turbine exhaust vacuum, cycle efficiency efforts of pump out and, possibly, plant safety.

3. The safety aspects of the addition of the H_2/O_2 PCS to the conventional steam station are another area influencing the choices for design of such equipment. Both station operating personnel and the general public must be protected against undue hazard. Design criteria relative to plant safety are well defined for conventional fossil fuel steam stations in a series of codes such as the Boiler and Pressure Vessel Safety Code and the Power Piping Code, which are written and kept up to date by committees under the sponsorship of the American Society of Mechanical Engineers. Such codes have the force of law

in many localities on the basis of state and local ordinances. Additionally, safety codes and recommendations are available on combustion equipment, electrical equipment, and the like. Additionally, the OSHA provides safety regulations relative to provisions for safety of plant operating personnel. These codes and regulations furnish substantial guidance on design requirements but do not, at present cover the specific condition of the generation of additional steam by the combustion of H_2 and O_2 inside a pressure vessel and a full operating steam pressure and temperature.

4. Finally, there are series of safety regulations with respect to the effluents of the plant, i.e., its environmental impact. While these are well defined with respect to sulfer oxides, nitrogen oxides, particulates, carbon monoxide, etc., there are no present regulations governing the emission of H_2 or O_2 or water vapor.

An analysis of the operating and design requirements, which are broadly defined above, leads to a recognition that there are a number of potential problems in the design of the supplementary steam generating system that need satisfactory resolution to provide an acceptable and economically competitive system. These are discussed below.

The requirement for rapid acceptance and rejection of load by the H_2 PCS steam supply system has significant effect on the design. The startup and shutdown time of steam generating systems is typically affected by the thickness of the pressure parts and the generally accepted objective that fatigue cracking of the pressure parts be avoided by preventing the thermal stresses at any point in the system from exceeding the yield point of the material. Adherence to these standards frequently requires that the startup time of a 12.41×10^6 N/m^2/811 K/811K (1800 psi/1000 F/1000 F) temperature steam powerplant takes many hours, compared with the 10-minute startup time targeted for this peaking installation. The implication for the supplementary steam system is that it should be at operating temperature at all times and capable of being started in such a way that the temperature shocks involved are not sufficient to cause yielding in any of the affected pressure points.

The H_2, O_2, H_2O, and steam flow-rates for the primary high-pressure steam and the lower pressure reheated steam necessary to increase the station output from 140 to 160 MWe are defined in Table 9. The primary steam flow augmentation is derived from H_2/O_2 combustion, which produces a very high temperature pure steam, plus the addition of "tempering" feedwater flow which, by evaporation, brings the temperature of the H_2/O_2-produced steam down to 811 K (1000 F) to match the temperature of the steam produced by the fossil fuel boiler. Therefore, the steam leaving the high-pressure turbine, which is routed to the reheater in the fossil fuel boiler quantity will correspond to the 160 MWe operating level. Thus, it is not necessary to increase the flow of reheat steam but, since the boiler is operating at a condition where the heat added to the reheater corresponds to the 140 MWe, and not to the 160 MWe, level, it is expected that external heat addition to the reheat steam will be necessary to provide for 811 K (1000 F) temperature of reheat steam to the turbine. In this concept, the heat addition to the reheat steam is provided by the combustion of a small quantity of H2 and

TABLE 9. COMBUSTOR FLOWRATES

Primary Combustor	
Hydrogen Flow	0.653 lb/sec
Oxygen Flow	5.22 lb/sec
Water Flow	31.9 lb/sec
Steam	37.8 lb/sec = 136,000 lb/hr
Reheater	
Hydrogen Flow	0.142 lb/sec
Oxygen Flow	1.137 lb/sec
Steam	1.279 lb/sec = 4604 lb/hr

O_2 in the steam line connecting the exit of the high-pressure turbine with the inlet of the fossil fuel-fired boiler reheater. The flow proportions throughout the steam generating system, when operating under this concept, are slightly deviant from the conditions that would have existed had the 160 MWe been generated entirely by steam from the fossil fuel boiler, but these deviations are not significant to operation and can be accommodated, if necessary, by operating with the high-pressure steam flow very slightly below rated and the reheat steam flow very slightly above rated.

The equipment arrangement planned to provide the increase in steam flow and the increment in reheat enthalpy is shown schematically in Fig. 8. Most of the system shown would already exist in a typical plant. Only the combustors and their related equipment are additional to this concept. The storage system and compressors are not a part of this study, but are shown as a typical case. Design features of interest as shown by this arrangement are as follows.

Provisions for rapid acceptance and rejection of load by the H_2/O_2 primary steam generator are provided. The H_2/O_2 combustor is positioned between two spray-type desuperheaters. These desuperheaters are present state-of-the-art equipment available commercially from several firms. They are constructed so that the pressure parts of the steam piping are never subjected to excessive thermal gradients. The water supply may be turned on or off at any time, as required to modulate steam temperature, and no thermal shocks are caused (other than those caused by the change in steam temperature itself.) Provisions are made in the desuperheaters for thermal sleeves around the water injection piping and for a thin, internal, thermal sleeve in the steam piping to protect the piping itself from impingement by water droplets. The H_2/O_2 combustor is also designed so that its pressure parts operate at all times at the temperature of the flowing steam. Thermal sleeves are provided on the GH_2/GO_2 inlets to minimize thermal shock where the lines pass through the wall of the pipe. A combustion can is provided that is similar in some respects to the gas turbine can. It mixes the hot products of the H_2/O_2 combustion with the flowing steam to maintain the temperature of the can components within acceptable limits.

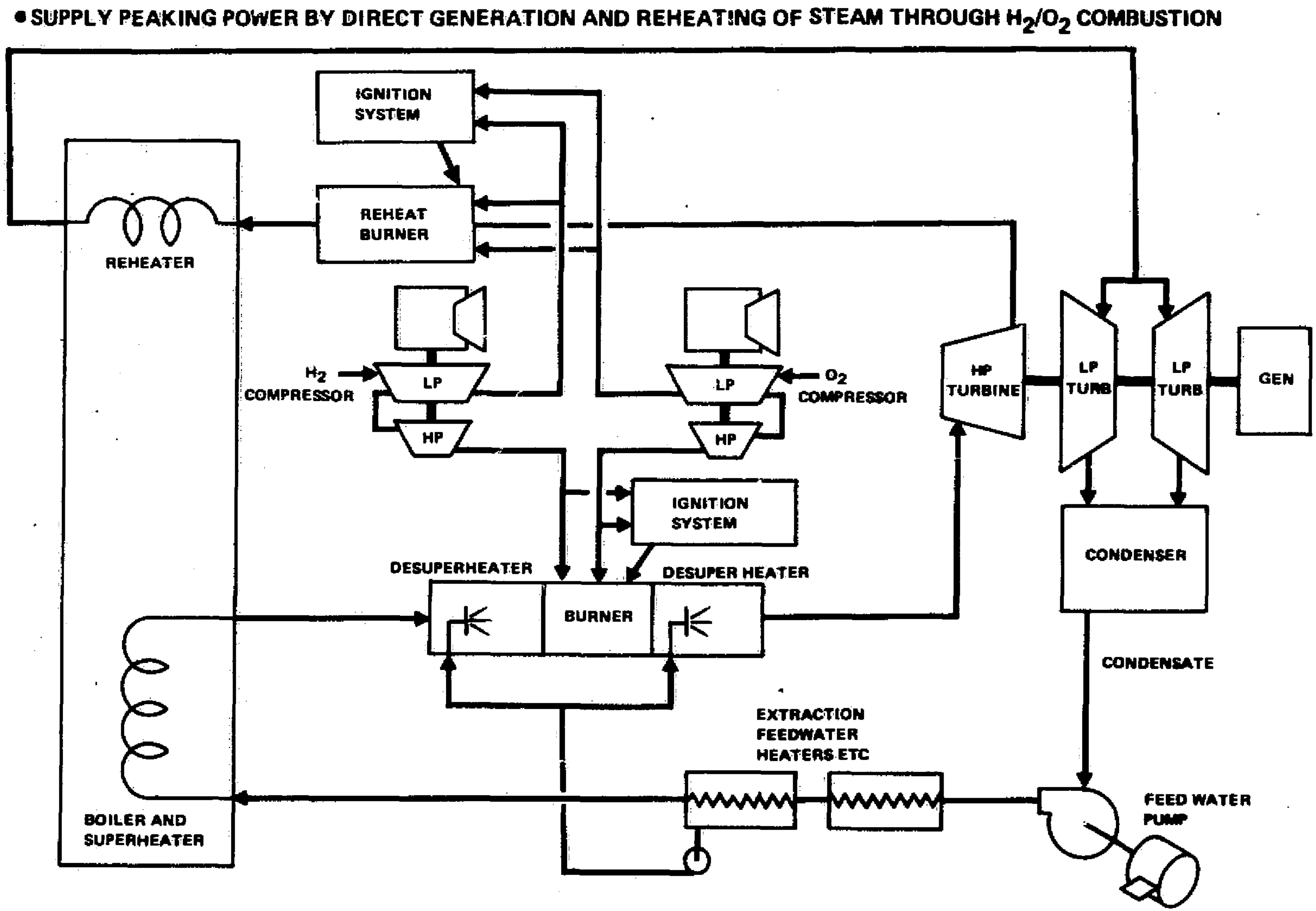


Figure 8. Supplementary Steam Generation Concept

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The can components, themselves, are not steam piping pressure parts. The operational concept for this equipment is that the steam piping and internals will be operating at 811 K (1000 F) prior to lightoff of the H_2/O_2 combustor. Lightoff will take place at roughly 25% of rated H_2 and O_2 flow and, under these conditions, the decrease in steam temperature at the upstream desuperheater will be less than 283 K (50 F), the rise in steam temperature at the burner will be less than 311 K (100 F), and the decrease in steam temperature at the downstream desuperheater will be less than 283 K (50 F). The result will be the commissioning of the H_2 combustor in a step increment to 25% of its rated output without producing unacceptable thermal gradients in the pressure parts of the piping or turbine. Further increases in steam output, from this point can be accomplished by simultaneously increasing H_2, O_2, and cooling water flow, thus maintaining low thermal gradients throughout.

The H_2/O_2 steam reheater does not require the addition of tempering water. As shown in the schematic (Fig. 8) it merely adds some sensible heat and a very small increment in weight flow to the steam in the line between the high-pressure turbine and the boiler reheater. The reheat burner would be ignited a short time after satisfactory operation of the primary burner had been attained on the order of several minutes. It, too, would begin firing at roughly 1/4 of rated capacity. The step increment in steam flow occasioned by lightoff of the primary hydrogen combustor and desuperheaters can be accommodated easily by a reduction of steam flow from the fossil-fuel-fired boiler, maintaining constant steam flow to the steam turbine until it is ready to accept the greater electrical load. Once ignited, it is expected that the heat input by the primary and reheater H_2 PCS systems can be varied at will over the range from 1/4 rated to full rated input, with the complete load change a matter of only a few minutes, certainly fast enough to accommodate the usual 1% per minute load change customary for the turbine.

In a general sense, the H_2 PCS concept defined schematically in Fig. 8 appears capable of meeting the operational requirements outlined above. There are however several areas that need detailed examination and/or development to provide satisfactory operation. Any solids present in the desuperheating water supplied to the primary steam line will be carried through the piping and into the steam turbine, where they may create a problem with deposits and/or erosion. The satisfactory answer to this problem appears to be to follow practices frequently used in existing stations that employ spray-type desuperheaters, i.e., either deionized water or the very pure condensate available from the high-pressure heaters. Both of these expedients provide satisfactorily pure water.

Reliable and fast response controls are required for several reasons. The steam flow must be proportioned to the load, the H_2 and O_2 flows must be proportioned to minimize the presence of noncondensibles in the condenser, and control of the ignition system must provide reliable ignition and positive protection against flameout. While there are many concepts for controlling the combined fossil-fuel-fired boiler and H_2 PCS steam flow to respond to general load demands, perhaps one of the simplest and most reliable would be to place the H_2 PCS load under manual control, permitting the existing fossil-fuel-fired boiler control system to provide constant steam pressure and respond to variations in turbine demand. With such a system, the H_2 feed would be under essentially manual control, and the O_2 feed would be proportioned to the H_2 feed to maintain a proper stoichiometric ratio. A trim on the proportion of O_2 to H_2 might be provided by a steam-sampling system that would respond to indications of H_2 and/or O_2 in the steam system. A steam temperature measurement downstream of the H_2 burner and desuperheater system would control the quantity of tempering water added in the upstream and downstream desuperheaters. The steam temperature upstream of the H_2 combustion system would be regulated by the boiler control system in the normal manner.

The control of ignition and flame safety appears to be one of the more critical and challenging technical problem areas. The present concept envisions an augmented spark igniter (ASI) which is basically a very small combustor that is ignited by an electrical spark and which operates at a mixture ratio that is both low enough so that uncooled metal walls can be used to contain the burning gases, but high enough that a mixture of H_2 and O_2 exposed to its flame will be ignited. Such an ASI could be mounted on the exterior of the combustion system where its ignition and stable operation could be sensed by thermocouples and/or radiation-type ignition detectors, and its hot gases conveyed to the interior of the combustor where ignition of main propellants would occur. Additional safeguards to detect ignition and verify the continued combustion of the main propellant also appears appropriate, and this equipment might take the form of thermocouples and/or radiation-type flame-detection equipment. Additionally, an overall system to verify the presence of the flame may be constructed with a sensing and computing system that integrates the overall effect of upstream and downstream steam temperatures, H_2 flowrate, the superheating water flowrate, and steam flow. A significant unbalance in enthalpy relationships among these quantities is cause for suspicion that the flame conditions are not under proper control.

The minimization of noncondensible gases in the steam produced by direct combustion of H_2 and O_2 is important because these gases will wind up in the extraction heater and the condenser, from which they will have to be removed to maintain correct pressure balances and heat transfer effects. The quantity of noncondensible gases produced via deviation of the mixture ratio from stoichiometry and/or the presence of contaminant gases in either the H_2 or O_2 are presented in Table 10. The presence of noncondensible gases affects the overall system performance because they must be pumped from the condenser against the differential pressure between the condenser and the outside atmosphere. Additionally, some of these gases such as O_2 and CO_2 may result in corrosion effects in the turbine and/or condenser, and the presence of H_2 may present a safety hazard.

Analysis indicates that the stoichiometric ratio of O_2 to H_2 can probably be maintained via flow proportioning (trimmed from hydrogen concentration) to a deviation of approximately 1/2 of 1% from exact stoichiometry. Present thinking is that it will be most satisfactory to provide a slight excess of H_2 in the combustion products to avoid the corrosion possibilities existing with free oxygen in the wet portions of the turbine and in the condenser. There will be no explosion hazard present so long as the hydrogen is mixed with the steam and, even in the condenser, the operating pressures will be so low that an explosion hazard with infiltrating air mixing with the H_2 will not exist.

TABLE 10. $^{\Delta}$ NONCONDENSIBLE GASEOUS STEAM FROM H_2/O_2 COMBUSTION

	Stoichiometric	/2% Excess O_2	1% Excess O_2	1/2% Excess H_2	1% Excess H_2
Electrolytic H_2 Electrolytic O_2	0	0.05	0.11	0.11	0.22
Koppers/Totzek H_2* O_2 From Air Liquefaction**	1.5	1.6	1.6	1.6	1.7
U-Gas H_2* O_2 From Air Liquefaction**	1.2	1.3	1.3	1.3	1.4

$^{\Delta}$Standard ft^3 of noncondensible gas/pound of steam; typical 25 MW steam condenser is provided with a 2-stage ejector for 4×10^{-5} scfm of dry air/pound of steam.

*Typical composition of coal-divided hydrogen

	%BY VOLUME	
	K/TOTZEK	U-GAS
CO	0.1	0.1
H_2	93.1	94.3
CH_4	5.5	4-8
N_2 and AR	1.3	0.8

**Typical composition of commercial oxygen produced by air liquefaction:
99.5% O_2
0.5% N_2 and argon

Once the noncondensible gases have been compressed from condenser operating pressure to atmospheric pressure, and the exhauster steam condenser, combustible mixtures of H_2 and air may exist and these will be vented under conditions guarding against accidental ignition.

The final item of some concern relative to the safety of the plant is the possibility that control system malfunctions would result in the addition of so much water to the primary steam desuperheaters that droplets of water would reach the turbine, and cause damage. However, the H_2 PCS system differs from the conditions existing in conventional plants, in which desuperheaters are installed for superheat temperature control, in that those temperature control desuperheaters are usually located between a primary and secondary section of the superheater. Analysis indicates that protection against this eventuality can be provided with a "knock-out drum" incorporated in the steam line between the final desuperheater and the turbine. This drum would effectively remove any slugs of water or moisture that might possibly be traveling with the steam.

The R. M. Parsons Co., one of the project team members on this contract, is familiar with the engineering of coal-fuel steam stations and furnished equipment arrangement drawing layouts. A conceptual layout of the location of the primary steam combustors and desuperheaters and the secondary steam combustors is indicated in Fig. 9 and 10. The primary steam generator and desuperheaters are located in a protected but somewhat remote area of the station in the main steam line between the boiler and the high-pressure turbine. The combustor for reheat supplementation is located at the same elevation in the cold reheat line between the high-pressure turbine and the boiler reheater. The location provided is accessible for maintenance in all weather and permits minimum disruption to the usual routing of the high-pressure and reheat steam lines. Operation of the combustors and desuperheaters will be entirely by remote control from the control room. Provision for safety sheilding of the combustors may be made if a more thorough analysis indicates this to be necessary.

Technical Analysis

There have been an increasing number of fossil-fuel-fired steam power plants that have to operate at reduced capacities for the necessary reasons of switching to poorer grade fuels and pollution abatement. It has been suggested that H_2/O_2 direct combustion steam generation might be able to provide an economical approach to supplement the existing boilers with the additional steam generating capabilities to run the turbines at full loads, especially during the peaking periods. Such H_2/O_2 direct steam generators not only would require low initial capital investment but also would enable quick startup and shutdowns, which are necessary in applications as peakers or spinning-reserve units.

To assess the effects of the use of an H_2/O_2 supplemental steam generator on the overall plant performance, detailed cycle analyses are conducted as a typical mid-range fossil-fuel steam power plant of 160 MW with throttle steam condition of 12.41×10^6 N/m^2 and 811 K (1800 psig and 1000 F) and reheat temperature of 1000 F. The steam turbines are tandem-compounded 3600-rpm, 2-inch Hg

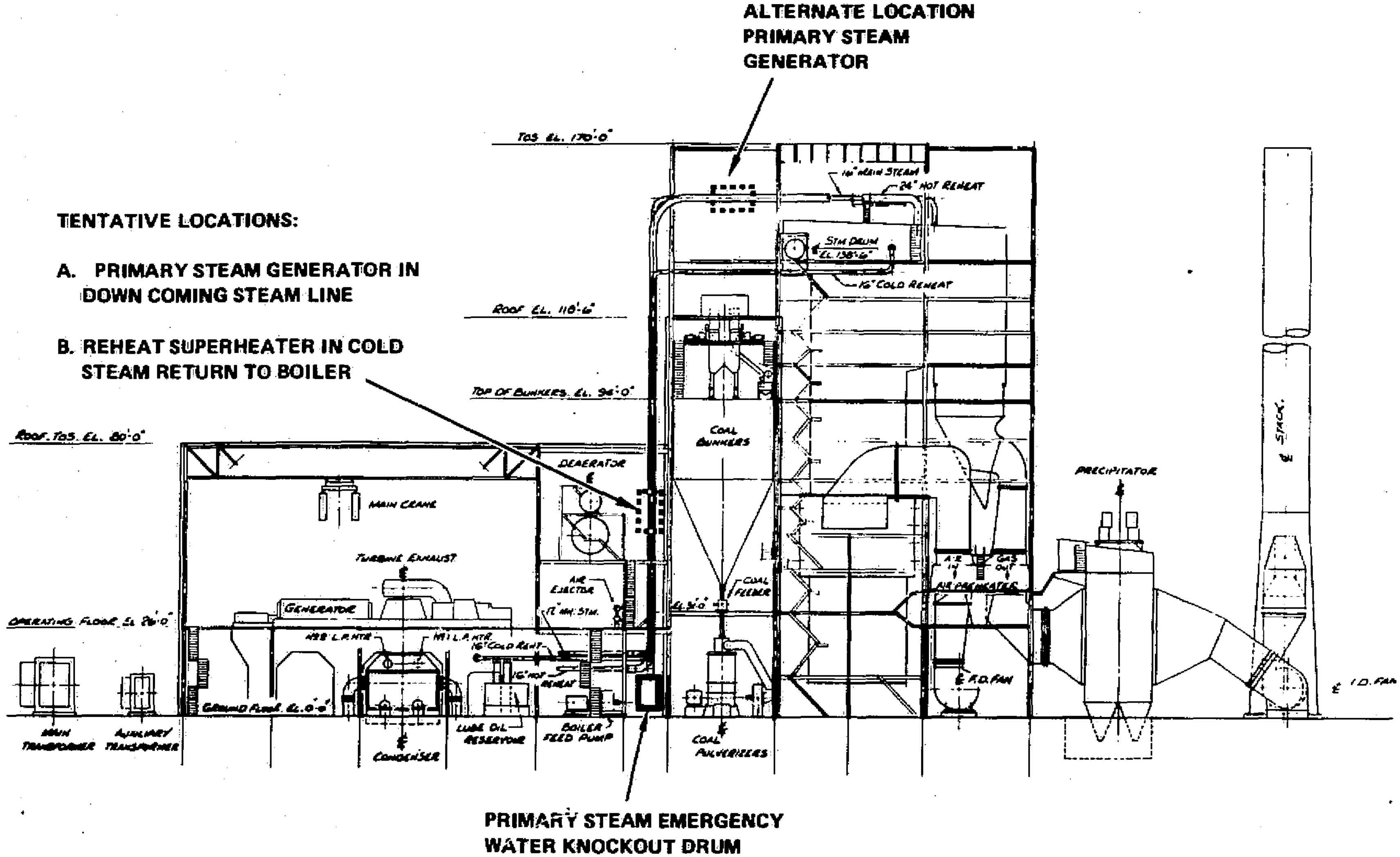

Figure 9. Area Location Showing Elevation

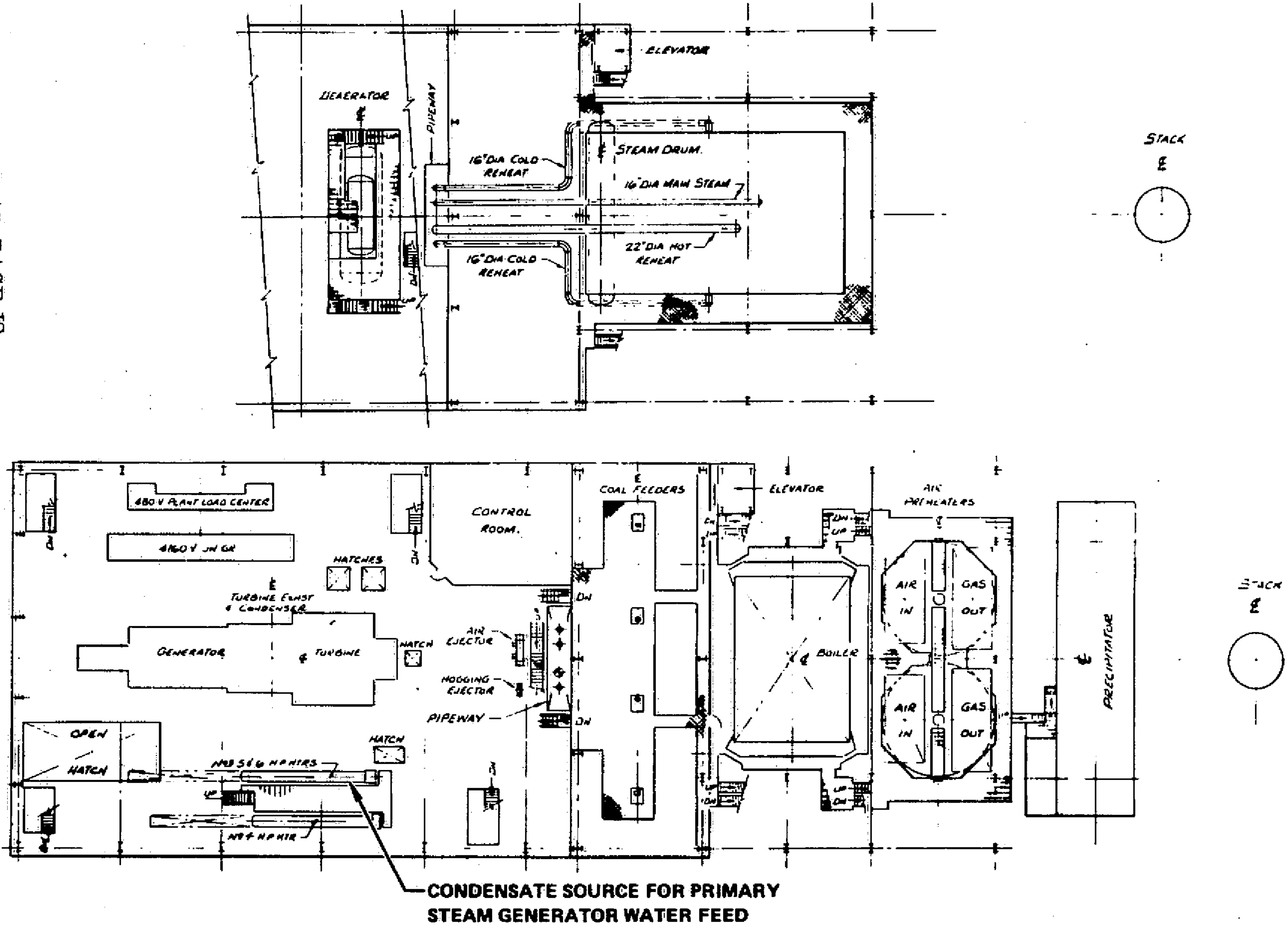

Figure 10. Operating Floor and Roof Plans

condensing pressure, double-flow low-pressure, with 23-inch last-stage bucket length, and a one-row 34-inch pitch diameter governing stage. The feedwater pump is electric-motor driven and has a 75-percent pump efficiency and 81.6-percent drive efficiency.

To facilitate the computations in the analyses, Syntha II Powerplant Design Computer Program was employed (Ref. 1) using the fossil-fuel (oil) steam turbine performance prediction procedure given in Ref. 2 .

Four cases were analyzed which included: (1) the baseline full-load (valve wide open, VWO) case of 161 MW, (2) reduced-load case of 144 MW, (3) full-load with the existing boiler providing heat input to produce 144 MW and the supplemental steam generator providing the remainder, and (4) which is same as case (3), except that the effect of noncondensibles is accounted for by raising the ejector steam flow.

Figure 11 shows the plant arrangement and mass balance for the baseline full-load case with a generator output of 161 MW. The net thermal efficiency (after deducting the feedwater pump power requirement) is 41.77%. The overall plant efficiency, assuming 15% boiler and furnance losses, amounts to 35.50. Figure 12 shows the reduced load case (about 89% of full load). A slight drop in net thermal efficiency is noted because of the decrease in turbine efficiencies. Figure 13 shows the full-load case with the supplementary H_2/O_2 steam generator providing about 11% of the total heat input. The net thermal efficiency is not different significantly from that for case (1) because there is no boiler or furnace loss associated with the H_2/O_2 combustion steam generator. Figure 14 shows the full-load case with the supplementary steam generator, which is the same as case (3), except for taking into account the effect of noncondensible gases. A summary of these cases is presented in Table 11.

In conventional steam powerplants, the only source of noncondensible gases is the air leakage into the low-pressure turbines and the condenser; and the amount is usually negligible. In H_2/O_2 steam generators, however, the noncondensibles are produced in the system either from incomplete combustion of H_2 and O_2 from the mixture ratio control tolerances or from impurity gases carried by H_2 and O_2. For instance, at a combustion efficiency of 98%, the amount of noncondensibles will be equal to 0.02 lb for each pound of combustion steam generated. For operational reasons, the mixture ratio control will be set up to give fuel-rich combustion as explained earlier.

Steam jet ejectors are usually employed for pumping the noncondensibles out from the condenser, mainly because of the availability of low-pressure steam, 4.13×10^5 to 8.26×10^5 N/m^2 (60 to 120 psia), and the recovery of the ejector exhaust latent heat for feedwater heating. The amount of steam required for a twostage steam jet ejector operating at ejector suction pressure of 3.8 to 6.3 cm Hg (1.5- to 2-inch Hg) ranges from 8 to 10 lb per lb of noncondensible. This range would correspond to the 0.2 lb of ejector steam requirement per lb of combustion steam generated (based on a 98% combustion efficiency and 10 lb of steam per lb of noncondensible). Figure 14 shows the ejector steam

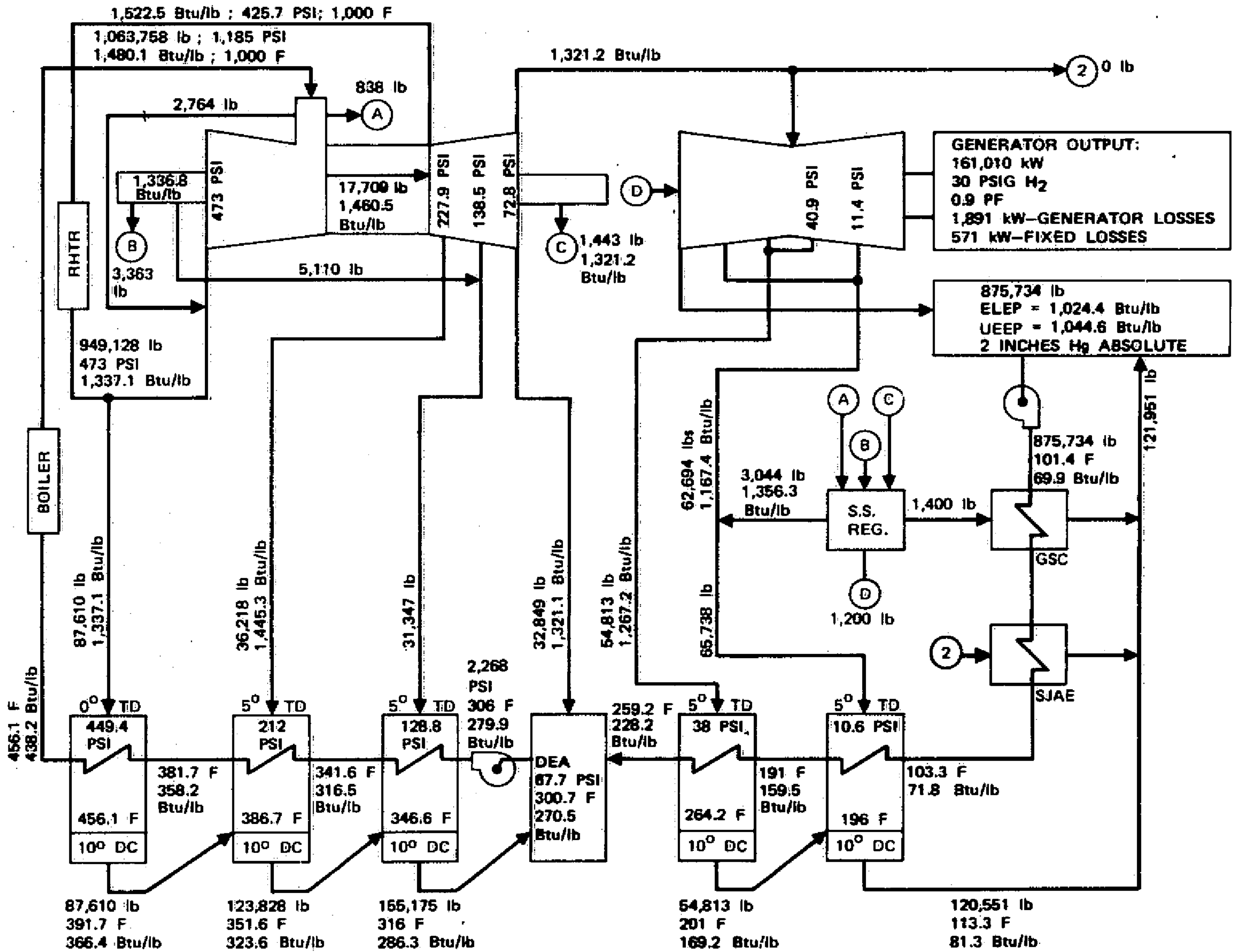

Figure 11. Typical Fossil Steam Turbine Cycle 161 MW, 1815 psi/1000 F/1000 F Full Load Condition

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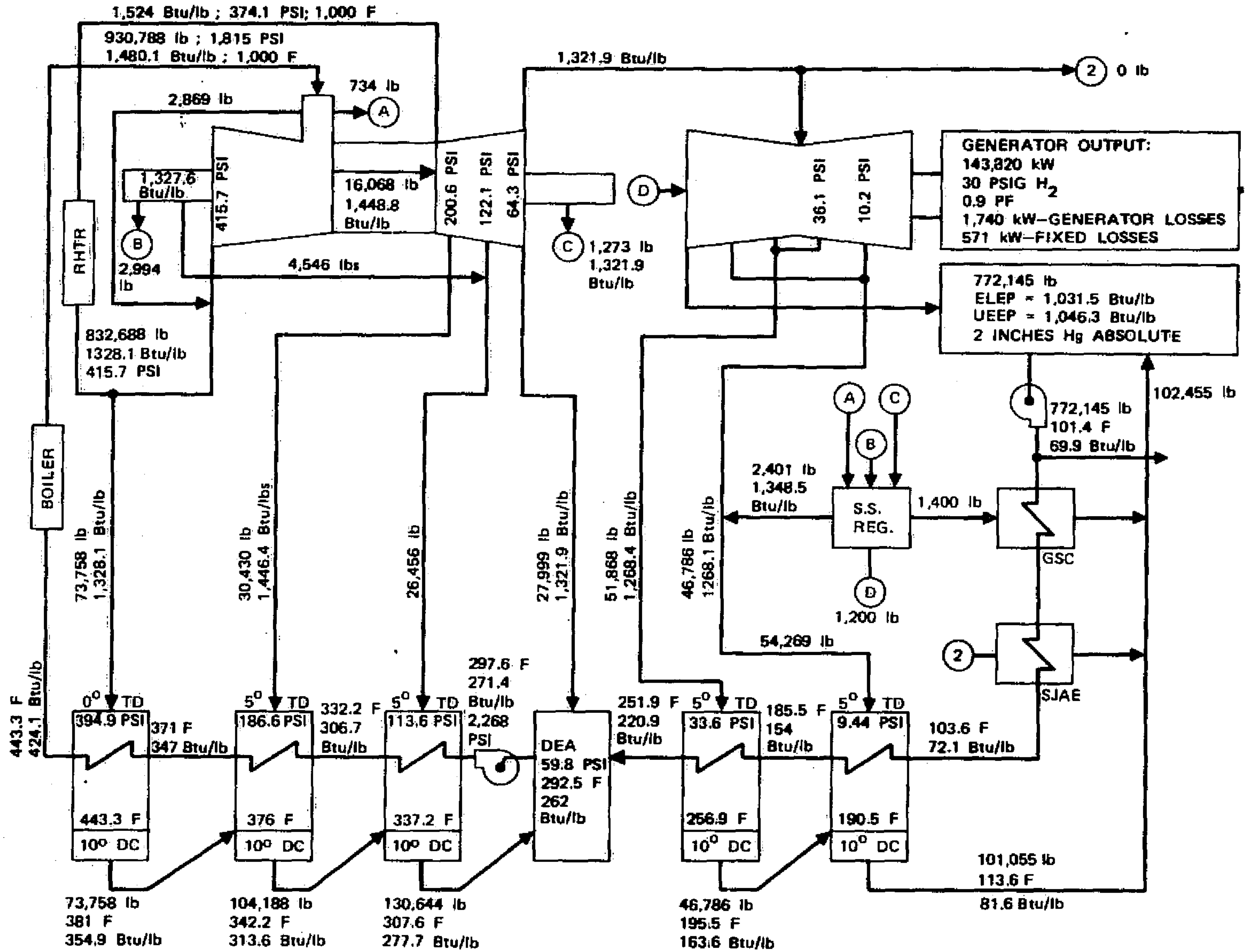

Figure 12. Reduced Load Condition at 143.8 MW

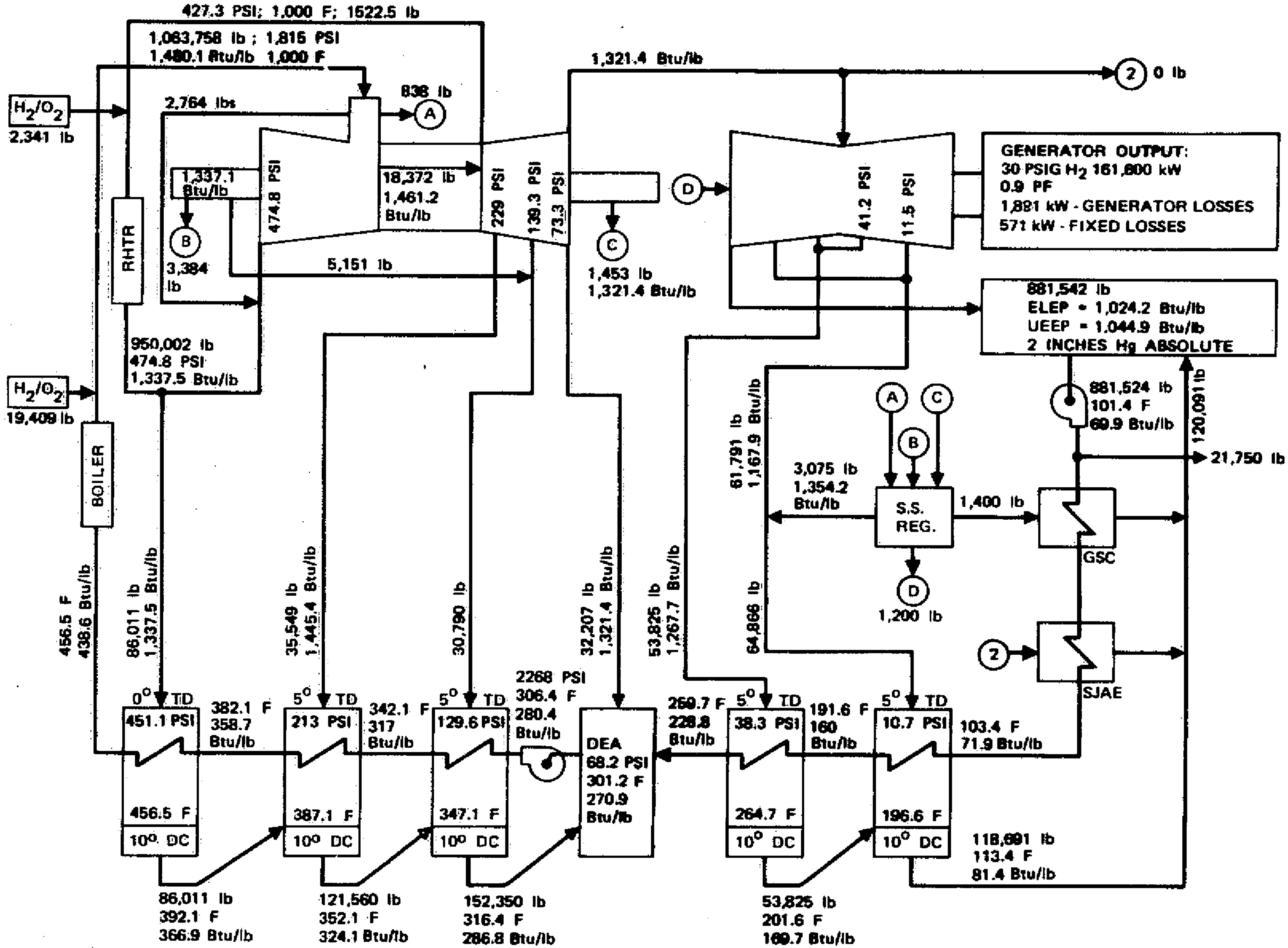

Figure 13 Full Load at 161.6 MW With Supplemental H_2/O_2 Combustion Steam Generator (Zero Ejector Steam Flow)

TABLE 11. SUMMARY BALANCES FOR SUPPLEMENTARY CYCLES (IN MW)

	Case 1		Case 2		Case 3		Case 4	
Bus Bar Output	157.67		140.44		157.73		157.54	
Feedwater Pump Power	3.87		3.38		3.87		3.87	
Generator Output		161.54		143.82		161.6		161.41
Generator + Fixed Losses	2.46		2.31		2.46		2.46	
Turbine Shaft Output	164.00		146.13		164.06		163.87	
Heat Input From Fossile Boiler	376.73		335.73		335.73		335.73	
Fossil Boiler Heat (= 0.85)		554.31		394.98		394.98		394.98
Heat Input From H_2O_2 Combustor	--	--	--	--	43.44	43.44	43.42	43.42
Heat Input From Pump	3.16	3.16	2.76	2.76	3.15	3.15	3.16	3.16
Heat Output From Condenser	-215.89	-215.89	-192.36	-192.36	-218.26	-218.26	-218.44	-218.44
Net Heat Input	164.00	230.48	146.13	205.38	164.06	223.31	163.87	223.12
Net Cycle Thermal Efficiency, %	41.85		41.83		41.6		41.55	
Net Plant Efficiency, %		35.57		35.56		35.98		35.94
Equivalent net Thermal % for Supplementary H_2/O_2 Combustor, % $\left(\frac{\Delta \text{ Generator Output}}{\Delta \text{ Heat Input}}\right)$					40.93		40.49	
Percent of H_2/O_2 Combustor Heat Input, %					11.46	9.91	11.45	9.90

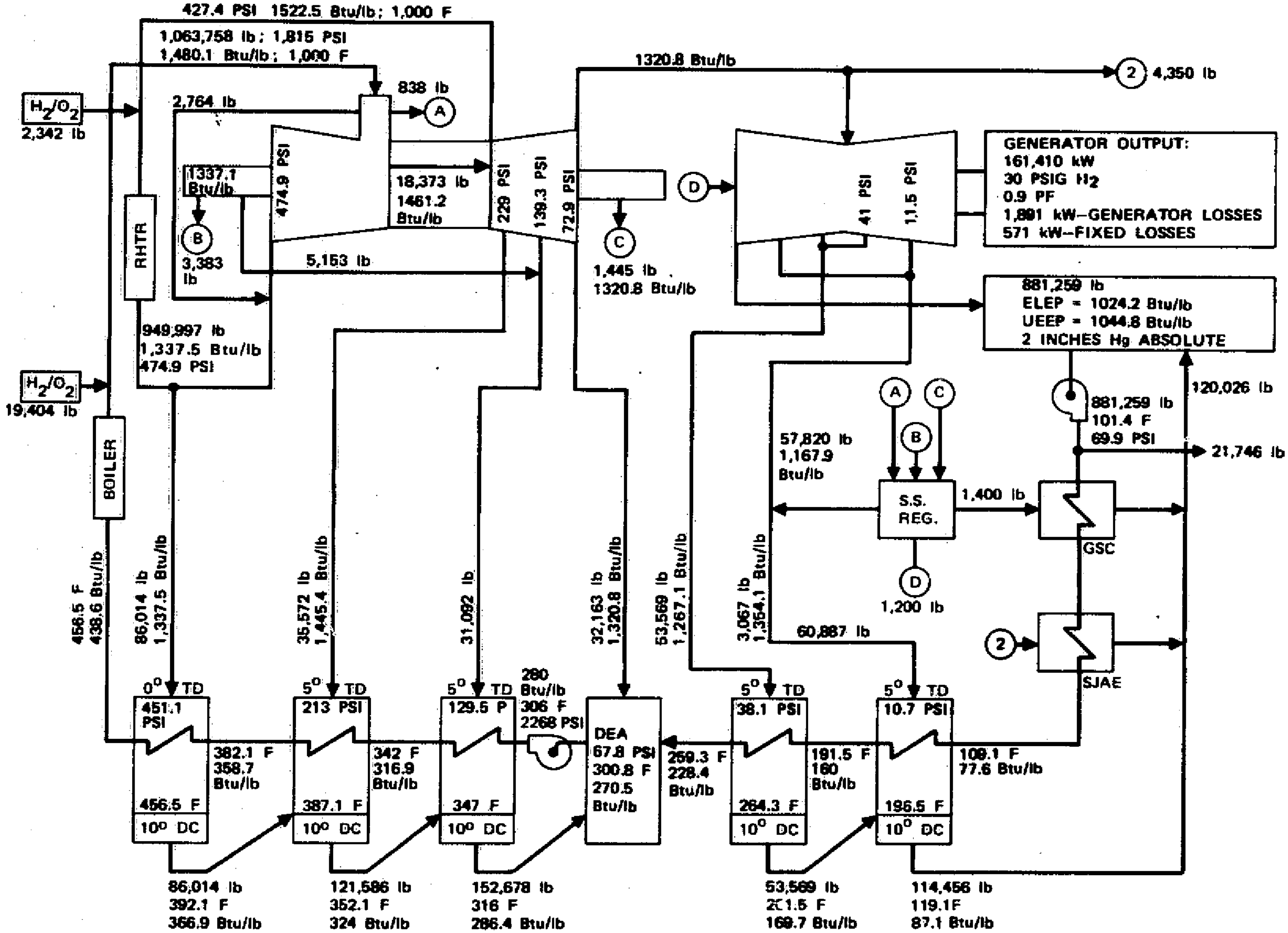

Figure 14. Full Load at 161.4 MW With Supplemental H_2/O_2 Combustion Steam Generator (4350-Pound Ejector Steam Flow)

extracted from the low-pressure turbine exhaust at 6.40×10^5 N/m^2 (92.9 psia) and a rate of 4350 lb/hr. The ejector exhaust is condensed in the steam jet air ejector (SJAE) condenser by the feedwater. Upstream of the SJAE condenser is the gland steam condenser (GSC) which condenses the gland leakoff steam from the low-pressure turbine shaft seal supplied by the sealing steam regulator.

Table 11 gives the summary heat balances for the four cases. Comparison of case (2) with case (3) and (4) shows only a small improvement of about 0.3 points in overall plant efficiency, because the supplementary combustor heat input constitutes only about 11% of the total heat load. The equivalent thermal efficiency for the supplementary $H_2/O2$ steam generator is based on the difference of heat inputs and the difference of power outputs from the partial load case (2). Only a small drop in efficiency is noted with the noncondensible case, which also is due to the fact that the combustion steam flow-rate is only about 2% of the total steam flow.

System Design

The basic steam generation system is designed as a 20-MW supplementary system to be added to an existing 160-MW system. The base system generates 1,098,000 lb/hr of steam at 12.41×10^6 N/m^2 and 811 K (1800 psia and 1000 F). The supplementary system is designed to consume 2350 pounds of hydrogen, 18,800 pounds of oxygen, and 115,000 lb/hr of preheated feedwater to supply an additional 136,000 lb/hr of steam at the same exit conditions. Details of this system are listed in Table 12.

The steam generator is designed for through-flow of the main steam at all times (even when the supplementary system is idle) and for three-step generation of the supplementary steam to minimize system thermal excursions. The hardware is thus fabricated in three major subassemblies: an upstream de-superheater (or atemperater), a hydrogen/oxygen burner-mixer, and a downsteram de-superheater.

The upstream de-superheater follows conventional steam powerplant practice as closely as possible. In actuality this unit may be purchased from a regular industry source. This first unit injects roughly half of the feedwater, depressing the inlet steam temperature 70 F. The water is sprayed into the center of the steam flow with an atomizing spray nozzle. The combined steam and atomized water is accelerated in a Venturi section to further atomize the liquid droplets by aerodynamic shear and to aid in the mixing process. The downstream diffuser section deccelerates the mixed flow to recover the dynamic pressure. All elements of the assembly are provided with heat sheilds to minimize the rate of temperature change and thus avoid thermal fatigue. The entire pipe section is provided with a thermal liner, and a portion of the steam flow maintained adjacent to the wall with little or no temperature change. Slip joints are provided in all thermal shield or liner assemblies to permit differential expansion without structural loading. An access flange is provided to allow removal of the spray for service or inspection without disassembly of the main flowline.

The central section of the upstream de-superheat is the burner-mixing section shown in drawing AP77-213 (Fig. 15). In this section the heat is added to the steam flow by direct combustion of hydrogen and oxygen within the steam flowline.

TABLE 12. COMBUSTOR COMPONENT FOR 20 MWe SUPPLEMENTARY SYSTEM DESIGN CHARACTERISTICS

Primary Combustor

- Diameter, 4.0 in. ID 8.0 OD
- Length, 96-in. Burner = 12 Feet of Sleeve Lined Mixing Section
- Hydrogen Flow 0.653 lb/sec at 2100 psia 1.0 in. ID Approximate Supply Pipe Size
- Oxygen Flow 5.22 lb/sec at 2100 psia 1.5 in. ID Approximate Supply Pipe Size
- Water Flow 31.9 lb/sec at 2100 psia

37.8 136,000 lb/hr

Separator (Cyclone)

- Inlet, 5 in. x 2.5 in.
- Diameter, 15 in. ID 24 in. OD
- Height, 30 in. 48 in. Overall

Reheater

- Pressure, 450 psi
- Inlet Diameter, 7.5 in. ID
- Outlet Diameter, 8.0 in. ID
- Length, 60 in. Burner, No Mixer Length Needed
- H_2 Flow, 0.142 lb/sec at 520 psia, 0.75 in. ID
- O_2 Flow, 1.137 lb/sec at 520 psia, 1.0 in. ID

The burner head is similar to a classic rocket engine ejector assembly that is used with these same reactants as propellants. Both fuel and oxidizer are injected as a high-pressure gas. The combustion mixture ratio is held as closely as possible to theoretical stoichiometric, which produces superheated steam as combustion products that can be mixed directly with the existing steam and resulting in no heat transfer loss.

The burner section in the burner-mixer includes a basically cylindrical section that uses film cooling for the combustor walls. This section is designed to allow the reaction to be almost fully completed before the steam is mixed with combustion products. The further downstream section is patterned after the burner basket of a gas turbine combustor section. This is primarily a mixer section that combines louvers and eyeleted holes to allow steam to penetrate into the core of hot combustion products. Mixing is a combination of penetration and turbulence in the combustor section.

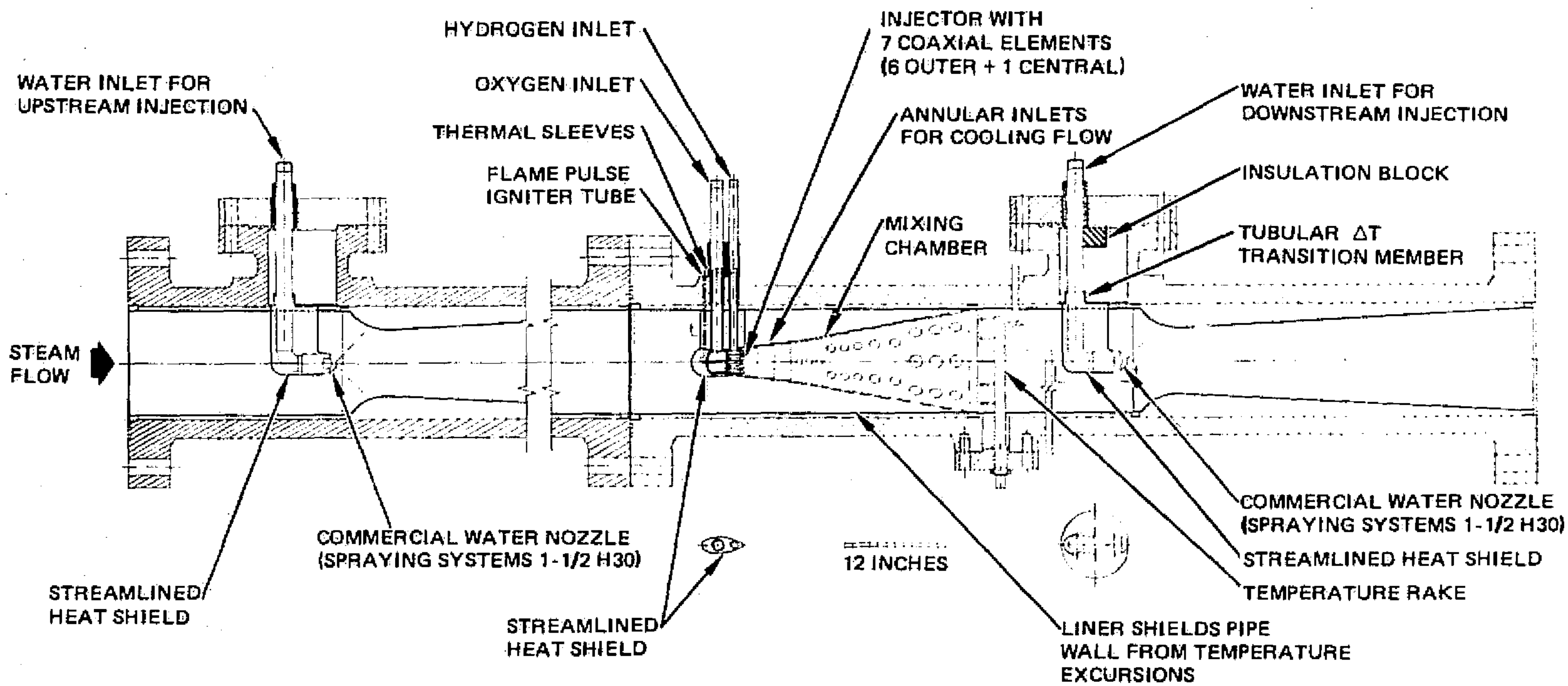

Figure 15. Main Line Supplemental Steam Generator
(Drawing AP77-213)

This hardware section also uses thermal shielding to protect the primary structure from excessive thermal cycling. The temperature rise through this section is limited to 333 K (140 F) by the three-stage process; however, all structure is still protected by the thermal liner that provides a jacket of nominal temperature steam flowing adjacent to the walls. The support and feed structure for the burner sections are enclosed in a streamline strut to provide both aerodynamic and thermal protection for the feedlines and the burner support.

The thermal liner is fabricated of a light-gage corrosion-resistant alloy and is suspended in a slipjoint type of structural assembly to permit assembly and differential structural expansion without any of the burner structure loaded portion yielding or buckling. The burner section proper is an assembly consisting of a small injector for injecting and mixing the H_2/O_2 and a mixer "burner can" section for mixing hot combustion gases with the steam. The mixer structure strongly resembles a gas turbine burner "can," because the functions of both are very similar; these functions are the cooling of the structure with the steam flow and the mixing of the cooler steam with the hot core from the burner.

Thermal fatigue is avoided by using nonloaded thin sections that can heat rapidly and track the temperature changes without building up induced stresses due to thermal gradients. Cooling and mixing are accomplished with a series of louvers and eyeleted holes in the burner basket.

The injector configuration is a multiple-element system that is typical of the H_2/O_2 injectors for rocket engine systems. The coaxial injection element has been selected (similarly as most H_2/O_2 engine systems) because it has demonstrated good performance and good mixing with these reactants. The symmetrical nature of this element also assures against hot streaks in the combusting gases, which might damage the burner walls. The central oxidizer stream is injected at relatively low velocity, with the concentratic hydrogen stream mixing by shear and turbulence as a result of its higher injection velocity.

The burner ignition will be accomplished by a combustion wave ignition system. This system permits the bulk of the equipment to be located outside of the stream flow through the duct. This unit consists of an external chamber in which a combustible mixture of H_2/O_2 is introduced and is flowing through a tube routed to the injector face. This mixture is ignited with an electric spark. The combustion process rapidly pressurizes the chamber and a detonation-like flame front propagates down the tube to the injector face. This energy will ignite a low-flow pilot element that will ignite the balance of the injector flow. The big advantage to this system is that only a single tube need be routed to the injector face, and this tube can be bundled with the reactant feedlines. All major items that may require service, such as valves, spark plugs, electrical leads, etc., are external to the basic system and may be readily serviced.

Downstream of the burner mixer, instrumentation will be added to determine the average heat profile and the enthalpy of the exiting steam and to estimate the amount of cooling water to be added by the de-superheater located downstream of the burner. This de-superheater also may be a purchased because it also is an existing piece of equipment.

The total steam generator assembly will fit in a straight run of the system main stream line with little envelope increase over the existing steam distribution system. A thermal liner section may be added to several feet of the downstream steam line to assure against thermal stress.

Reheater

The reheater burner also will be fitted in line with the base line steam flow and virtually will be identical to the burner section of the steam generator previously described. The primary difference will be the longer and larger diameter mixing "basket" commensurate with the larger diameter of the steam reheat line. The reheat burner is shown in drawing AP77-214 (Fig. 16), and is designed similar to the steam generation burner. For these operating conditions the ignition system and the central burner injection hardware can be identical in size and configuration to those of the generator burner, with the possible exception of some of the injection orifice sizes which would be adjusted slightly to optimize performance at the lower pressure and lower flowrates. This hardware is also fitted with the steam temperature stabilized thermal liner to avoid thermal cycling of the structural duct, even though the steam temperature excursion in this system is within the 283 K (50 F), range allowed by structural code.

Sizes of the equipment needed for this system are provided on the drawings and in Table 12.

Supplementary Steam Generation Economics

The economic evaluation performed for the SSG system is based upon the addition of 20 MWe to an existing 160 MWe plant that has been derated to 140 MWe. Steam conditions for this facility are 12.41×10^6 N/m^2/811 K/811 K (1800 psi/1000 F/1000 F). Costs were determined only for the differential caused by the addition of the H_2/O_2 combustor. The ground rules and criteria specified in the Study Objectives and Criteria section were applied to this evaluation.

Table 13 lists the elements included in the capital cost estimate for the addition of the SSC system. The combustor costs include both a primary and a reheater. As discussed in the Technical Analysis section, it is possible that evaporation of the cooling water may at times be incomplete. Therefore, to protect the turbines, a water separator is included in the cost.

Estimates were made for additional steam and fuel supply lines and for additional control required to operate the combustors and to integrate the combustor system into the facility. No installation time is charged against the system, because, at the advice of consultants, installation would be performed during a routine plant shutdown for other reasons and therefore should not be charged against the system. A 2-day startup was included, however, as a conservative pad to allow for the operational complexities expected at first startup of the system. The result is a capital cost of $127/kW of added capacity for the SSG system.

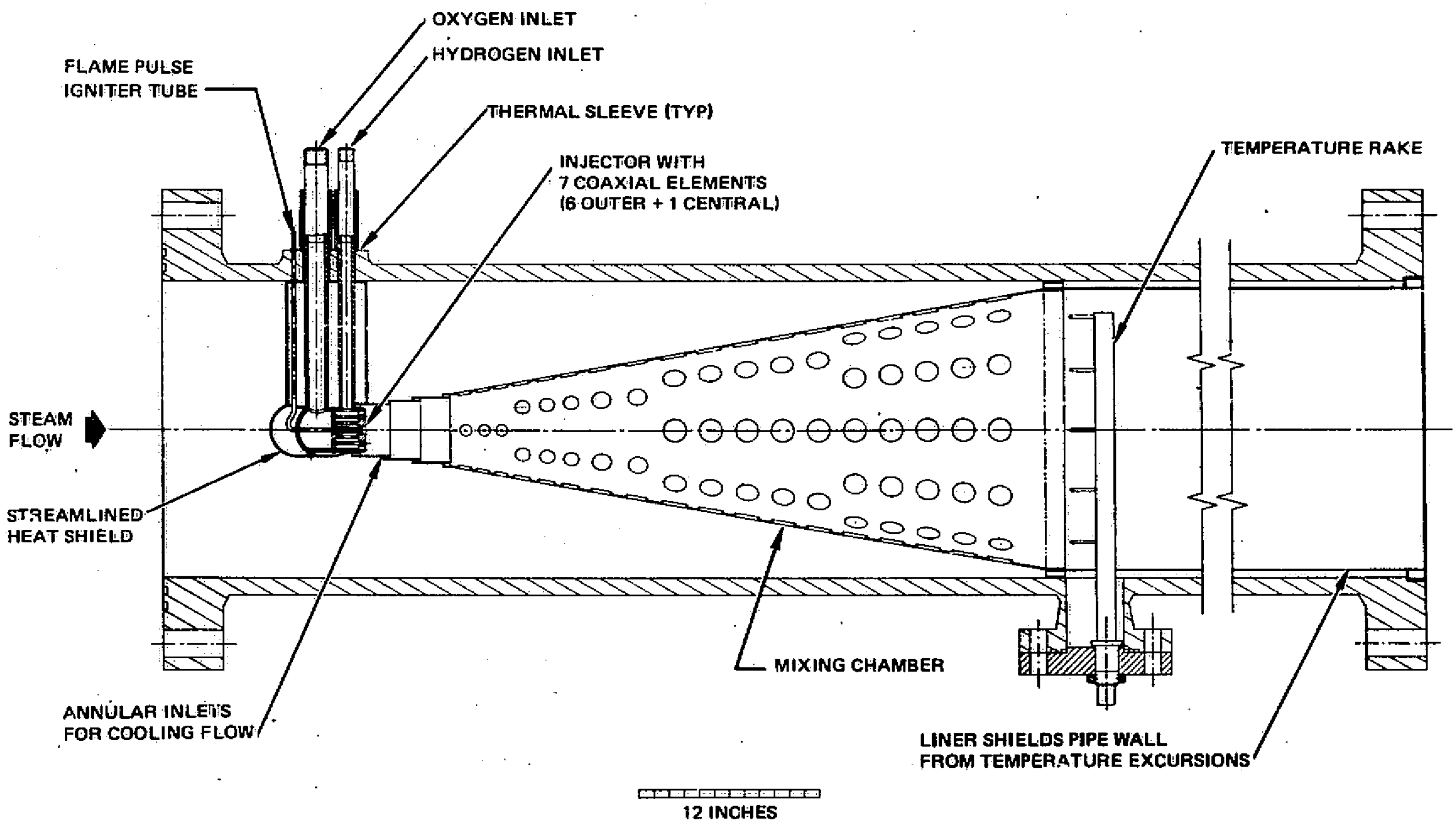

Figure 16. Interstage Reheat Steam Generator
(Drawing AP77-214)

TABLE 13. SUPPLEMENTARY STEAM GENERATION-TYPICAL

- Existing 160 MWe Coal Station Limited to 140 MWe
  Add 20 MWe $h_2/0_2$ Steam Generator and Reheater
- Steam Conditions - 1800 psi, 1000 F/1000 F
- Capital Costs

• H_2 Combustors and Separator	700,000
• Additional Piping, Controls, etc.	1,700,000
• 2-Day Startup	130,000
Total	$2,530,000

- Capital Cost per kW of Added Capacity = $127

To establish a COE for this system, an efficiency of 42% was estimated. This efficiency percentage is less than the thermal efficiency of the system to allow for losses due to increased noncondensibles and for combustion efficiency being less than 100%. No boiler losses are experienced with this system. A design and installation time of 1 year was estimated to obtain a base to determine fixed costs. Table 14 provides a COE detailed breakdown for this system for several duty applications. Labor and maintenance costs were estimated based on experience of our consultants and of Rocketdyne with other related systems. The COE determined for the SSG is the lowest of all applications up to about 3000 hr/yr use; this difference results from the low capital cost and high efficiency (i.e., low heat rate). Thereafter, the SSG and hydrogen gas turbine are very close, mainly because of fuel cost differences.

Figure 17 shows a comparison of COE for a coal plant, a hydrogen-fueled gas turbine, and the SSG system. As indicated, the SSG system results in the lowest COE through intermediate load duty of about 3000 hr/yr. Thereafter, the COE for a coal-fired plant is the lowest.

The low capital cost of the SSG system and the competitive COE makes this comparison attractive for other applications. Some other applications were examined superficially, and the results show sufficient promise to warrant additional effort.

The SSG system provides a capital savings when base load capacity, which must be replaced, is lost at several locations. One can construct a new plant to provide all the lost capacity or add the lost capacity by means of SSG systems added to underrated plan. Table 15 provides a simple example of this application. It is evident that not all factors have been considered, however, the potential savings are large enough to justify further evaluation.

Another potential use of this system is to encourage conversion from oil or gas to coal. A capacity loss is usually incurred when this fuel conversion is made. Table 16 shows a case where the differential cost between coal and oil more than offsets the additional cost of H_2/O_2 used by the SSG system to supplement the capacity loss. The net differential is not a savings, of course, because

TABLE 14. 20 MWe-VARIABLE LOAD SERVICE – SUPPLEMENTARY STEAM GENERATION

	Hours/Year				
	500	1000	2000	4000	7500
Fuel at $6.00/10^6$ Btu x 10^{-6}	0.49	0.98	1.96	3.93	7.37
Labor at \$25/hr x 10^{-6}	0.15	0.20	0.25	0.30	0.50
Maintenance	0.03	0.05	0.05	0.07	0.07
Fixed Cost at 18% (\$127/kW)	0.45	0.45	0.45	0.45	0.45
Total	1.12	1.68	2.71	4.75	8.39
COE, mils/kW-hr	112	84	67.7	59.3	55.9
Heat Rate = 8200 Btu/kW-hr					

TABLE 15. CAPITAL SAVINGS

- Basis
  - 5 to 100 MWe Plants Each Derated 20%
- Options
  - A - Build a New 100 MWe Plant
  - B - Add Capacity With 5 to 20 MWe Supplementary Steam Generator
- Comparison
  - A = At \$755/kW Installed (No Escalation) = $\$75.5 \times 10^6$
  - B = At \$127/kW Installed (No Escalation) = $\$12.7 \times 10^6$
- Options of Supplementary Steam Generation Provides Substantial Capital Savings of 60 to 70%

TABLE 16. OIL-COAL CONVERSION ECONOMICS

- Basis
  - 160 MWe Oil-Fired Baseload Plant
  - Derated to 140 MWe by Coal Conversion
  - 20 MWe Added by H_2/O_2 Supplementary Steam Generaion
- Accounting (1-Year Basis)

• Oil Cost at $\$3.00/10^6$ Btu	$\$42 \times 10^6$	
• Coal Cost at $\$1.00/10^6$ Btu	$\$14 \times 10^6$	
• Savings/Year		$\$28 \times 10^6$
• Cost of H_2/O_2 Installation	$\$1.2 \times 10^6$	
• Added Fuel Cost	$\$8.5 \times 10^6$	$\$9.7 \times 10^6$
• Net 1st Year Differential		$\$18.3 \times 10^6$

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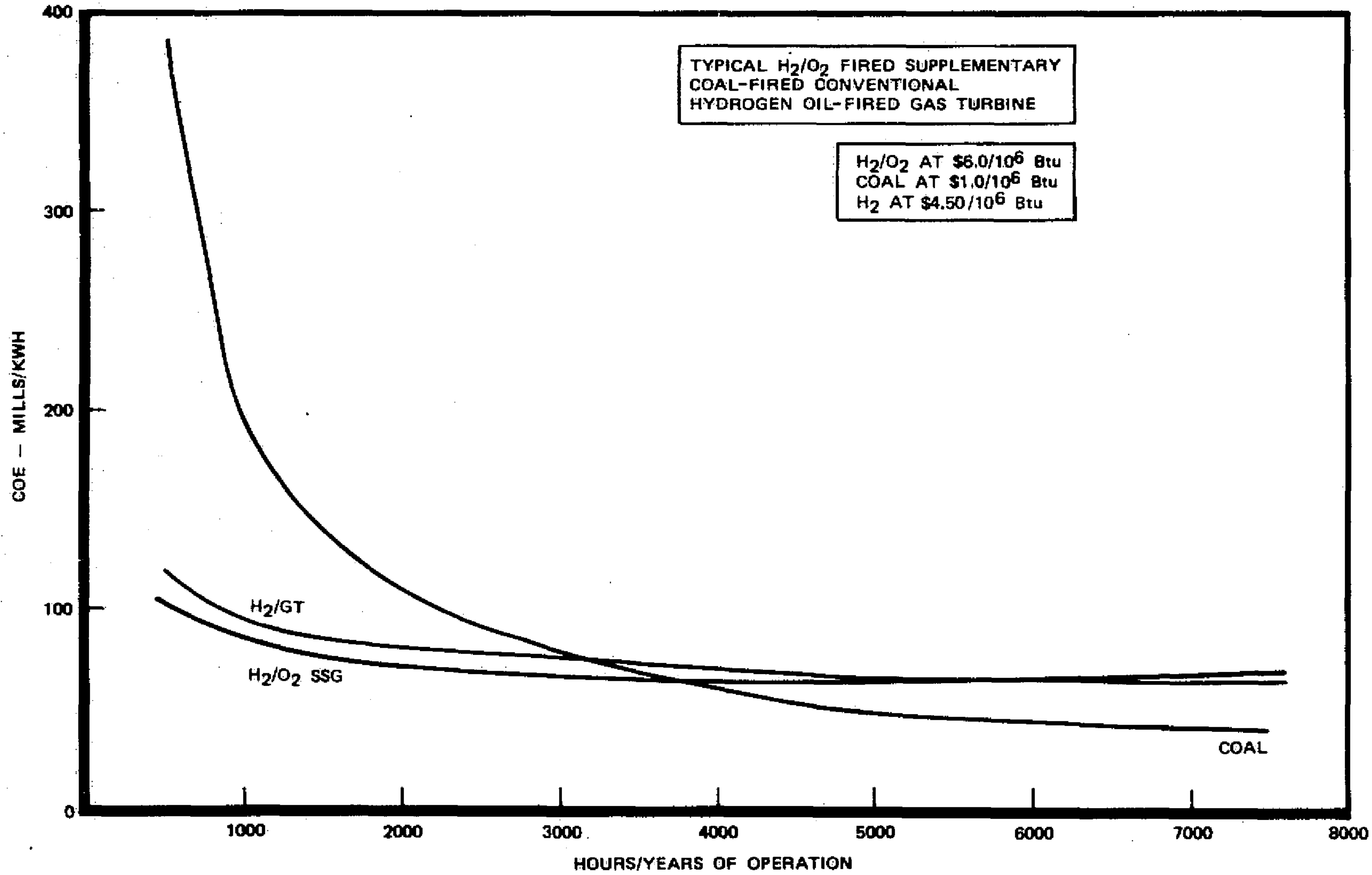
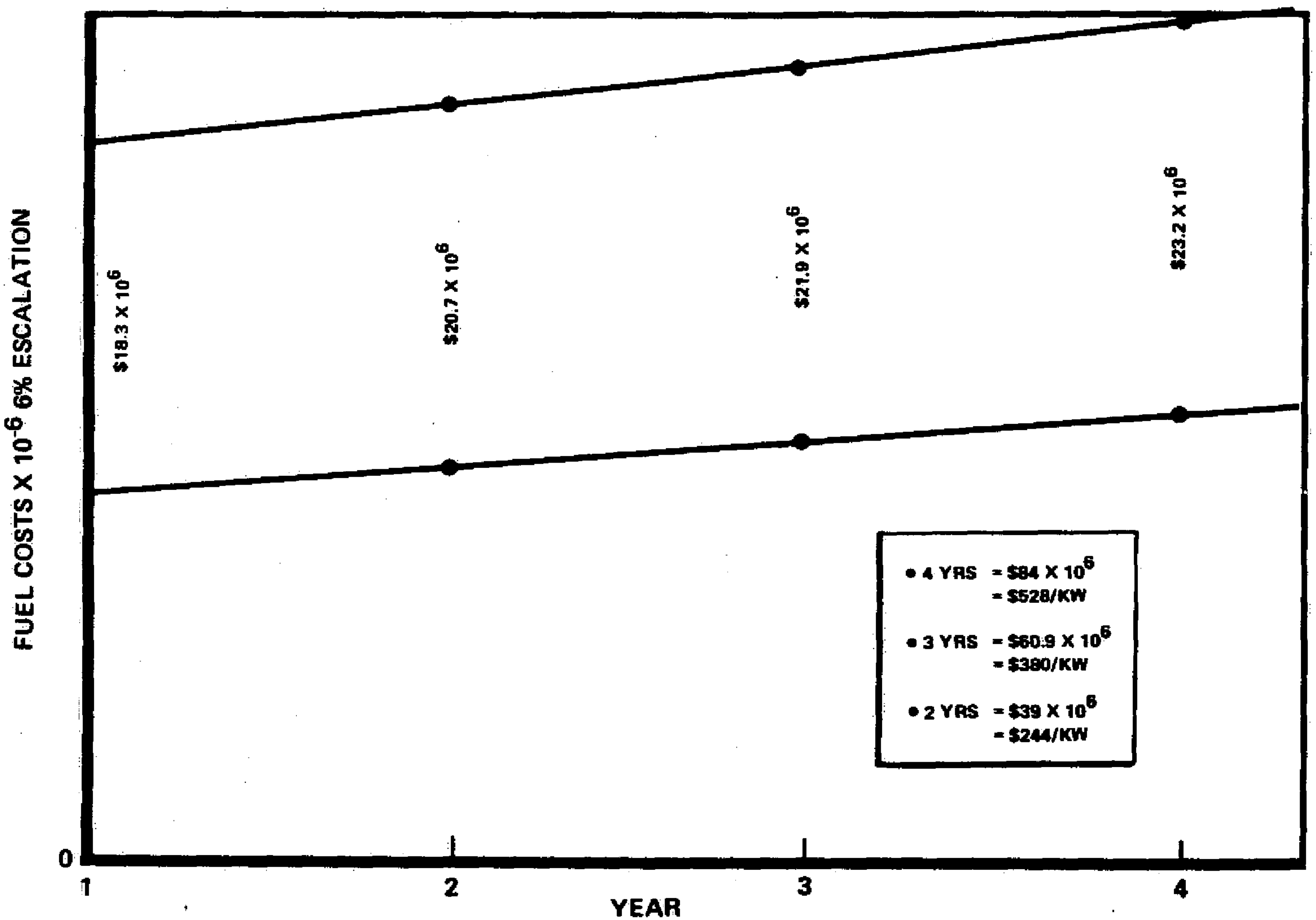


Figure 17. COE Comparison

the cost of conversion and other factors have not been included. Figure 18 shows how the fuel cost differential increases over 4 years due to a fuel cost escalation. Based on these estimates, it appears that the loss in fuel conversion costs can be regained in a 2- to 3-year period. Thereafter, most of the differential can be considered as savings.

Encouraging conversion from oil or gas to coal has an additional benefit in that it assists in the national goal of reducing oil imports. A conservative estimate of the amount of oil used by a 160 MWe plant is 2.5×10^6 bbl/yr. If 30% of the gas- or oil-fired plants converted to coal from oil or gas an equivalent net savings of 10^9 barrels of oil per year could be realized.

These applications are simplistic, of course, but they do indicate sufficient impact to justify further evalation of the supplementary steam generation system for utility use.



PAYBACK SHOULD OCCUR BETWEEN 2-3 YEAR
FASTER PAYBACK WITH LARGER PLANTS

Figure 18. Oil Coal Conversion Payback

ERICSSON CYCLE

System Description and Technical Analysis

The ideal Ericsson cycle consists of two isothermal processes, each of which is followed by a constant-pressure process. By means of heat regeneration between the two constant-pressure processes, the Ericsson cycle can be made to approach the ideal Carnot cycle. In practice, however, the isothermal expansion process can be approximated only with multiple reheats or reburns between expansions, and the isothermal compression process can be approximated with multiple inter-coolings between compressions or constant temperature condensing.

In the steam Ericsson cycle employing direct H_2/O_2 combustion, the multiple reheats can be readily accomplished by means of staged combustors installed in the crossover lines between appropriate turbine-stage groups, while the isothermal compression process would be accomplished by the steam condensing at constant temperature. Direct H_2/O_2 injection into the crossover lines dispenses with the lengthy and costly high-temperature piping leading to and from the reheaters as required in a conventional furnace, and also minimizes the pressure drops across the reheaters. The reheat temperature leaving each combustor can be controlled by varying the amount of H_2/O_2 injection either in stoichiometric or off-stoichiometric ratio. As will be shown later, the former method of attemperation is preferred due to its greater degree of heat recuperation.

Figure 19 shows the schematic diagram of a steam Ericsson cycle with four stoichiometric H_2/O_2 combustors, with 24.13×10^6 N/m^2 (3500 psia) steam throttle pressure and 811 K (1000 F) steam inlet and reheat temperatures. Figure shows the corresponding T-S diagram. The pressure ratio across the first four turbine-stage groups are proportioned to give approximately equal and reasonable enthalpy drops across each stage group. The superheated exhaust steam from the fourth turbine is then cooled to saturation by passing it through a surface-type heat exchanger (recuperator) before it is further expanded down to the condensing pressure in the low-pressure turbine. The heat recuperation is necessary in the steam Ericsson cycle to recover some of the exhaust heat for partial feedwater heating as well as to lower the steam temperature entering the low-pressure turbine for lower condenser loss. The pressure of the exhaust steam at which heat recuperation is to be carried out is determined by consideration of the number of reheats, the steam density and pressure drop across the recuperator, and the size and cost of the recuperator. Obviously, the lower the recuperator exhaust steam pressure, the greater the number of reheats that can be incorporated and the larger the recuperator size will have to be to accommodate the larger steam volume to be handled at a lower exhaust pressure. In this application, an exhaust pressure of 20 psia seems to represent an optimum tradeoff between the number of reheats and the recuperator size.

In addition to the recuperator, conventional steam extraction feedwater heaters are provided at appropriate extraction pressure levels. In the steam Ericsson cycle, shown in Fig. 20, three feedwater heaters are provided with two extracting steam from the low-pressure turbine and one from the exhaust of the No. 1 turbine. The approximate turbine efficiencies were derived from Ref. 2 for a 150 MW steam powerplant. The steam conditions and assigned steam pressure drops across the various components are indicated in Fig. 19. The mass and heat balances are based on .454 kg (1 pound) of feedwater entering the 1-pound combustor.

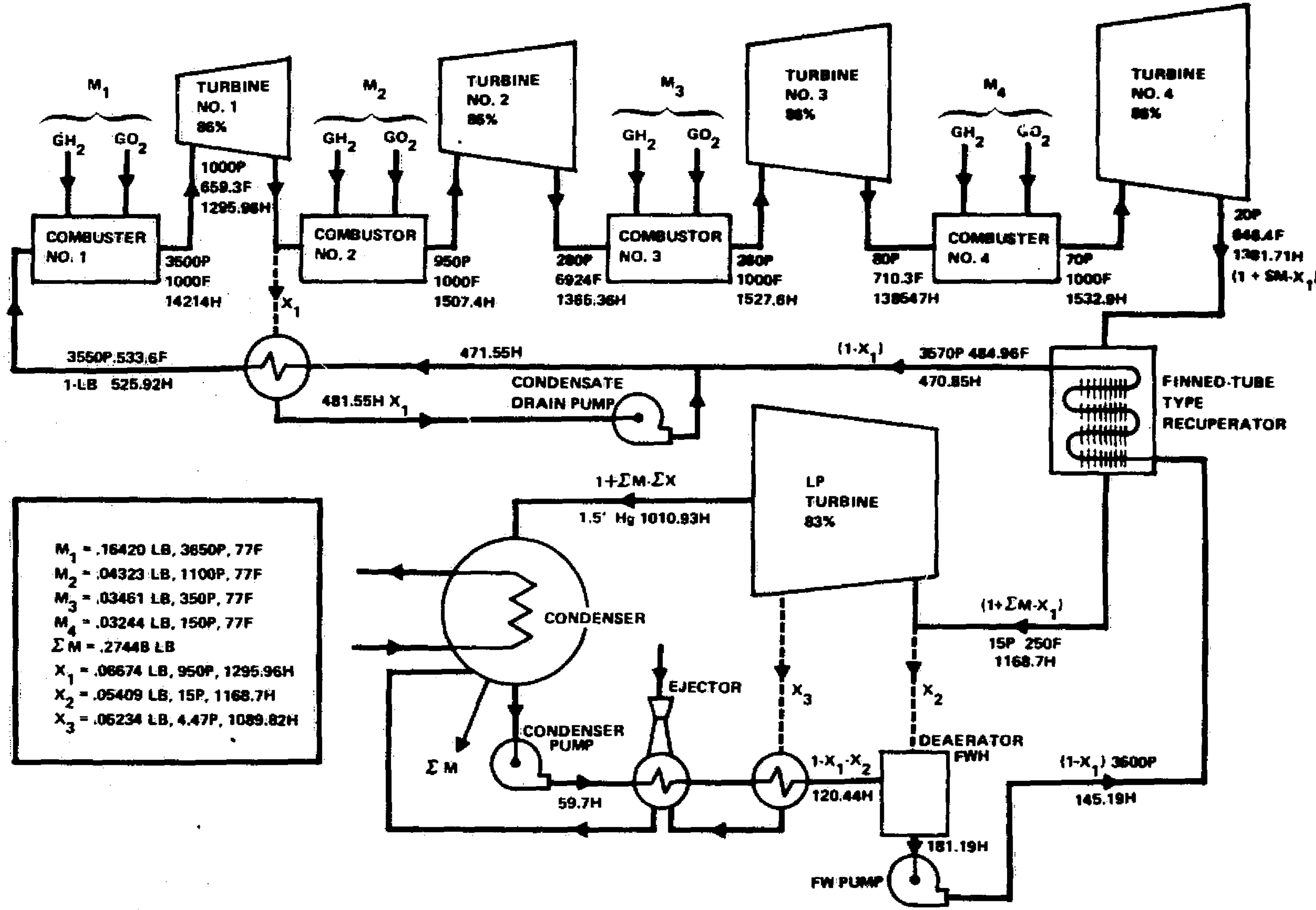


Figure 19. Multiple Reheat, Ericsson Cycle
(3500 psi/1000 F/1000 F/1000 F/1000 F)

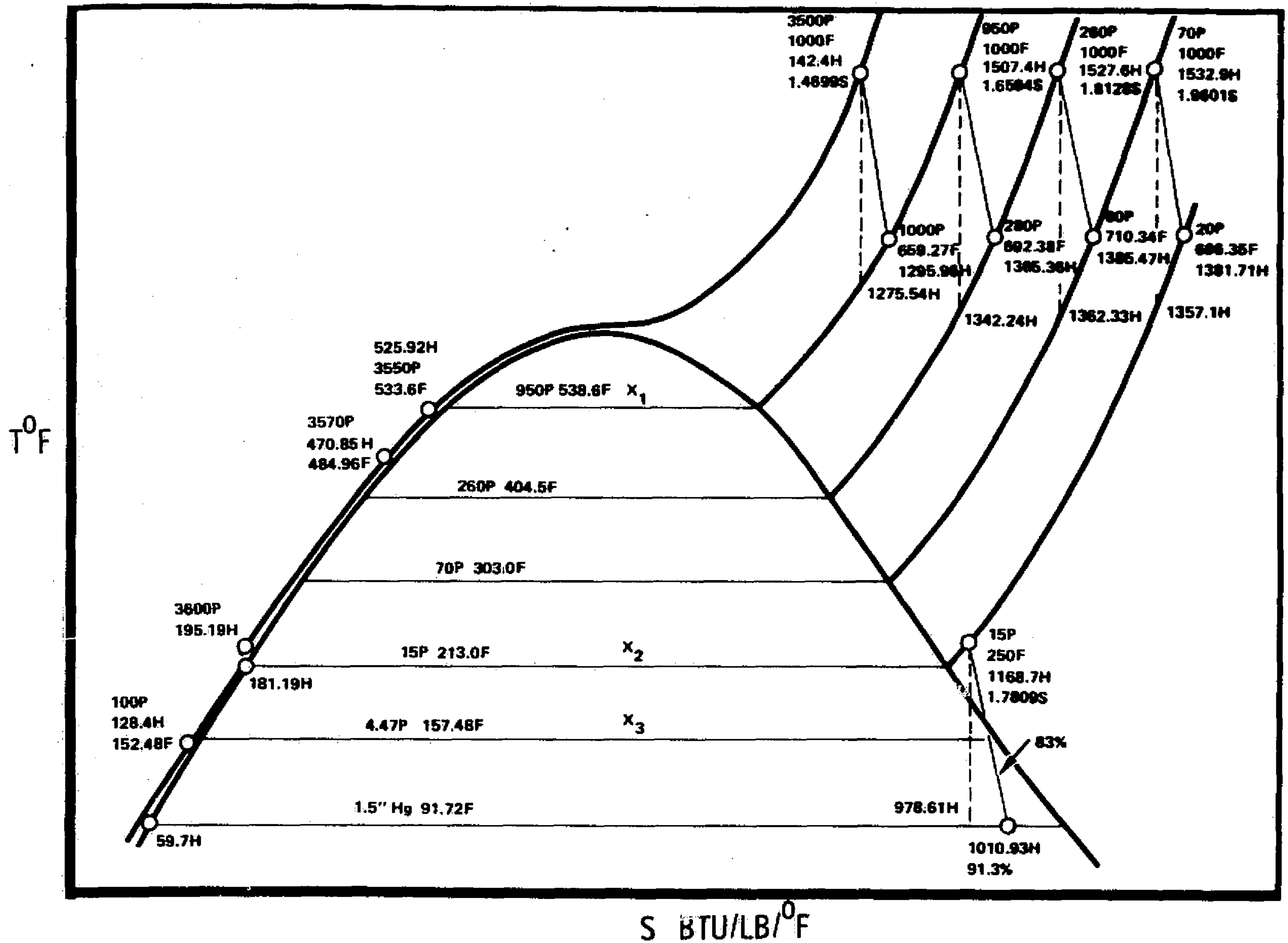


Figure 20. Multiple Reheat, Ericsson Cycle
(3500 psi/1000 F/1000 F/1000 F/1000 F)

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Table 17 lists the heat balance of the Ericsson cycle with 24.13×10^6 N/m^2/811 K/811 K/811 K/811 K (3500-psia/1000 F/1000 F/1000 F/1000 F) steam conditions. The gross thermal efficiency is calculated as the total turbine output divided by the high heating valve (HHV) of the H_2 gas consumption at 292 K (77 F), while the net thermal efficiency is based on the total turbine output minus the feedwater pump power consumption. Although the net (cycle) thermal efficiency of 43.88% for the Ericsson cycle with the given steam conditions appears no better than the net thermal efficiency (44 to 45%) of a conventional modern (single) reheat steam cycle with 24.13×10^6 N/m^2/811 K/811 K (3500-psia/1000 F/1000 F) steam conditions, it must be pointed out that, in the case of the direct H_2/O_2 combustion steam Ericsson cycle, the net cycle thermal efficiency is actually equal to the overall plant cycle efficiency; whereas, in conventional steam plants, boiler and furnace losses must be included to obtain the overall plant cycle efficiency. With H_2-fueled furnaces, these losses may amount to more than 17% of the H_2 HHV because of its high unrecoverable latent heat loss (15.4% of HHV) to the stack. Hence, comparable plant cycle efficiency for a conventional modern steam powerplant with a hydrogen gas-fueled furnace will be around 36.5 to 37.5%.

TABLE 17. MULTIPLE REHEAT, ERICSSON CYCLE*

HEAT INPUT (HHV) TO COMBUSTOR	NO. 1	1121.486	
	NO. 2	295.270	
	NO. 3	236.377	
	NO. 4	221.535	
		1874.668	
CONDENSER LOSS			1048.116
WATER THROWAWAY LOSS			4.038
FEEDWATER PUMP ENERGY		13.066	
POWER OUTPUT			835.580**
		1887.734	1887.734
GROSS THERMAL EFFICIENCY =	44.57%		
NET THERMAL EFFICIENCY =	43.88%		

*3500 psi/1000 F/1000 F/1000 F/1000 F

**Turbine No. 1	144.973
No. 2	162.525
No. 3	167.601
No. 4	182.598
Low-Pressure Turbine	177.883

Stoichiometric combustion of GH_2 and GO_2 is assumed in all combustors. Although off-stoichiometric staged combustion is possible with excess fuel concentration progressively reduced to zero at the last stage, it is not deemed to be as efficient as the stoichiometric combustion case. This can be explained by the fact that the cycle performance improves with the increase of the ratio of the diluent steam flow to the combustion steam flow since the water formed with H_2/O_2 combustion is discarded after the condenser and is not recirculated through the feedwater (or feed steam) circuit. To realize the benefit of exhaust steam recuperation and feedwater heating by steam extractions, a substantial portion of the available heating value of H_2 is used to heat the combustion water from the reference room temperature to the turbine inlet steam condition. With recirculated diluent water, however, the required heating spans only from the recuperated feedwater (or steam) temperature to the turbine inlet condition, thus resulting in higher cycle efficiency with a higher ratio of recirculated diluent water to combustion water flows. In staged combustion, the required water dilution is reduced by the off-stoichiometric combustion and, therefore, it yields the lowest ratio of diluent water to combustion water flows, and gains the least benefit of recuperation. Unless some means is devised to recuperate the GH_2/GO_2 with the exhaust steam heat, this loss of benefit is not recovered.

To determine the effect of turbine steam inlet (and reheat) temperature and inlet pressure on the steam Ericsson cycle efficiency, a parametric analysis of the Ericsson cycle was carried out. The turbine inlet and reheat temperature was varied from 811 K to 1366 K (1000 F to 2000 F) at two inlet pressure levels: 24.13 x 10^6 and 6.89 x 10^6 N/m^2 (3500 and 1000 psia). For the 24.13 x 10^6 N/m^2 (3500-psia) inlet pressure cases, the cycles were provided with five stages of expansion turbines (including low-pressure and four combustors for the 1000-psia pressure cases, four stages of expansion and three combustors were used. The recuperator which is located between the low-pressure turbine inlet and the last intermediate-pressure turbine exhaust, serves to lower the low-pressure turbine inlet temperature and reduce the condenser loss by exchanging heat between the superheated exhaust steam and the feedwater. The reason for locating the recuperator upstream of the low-pressure turbine is to provide a reasonable exhaust pressure so that the steam volume flow and the size of the recuperator will not be excessive.

Figure 21 presents a plot of the Ericsson cycle efficiency versus steam inlet temperature at the two pressure levels of 24.13 x 10^6 and 6.89 x 10^6 N/m^2 (3500 psia), 1366 K (2000 F) inlet steam conditions, the gross cycle efficiency is about 55.7%. The efficiency values at 6.89 x 10^6 N/m^2 (1000 psia) inlet pressure are approximately 4.5 to 5.5 points lower than the corresponding values at 24.13 x 10^6 N/m^2 (3500-psia) inlet pressure, which indicates that there is little advantage of utilizing Ericsson cycles for low steam inlet pressures.

As shown by the plot, the potential of the steam Ericsson cycle is achieving high overall plant efficiency hinges mainly upon the development of high-temperature steam turbines. On a far-term basis, it is conceivable that such

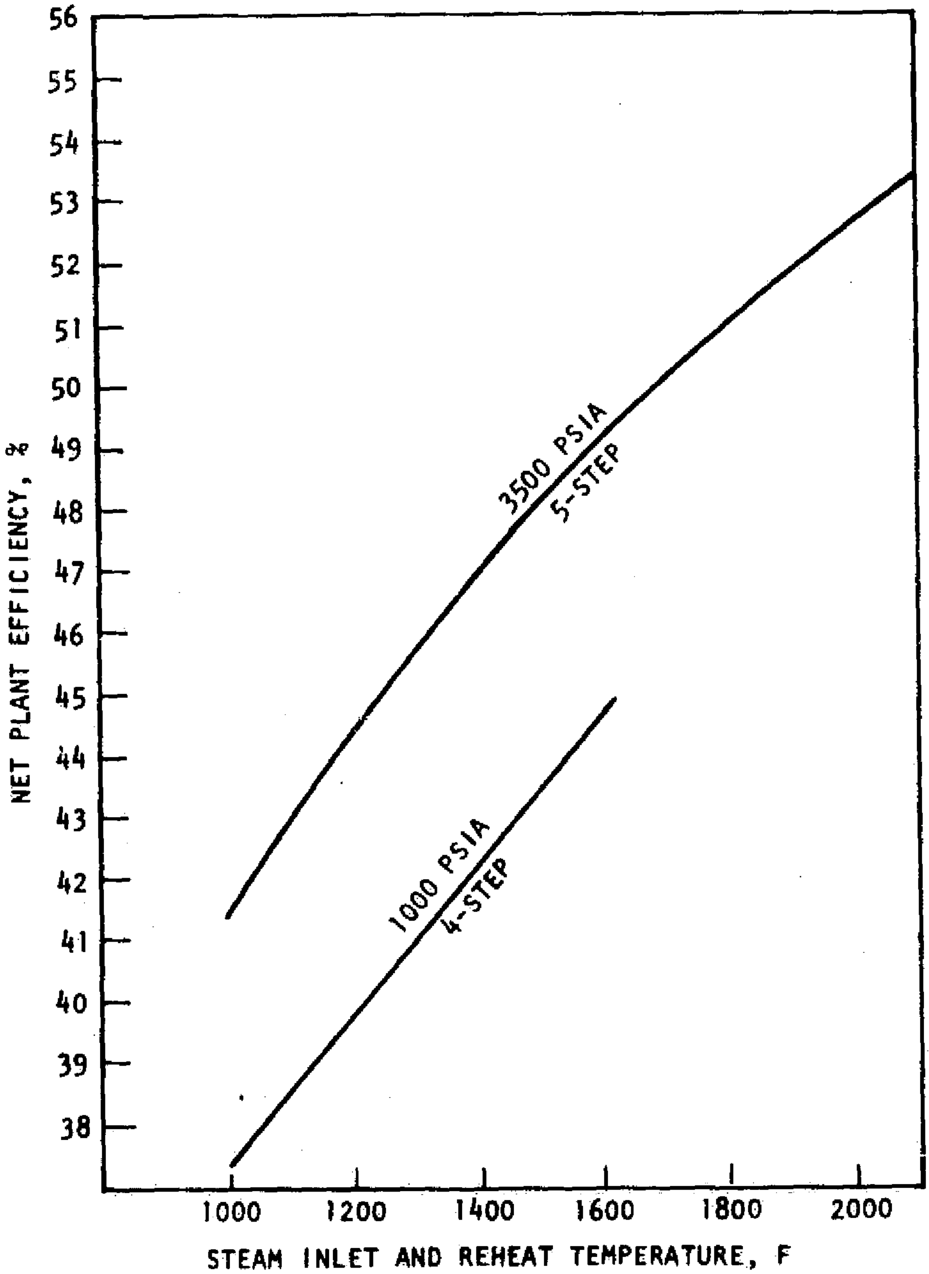

Figure 21. Steam Ericsson Cycle Efficiencies With GH_2/GO_2 Stoichiometric Combustion

steam turbines with ceramic blades operating near 1366 K (2000 F) steam temperature could be developed. Further discussion will be presented in the turbine technology assessment section.

Figures 22 and 23 show the schematic and T-S diagrams of the 24.13×10^6 N/m^2/1366 K/1366 K/1366 K/1366 K (3500-psi/2000 F/2000 F/2000 F/2000 F) steam Ericsson. Table 18 gives the heat balance of the cycle and a net cycle efficiency of 55.2%, which is believed to be higher than can be obtained with H_2 fuel cells (Ref. 3).

TABLE 18. STEAM ERICSSON CYCLE* HEAT BALANCE FOR STEAM CONDITIONS

HEAT INPUT (HHV) TO COMBUSTOR	NO. 1	1336.063	
	NO. 2	469.025	
	NO. 3	467.304	
	NO. 4	471.687	
		2744.080	
CONDENSER LOSS			1224.917
WATER THROWAWAY LOSS			5.910
FEEDWATER PUMP ENERGY		14.000	
POWER OUTPUT			1527.253**
		2758.080	2758.080
GROSS THERMAL EFFICIENCY = 55.66%			
NET THERMAL EFFICIENCY = 55.15%			

*3500 PSI/2000 F/2000 F/2000 F/2000 F

**TURBINE	NO. 1	306.126
	NO. 2	320.320
	NO. 3	328.113
	NO. 4	364.544
LOW-PRESSURE TURBINE		208.150

System Design

The recuperator is one of the critical components that affect the performance and the cost of the Ericsson cycle conversion system. It also replaces functionally some of the steam-extraction feedwater heaters. It is a surface-type heat exchanger with relatively low-pressure steam on one side and high-pressure water on the other. Since the heat transfer coefficient on the steam side is much lower than that on the other side, it is desirable to utilize an extended-surface heat exchanger such as a finned-tube (Fig. 24) or a plate-fin (Fig.25) type of design from the standpoint of size and cost.

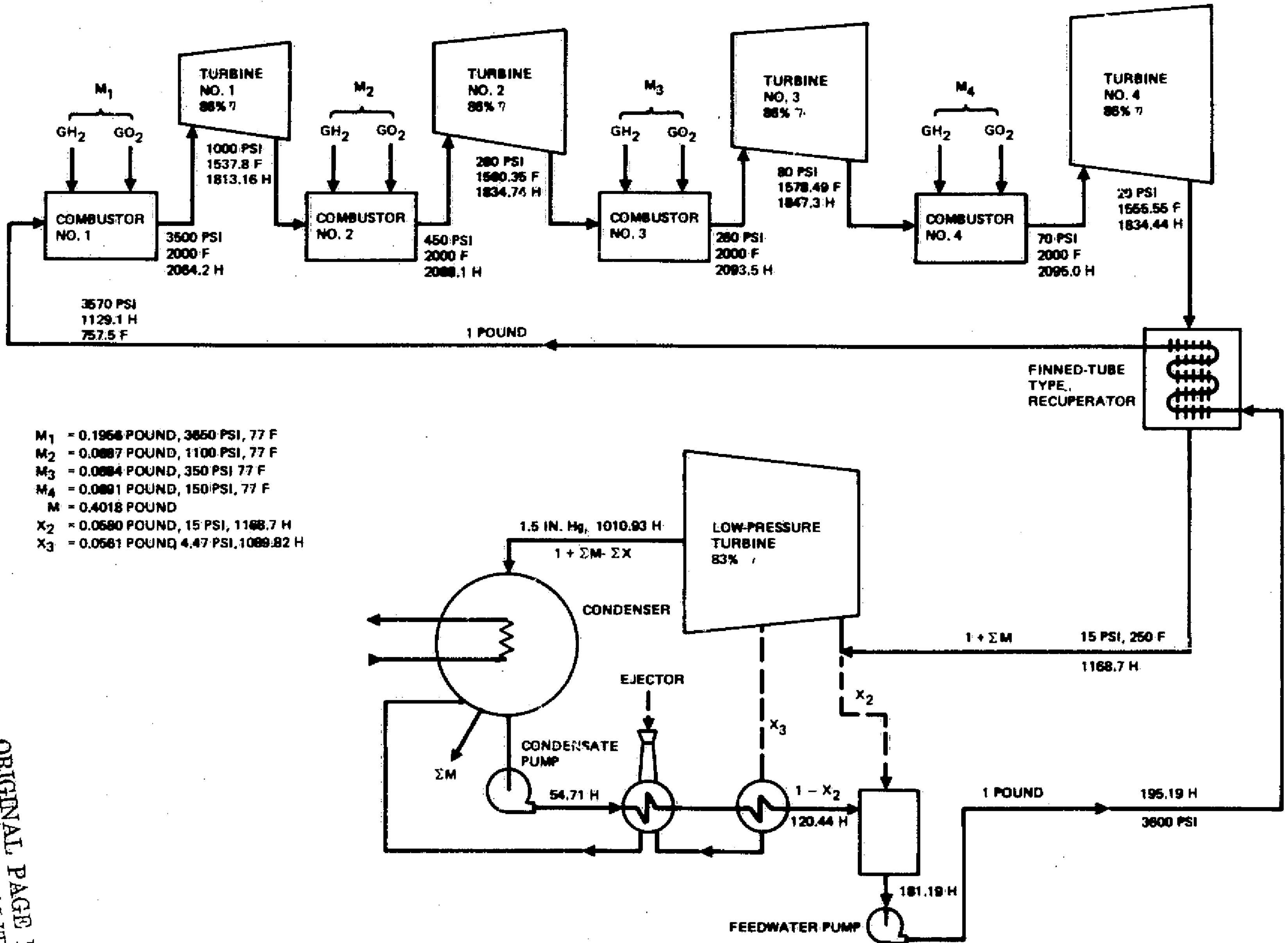

Figure 22. Steam Ericsson Cycle With Multiple Reheats (3500 psi/2000 F/2000 F/2000 F/2000 F)

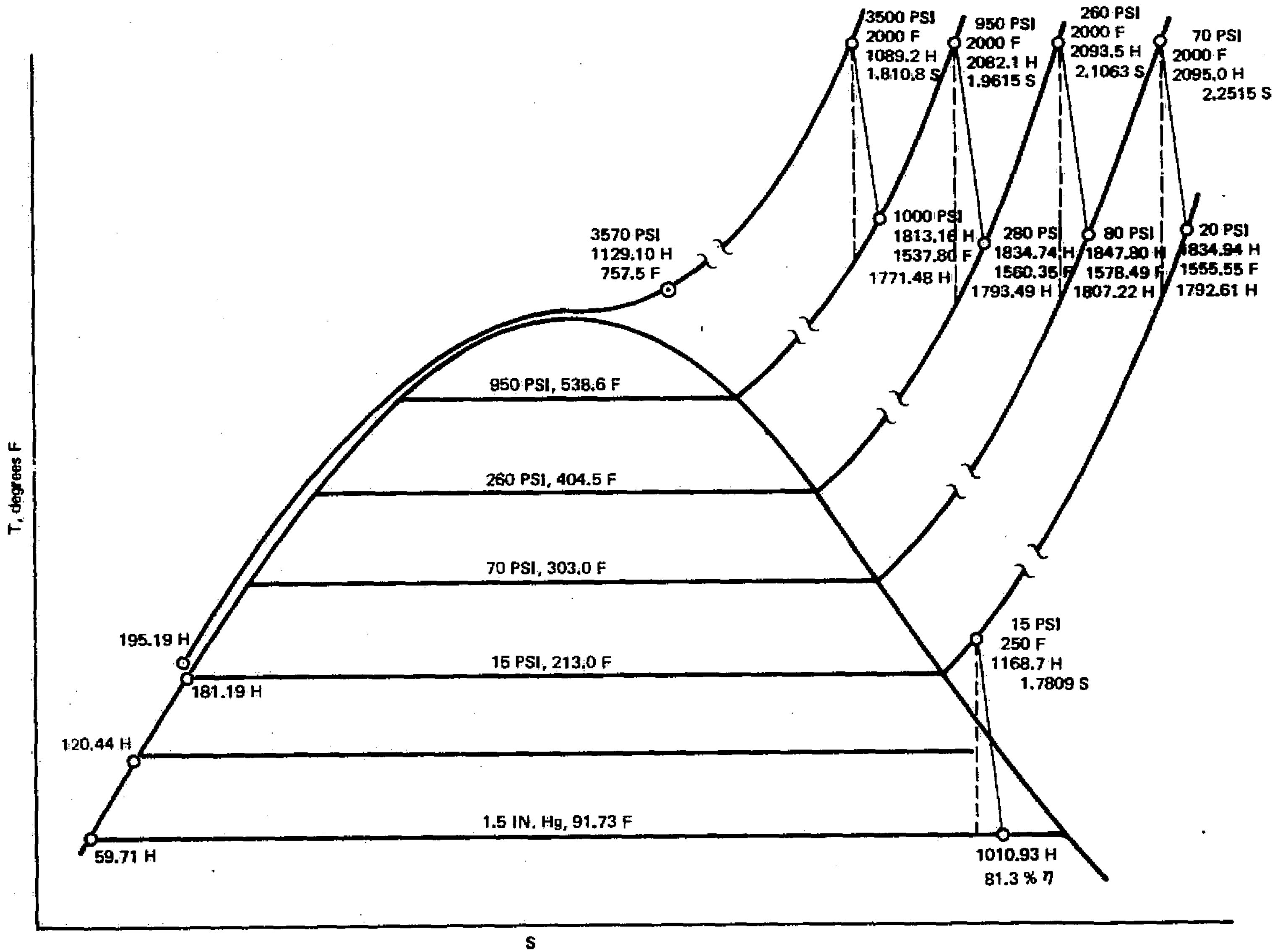

Figure 23. Steam Ericsson Cycle With Multiple Reheats
(3500 psi/2000 F/2000 F/2000 F/2000 F)

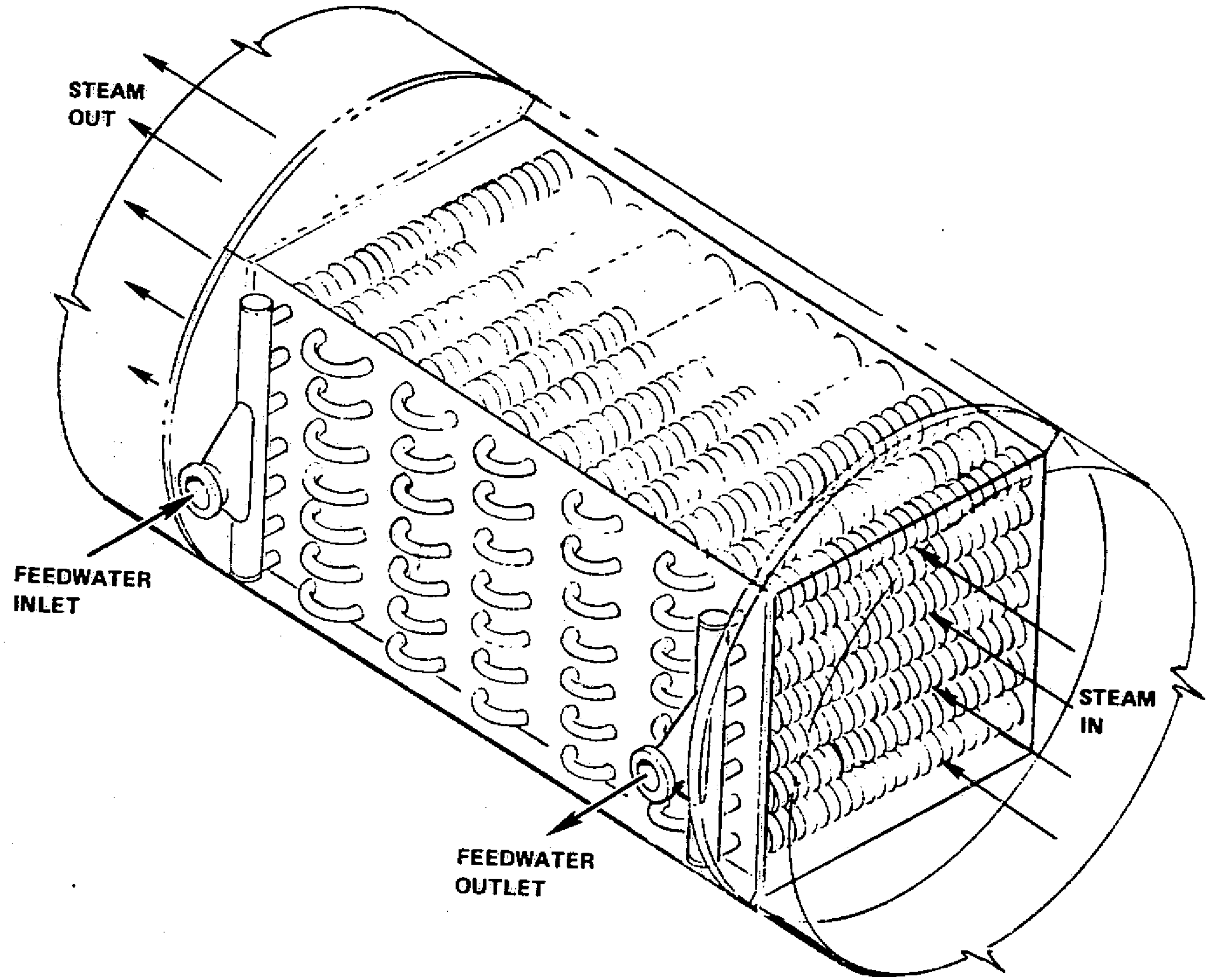

Figure 24. Ericsson Cycle Finned-Tube Recuperator

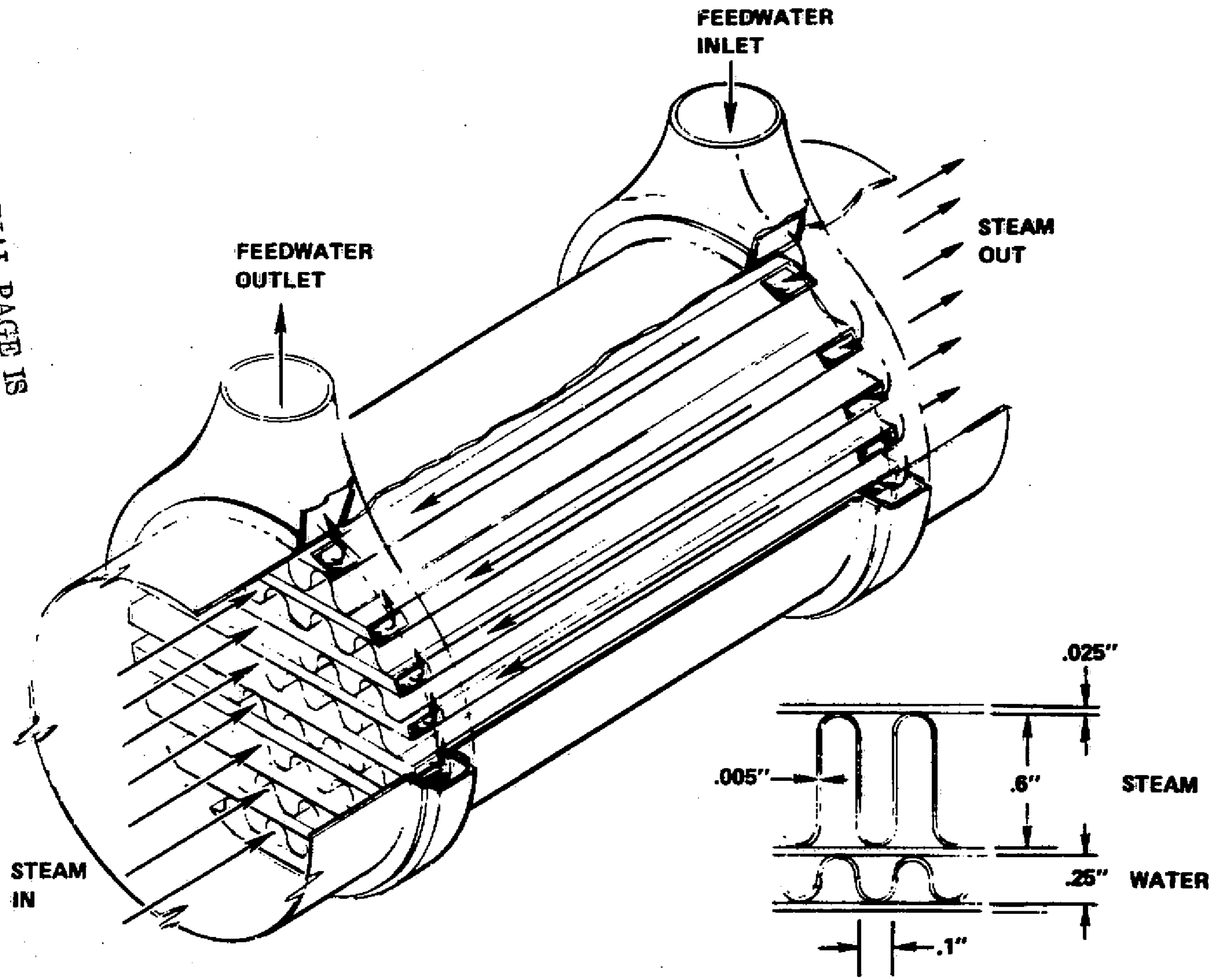

Figure 25. Ericsson Cycle Plate Fin Recuperator

Preliminary design analysis of the recuperator was carried out for a 100 MW power output Ericsson cycle conversion system with five stages of turbines and 3500-psia and 1000 F steam conditions using the heat exchanger computer program. Finned-tube and plate fin-type heat exchangers with cross-counterflow arrangement were analyzed using both copper and steel (Table 19). The initial computer run at rated conditions gave a heat exchanger frontal area of about 4.6 m^2 (50 ft^2) 2.1 x 2.1 m (7 x 7 feet) and a depth of 14 feet with steel fins. With further cycle refinement to allow higher-temperature steam to the low-pressure turbine, a size reduction can be obtained. The most likely recuperator size for this application would be the 4.6 m^2 (50 ft^2) frontal area with a 3.3 m (10.8-feet) length using steel. Copper heat exchangers, although smaller in size, are not considered acceptable in utility systems.

The multiple reheat cycle (Ericsson cycle) direct combustion steam generators and reheaters were sized for 100 MW total output. The No. 1 steam generator of the Ericsson cycle utilizes direct combustion of H_2 and O_2 (assumed to be ambient-temperature gases) to vaporize and superheat feedwater for the total steam flow. Relative flowrates of the input fluids for the (3500 psia, 1000 F) system are:

Feedwater	51.5 kg/s (113.6 lb/sec or 409,000 lb/hr)
Hydrogen	0.94 kg/s (2.07 lb/sec or 7450 lb/hr)
Oxygen	7.5 kg/s (16.58 lb/sec or 59,700 lb/hr)

Direct rocket engine practice for combustion gas velocities would result in a combustor diameter of less than 15.2 cm (6 inches) for these flowrates. The resulting steam velocity would represent too much frictional flow pressure loss, so the burner design diameter was somewhat arbitrarily increased to 25.4 cm (10 inches) to reduce the velocity head of the flow steam to less than 5789 N/m^2 (2 psi).

Under normal operation, the degree of superheat and the heat of the insulated steam piping will ensure that no liquid water will be present in the turbine inlet flow. However, at off design operation, or during startup or shutdown transients, some liquid may be delivered by the burner, making the inclusion of a steam-liquid separation system a recommended part of the system. A centrifugal cyclone-type assembly will probably be the best selection for this application, with a tangential entrance velocity of roughly 30.5 m/s (100 ft/sec); equivalent line ID = 23.6 cm (9.3 inches). The resulting cyclone will be 0.9 to 1.2 m (3 to 4 feet) in diameter and 1.2 to 1.8 m (4 to 6 feet) high. The structure will be insulated internally with a "thermal shield" of relatively light gauge material to minimize thermal shock. Some pilot burner-type steam flow is recommended during idle mode operation to avoid thermal shock to system components when the system is rapidly brought up to operating levels.

The steam reheaters for the 811 K (1000 F) reheat cycle were sized for a 100 MW system. The primary sizing criterion for these burners is the steam velocity, which results in an increasing burner diameter as the pressure is reduced in

TABLE 19. DIMENSIONS OF ERICSSON CYCLE RECUPERATOR

Type	Fin Material	Water Temperature, F	Steam Temperature, F	Heat Transfer, Btu/ft^2-hr-F		Steam-Side Heat Transfer Area, ft^2	ΔP, psi		Dimensions		Weight pounds
				Water	Steam		Water	Steam	Frontal, ft^2	Length, feet	
Plate Fin	Steel	486	248	258	49.4	47,977	0	2.99	35.3	7.7	29,600
	Copper	486	249	232	44.4	42,988	0	1.83	34.7	6.8	13,825
	Copper	485	250	333	63.3	31,538	0	5.19	22.1	7.8	10,143
Finned Tube	Steel	485	250	2521	51.7	64,015	22	5.61	50	14.1	26,862
	Copper	484	249	2516	51.7	50,743	17	4.23	50	11.2	22,808
	Copper	485	250	2515	42.0	55,740	19	1.71	70	8.8	25,054
	Copper	484	247	2506	33.7	64,015	22	0.71	100	7.0	28,774
Finned Tube	Steel	474	268	2507	51.9	49,182	17	4.3	50	10.8	20,638
	Copper	473	267	2501	51.9	39,033	13	3.28	50	8.6	17,545
	Copper	475	270	2502	42.2	42,624	14	1.34	70	6.7	19,159
	Copper	474	268	2496	33.8	48,401	16	0.55	100	5.3	21,756

Water Inlet Temperature = 218 F   Steam Inlet Temperature = 696 F in the later stages. The burner diameter is not much larger than the ducting required to extract and reinject the steam flow between the appropriate turbine stages. The duct sizes in the selected 100 MW system progress vary from roughly 27.9 cm (11 inches) inside diameter at the first reheat point to about 101 cm (40 inches) inside diameter at the final reheat stage (Table 20).

The physical layout of the direct combustion reheat burner resembles a gas turbine combustor in many respects, since the requirements are very similar. Both applications depend on a local, high-temperature combustion zone with dilution and mixing downstream to provide the desired uniform gas temperature delivered to a turbine stage. In one case the working fluid is air, which also provides the oxidizer for the combustion process, and in the other case the working fluid is steam, which is independent of the combustion reaction.

The steam reheater utilizes a central injector element where the H_2 and O_2 are injected and combusted. This injector element is typically less than one-fourth of the duct diameter. The combustion process is carried out at stoichiometric mixture with the high core temperature it provides. Steam is used as a film coolant in the combustor section, being brought in axially along the combustor walls by a series of slots and louvers. At the length where combustion is complete, the mixing section is initiated, with the steam being introduced as radial jets to penetrate and mix with the hot combustion gases. The basic concept and rough size of the last low-pressure combustor are shown in Fig. 26. This concept is very similar to the combustor for the supplementary steam generation shown earlier.

The Ralph M. Parsons Co. was furnished with a description of the planned PCS along with the data of Table 20 for the sizing of the steam generator and the three steam reheaters. A 100 MW installation size was chosen for this evaluation. Differential costs between a 100 MW H_2/O_2 advanced steam cycle installation and a 100 MW conventional coal-fired installation operating at $12.41 \times 10^6 N/m^2$ (1800 psi) were then established.

This portion of the study involved conceptual layouts of the 100 MW Ericsson cycle installation (for aid in cost estimation). These conceptual layouts are presented in Fig. 27 and 28. The layouts bring out the compact nature of a H_2/O_2 PCS as compared to a coal-fired installation, (whose boiler house alone would be substantially larger than the whole Ericsson cycle plant, and which would also require large plot areas for coal and ash handling, coal storage, and stack gas scrubbing).

Ericsson Cycle Economics

The cost of electricity (COE) for the Ericsson cycle was compared to a conventional coal-fired plant. A 100 MWe plant was used in each case, as suggested by contract, which results in a higher COE for each system. The comparative relationships are valid however.

Capital cost calculations were made as shown in Table 21. A three reheat Ericsson cycle ($24.13 \times 10^6 N/m^2$/811 K/811 K/811 K/811 K) (3500 psi/1000 F/1000 F/1000 F/1000 F) and a reheat steam coal fluid system ($12.41 \times 10^6 N/m^2$/811 K/811 K) (1800 psi/1000 F/1000 F) were evaluated. Capital costs include an SO_2 scrubber

TABLE 20. COMBUSTOR COMPONENTS FOR 100 MWe ADVANCED STEAM (ERICSSON CYCLE) DESIGN CHARACTERISTICS

- Primary Combustor
  - Diameter: 7.0-inch ID (12- to 14-inch OD)
  - Length: 96 inches + 20-foot sleeve-lined mixing section
  - Hydrogen Flow: 2.07 lb/sec at 3900 psia (1.5-inch ID line)
  - Oxygen Flow: 16.58 lb/sec at 3900 psia (-2.0-inch ID line)
  - Water Flow: 113.6 lb/sec
- Separator (Cyclone)
  - Inlet: 8 x 4 inches
  - Diameter: 30-inch ID (4-foot OD)
  - Height: 40-inches (6 feet overall)

Reheaters	No. 1	No. 2	No. 3
Pressure	950 psia	260 psia	75 psi
Inlet Diameter	10 inches ID	20 inches ID	37 inches ID
Outlet Diameter	11 inches ID	21 inches ID	39 inches ID
Length	60 inches	72 inches	96 inches
H_2 Inlet Diameter	1.0 inches ID	2.0 inches ID	3.0 inches ID
O_2 Inlet Diameter	1.5 inches ID	3.0 inches ID	4.0 inches ID
H_2 Weight Flow	0.55 lb/sec at 1100 psi	0.44 lb/sec at 300 psia	0.41 lb/sec at 88 psia
O_2 Weight Flow	4.37 lb/sec at 1100 psi	3.49 lb/sec at 300 psia	3.28 lb/sec at 88 psia

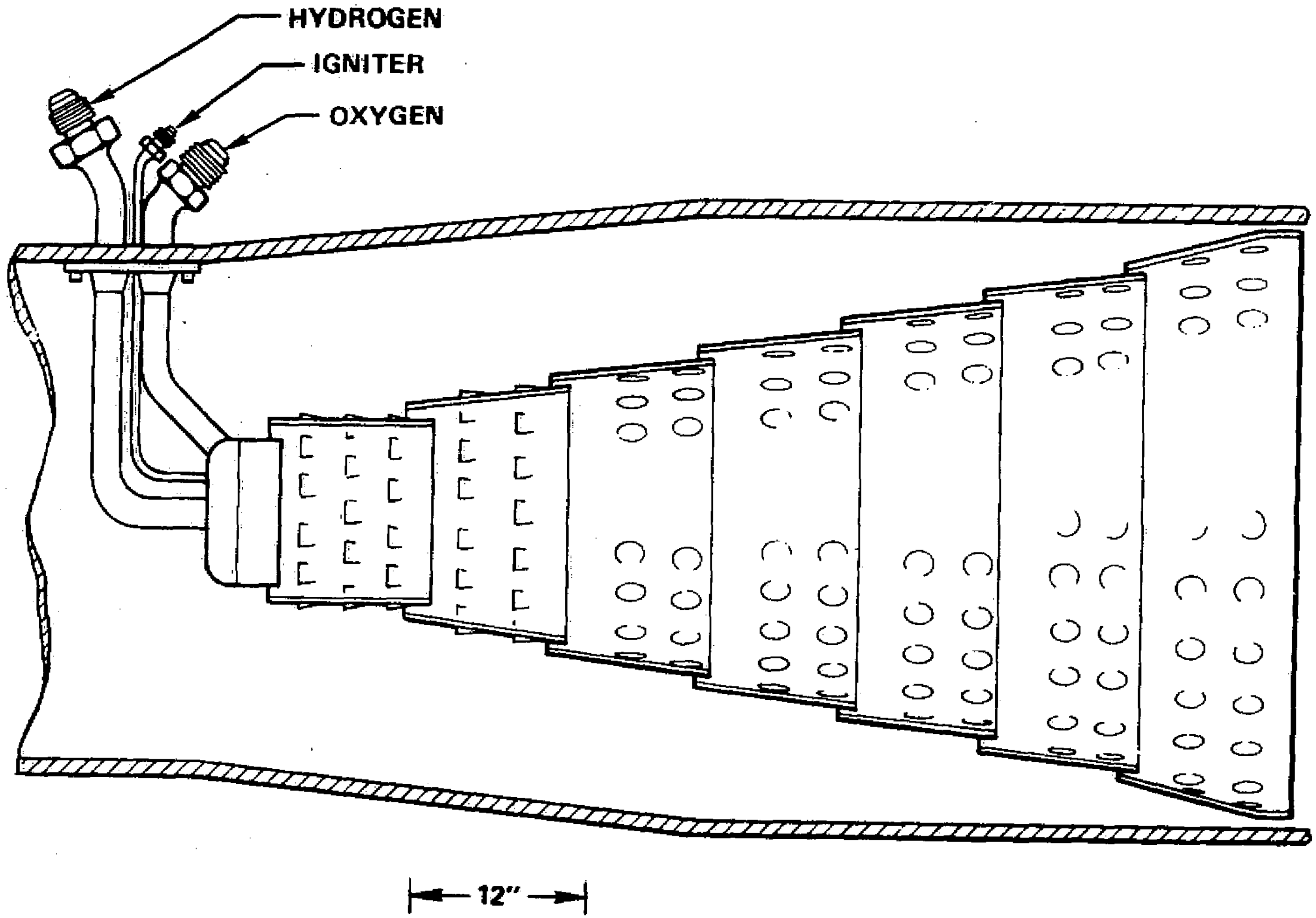

Figure 26. Hydrogen Oxygen Direct Combustion

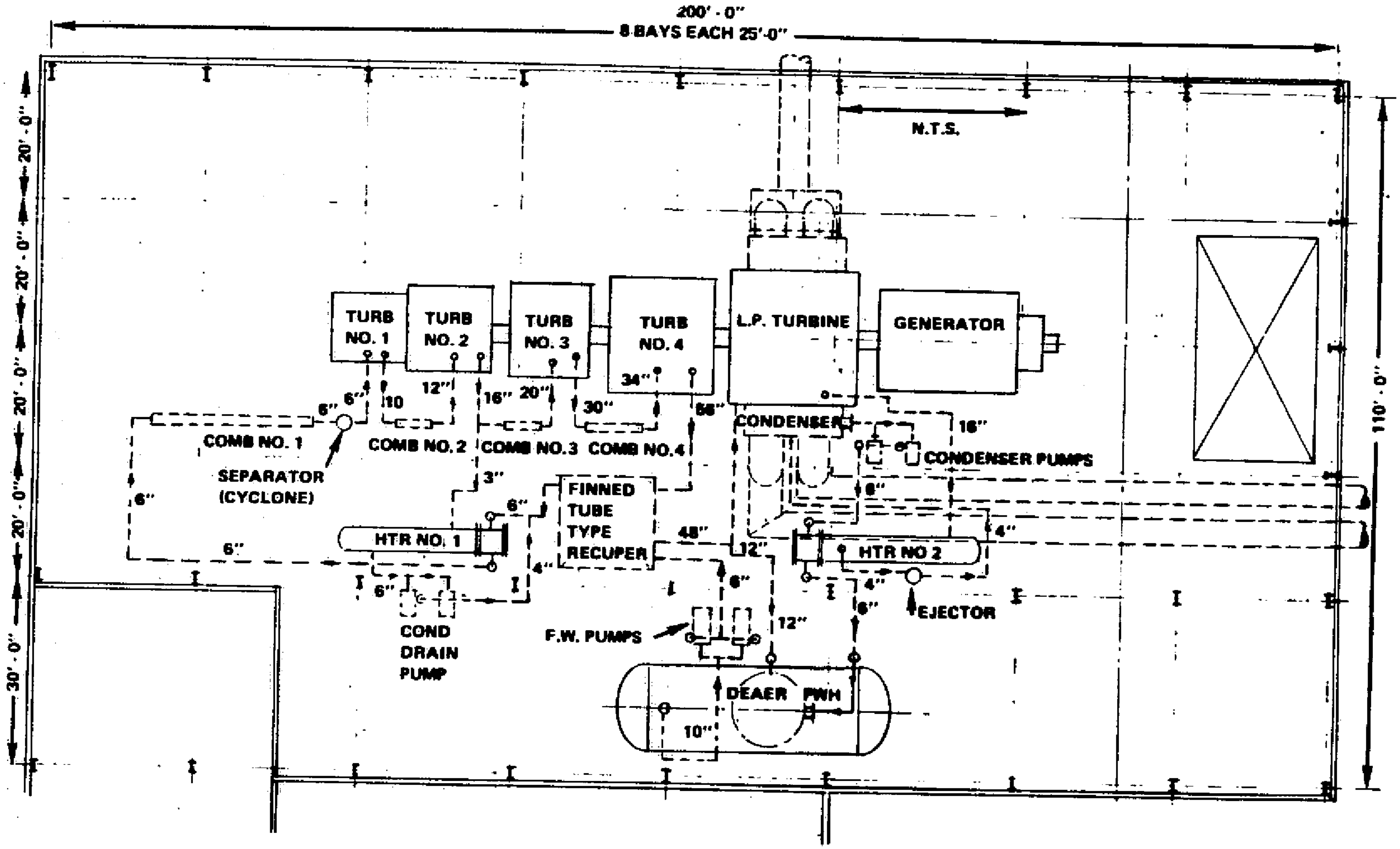

Figure 27. Multiple Reheat Ericsson Cycle

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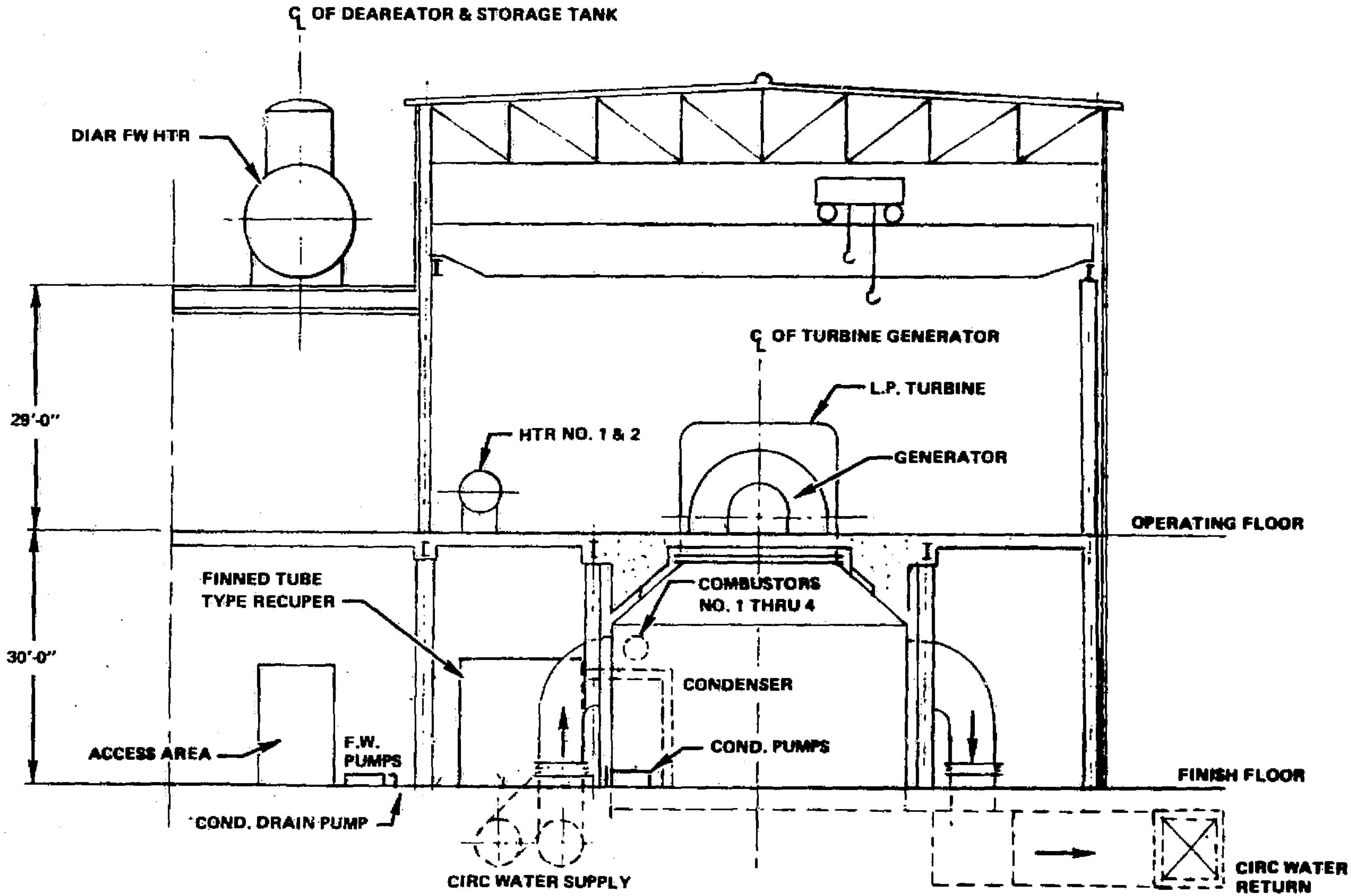

Figure 28. Multiple Reheat Ericsson Cycle

TABLE 21. 100 MWe ADVANCED STEAM CYCLE-ERICSSON DIFFERENTIAL CAPITAL COSTS VS CONVENTIONAL COAL

Ericsson

24.13×10^6 N/m^2/811 K/811 K/811 K/811 K
(3500 psi/1000 F/1000 F/1000 F/1000 F)

Differential Turbine Costs	$\$12.8 \times 10^6$
Differential Piping, Controls, etc.	$\$ 8.4 \times 10^6$
H_2 Combustors and Separator	$\$ 1.8 \times 10^6$
	$\$23.0 \times 10^6$
	\$230/kW

Conventional Coal Fired   Single Unit

12.41×10^6 N/m^2/811 K/811 K
(1800 psi/1000 F/1000 F)

Boiler Plant	\$335/kW
SO_2 Scrubber	\$120/kW
	\$355/kW

Comparison

	Ericsson	Conventional
"Boiler" Plant \$/kW-hr	100	455
Balance of Plant, \$/kW-hr	430	300
	530	755

for the coal-fired plant. For convenience in making the comparison of these two systems, it was considered that a portion of the existing coal plant would be usable for the Ericsson system, and this balance of plant cost could be used for both systems. The differential cost of the Ericsson system consists of the combustors, separators, and differential piping estimated at approximately \$100/kW-hr. The differential turbine cost was estimated at approximately \$130/kW-hr and was added to the coal balance of plant cost. The Ericsson cycle capital cost of \$530/kW-hr then consists of \$100/kW-hr for the boiler (i.e., H_2/O_2 combustors) and the \$300/kW-hr coal balance of plant cost added to the differential turbine costs of \$130/kW-hr.

Table 22 develops the COE based on the ground rules of Objectives and Criteria section. This table compares baseload COE for various H_2/O_2 and coal costs. As is evident, the COE for the coal-fired plant is less in all cases than for the Ericsson system. The driving factor, of course, is the cost of fuel.

TABLE 22. COST OF ELECTRICITY - ERICSSON - 100 MWe

BASE LOADED-7500 HOURS/YEAR

	Ericsson			Coal		
Installed Cost (including interest and escallation) 4-Year Period	72,000,000 ($530/kW)			103,000,000 ($755/kW)		
Cycle Efficiency		42%			33%	
Plant Work Force		37			57	
Fuel Cost, $\$/10^6$ Btu	4.50	6.00	7.50	.50	1.00	1.50
Yearly Fuel Cost, x 10^{-6}	26.78	35.71	44.63	3.88	7.75	11.63
Fixed Cost at 18% x 10^{-6}		12.95			18.50	
Labor at $25/hr		1.92			2.96	
Maintenance x 10^{-6}		0.64			1.90	
Total, $ x 10^{-6}	42.29	51.22	60.14	27.24	31.11	34.99
COE, mil/kW-hr	56.5	68.34	80.3	36.3	41.5	46.6

An estimate of the COE for several duty cycles was then made to examine the potential of the Ericsson system for peaking or intermediate load. The results are shown in Table 23. Although the COE for the Ericsson cycle is less than for coal up to about 2000 hr/yr use, this COE is considerably more than for the Supplementary Steam Generation Cycle.

Economically, therefore, the Ericsson cycle is not competitive primarily because of the high cost of fuel. The low capital cost of this system coupled with the high efficiencies attainable make this system more attractive for baseload application if H_2/O_2 costs were available at about $\$3.50/10^6$ Btu. Higher fuel costs (more than $\$3.50/10^6$ Btu) could still be competitive if higher (1366 K) (2000 F) temperature turbines become available.

TABLE 23. PARTIAL LOAD SERVICE

	Hours/Year	500	1000	2000	4000	7500
Ericsson Cycle	Fuel at 6.00/10^6 Btu, $\$ \times 10^{-6}$	2.38	4.75	9.5	19.1	35.71
	Labor $\$ \times 10^{-6}$	0.4	0.4	0.6	1.2	1.92
	Maintenance $\$ \times 10^{-6}$	0.3	0.3	0.3	0.4	0.64
	Fixed $\$ \times 10^{-6}$	12.95	12.95	12.95	12.95	12.95
	Total $\$ \times 10^{-6}$	16.03	18.40	23.35	33.65	51.22
	COE, mil/kW-hr	320	184	117	84.1	68.3
Conventional Coal Cycle	Fuel at 1.00 /10^6 Btu, $\$ \times 10^{-6}$	0.50	1.03	2.06	4.12	7.75
	Labor $\$ \times 10^{-6}$	0.7	0.8	1.2	2.0	2.96
	Maintenance $\$ \times 10^{-6}$	0.7	0.8	1.0	1.3	1.90
	Fixed $\$ \times 10^{-6}$	18.5	18.5	18.5	18.5	18.5
	Total $\$ \times 10^{-6}$	20.4	21.1	22.8	25.9	31.4
	COE, mil/kW-hr	408	211	114	64.8	41.5

HYDROGEN-FUELED GAS TURBINE

System Description and Technical Analysis

The use of H_2 fuel for a gas turbine appears feasible and attractive from the standpoint that H_2 is a clean-burning fuel with a wide flammability range. The effect of H_2 versus other fuels on the gas turbine heat rate (at a given gas inlet temperature) is small. The efficiency is slightly lower due to the high moisture content in the flue gas. The characteristics of H_2-fueled gas turbines should be similar to conventional gas-fired gas turbines. Figure 29 contains the four basic cycle arrangements examined. Other advanced systems are described later. The efficiency for the four cycles based on the HHV of H_2 (61,070 Btu/lb), as a function of compressor pressure ratio are shown in Fig. 30. The turbine inlet temperature is set at 1366 K (2000 F), which is expected to be attainable in the future with the ash-free H_2 fuel.

As seen in Fig. 30, the recuperated cycles both optimize at the low pressure ratios while the simple cycle optimizes at a higher pressure ratio. For the simple cycle, a pressure ratio below the optimum results in high stack loss. At high pressure ratio above the optimum, the additional compressor work is more than the turbine can provide. The recuperative cycles optimize at lower pressure ratios as a result of the unfavorable lower turbine exit temperature, which corresponds to the high expansion ratios, and the high air temperature due to the high adiabatic compression. The combined effects diminish the effectiveness of the recuperator. Neverhteless, the efficiency for the recuperative cycle at its optimum pressure ratio is 5 points higher than the simple cycle. With staged compression and intercooling, which minimizes both the flue loss and compression loss, the optimum efficiency is more than 2 points higher than the cycle with regeneration only. This arrangement is used in some recent closed-cycle gas turbine designs in Europe. In this country, the regenerative cycle has not been used except in very small scale. Major problems have been materials and cost. The combined cycle is more accepted in the United States.

Tables 24 and 25 show the results of the analysis performed on the Brayton cycle with regeneration and with and without intercooling.

A simple gas turbine with a 100 MWe output was analyzed. The results are shown in Table 26.

The high compressor pressure ratio and the turbine expansion ratio generally degrade the performance. The component efficiencies are indicated in the table. Figure 31 shows the effect of the component performance on the overall cycle efficiency. The turbine inlet temperature is maintained at 1366 K (2000 F).

Figure 32 is a plot of the cycle efficiency versus the turbine inlet temperature at the optimum pressure ratios. It shows, in general, that higher temperatures result in higher efficiency. For each temperature level, the combined cycle shows the highest efficiency.

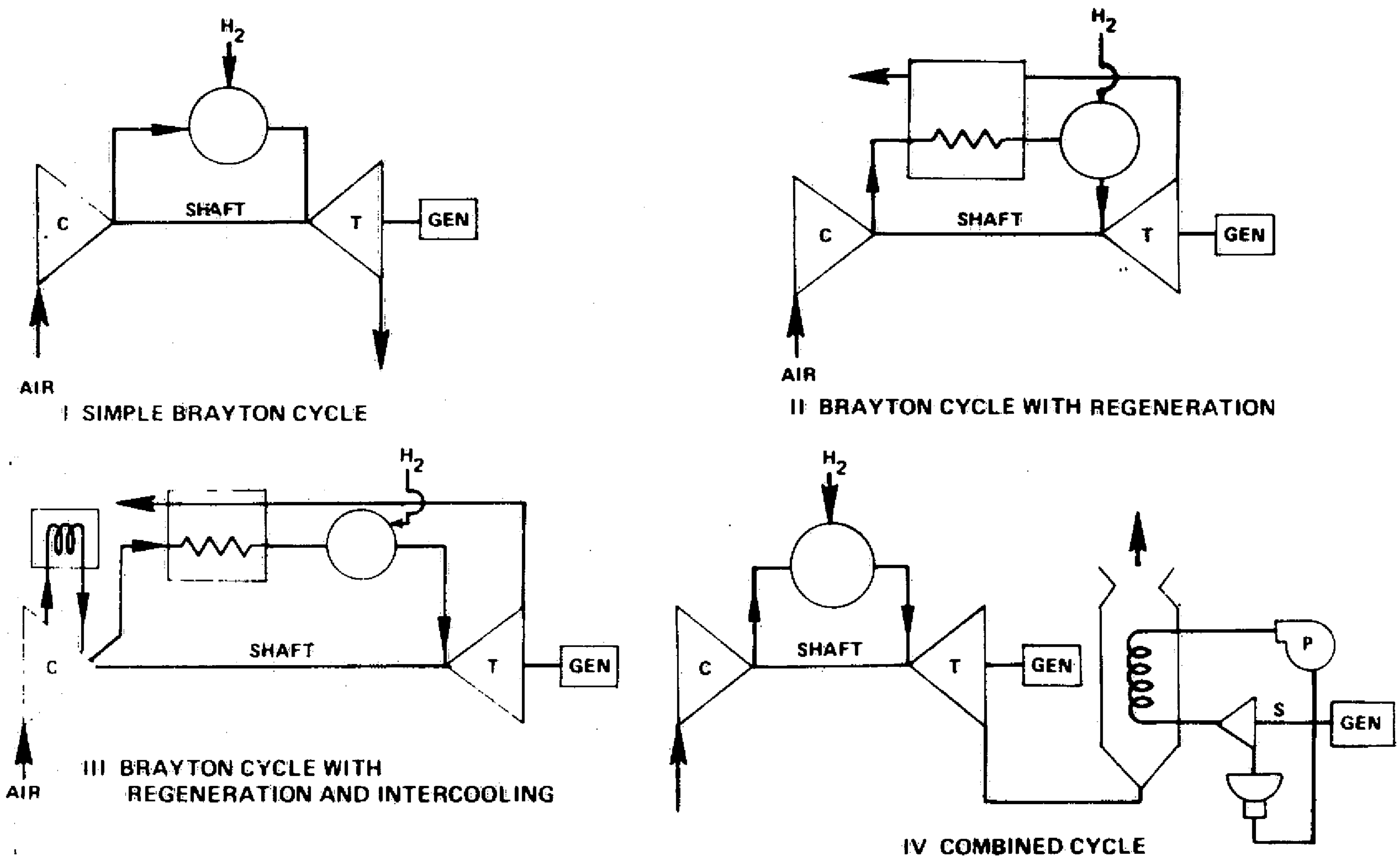

Figure 29. Gas Turbine Cycle Diagrams

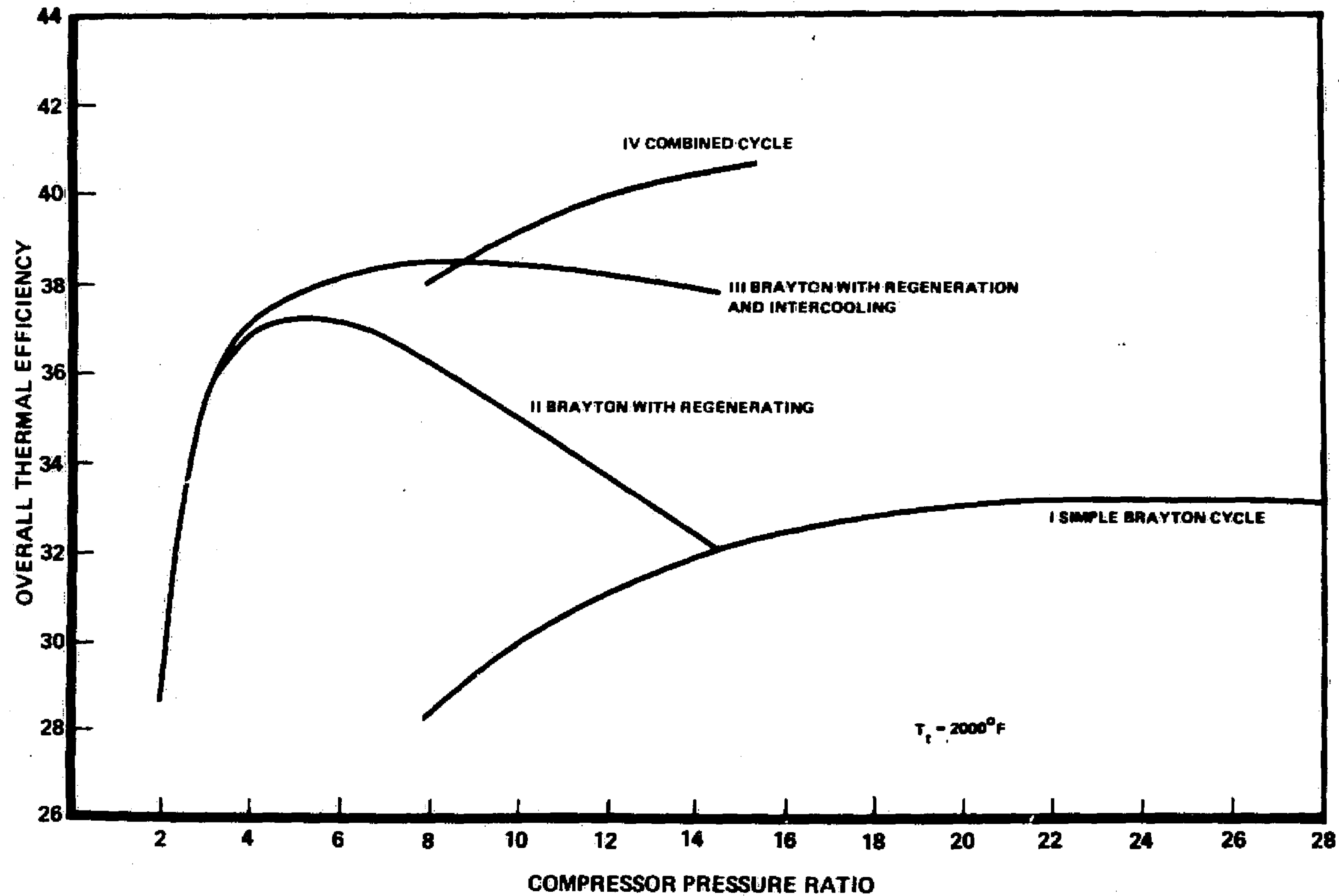

Figure 30. Compressor Pressure Ratio vs Thermal Efficiency

TABLE 24. BRAYTON CYCLE WITH REGENERATION

- Compression Pressure Ratio = 5
- Turbine Inlet Temperature = 1366 K (2000 F)
- Turbine Efficiency = 0.89
- Compression Efficiency = 0.87
- Regenerator Effectiveness = 0.85
  - Heat Rate = 9362 Btu/kW-hr
  - Efficiency = 36.46%
- Flowrate:
  - Air: 351 kg/s (2,787,300 lb/hr)
  - H_2: 1.93 kg/s (15,330 lb/hr)

TABLE 25. BRAYTON CYCLE WITH REGENERATION AND INTERCOOLING

- Two-Stage Compression with Intercooling
- Compression Pressure Ratio = 8 (-Efficiency = 0.87)
- Turbine Inlet Temperature = 1366 K(2000 F) (-Efficiency = 0.89)
  - Heat Rate = 9070 Btu/kW-hr
  - Efficiency = 37.63%
- Flowrate:
  - Air: 277 kg/s (2,200,000 lb/hr)
  - H_2: 1.87 kg/s (14,850 lb/hr)

TABLE 26. SIMPLE BRAYTON CYCLE

- Compression Pressure Ratio = 22
- Turbine Inlet Temperature = 1366 K (2000 F)
- Turbine Efficiency = 0.88
- Compressor Efficiency = 0.86
  - Heat Rate = 11,542 Btu/kW-hr
  - Efficiency = 29.57%
- Flowrate:
  - Air: 370 kg/s (2,938,000 lb/hr)
  - H_2: 2.38 kg/s (18,900 lb/hr)

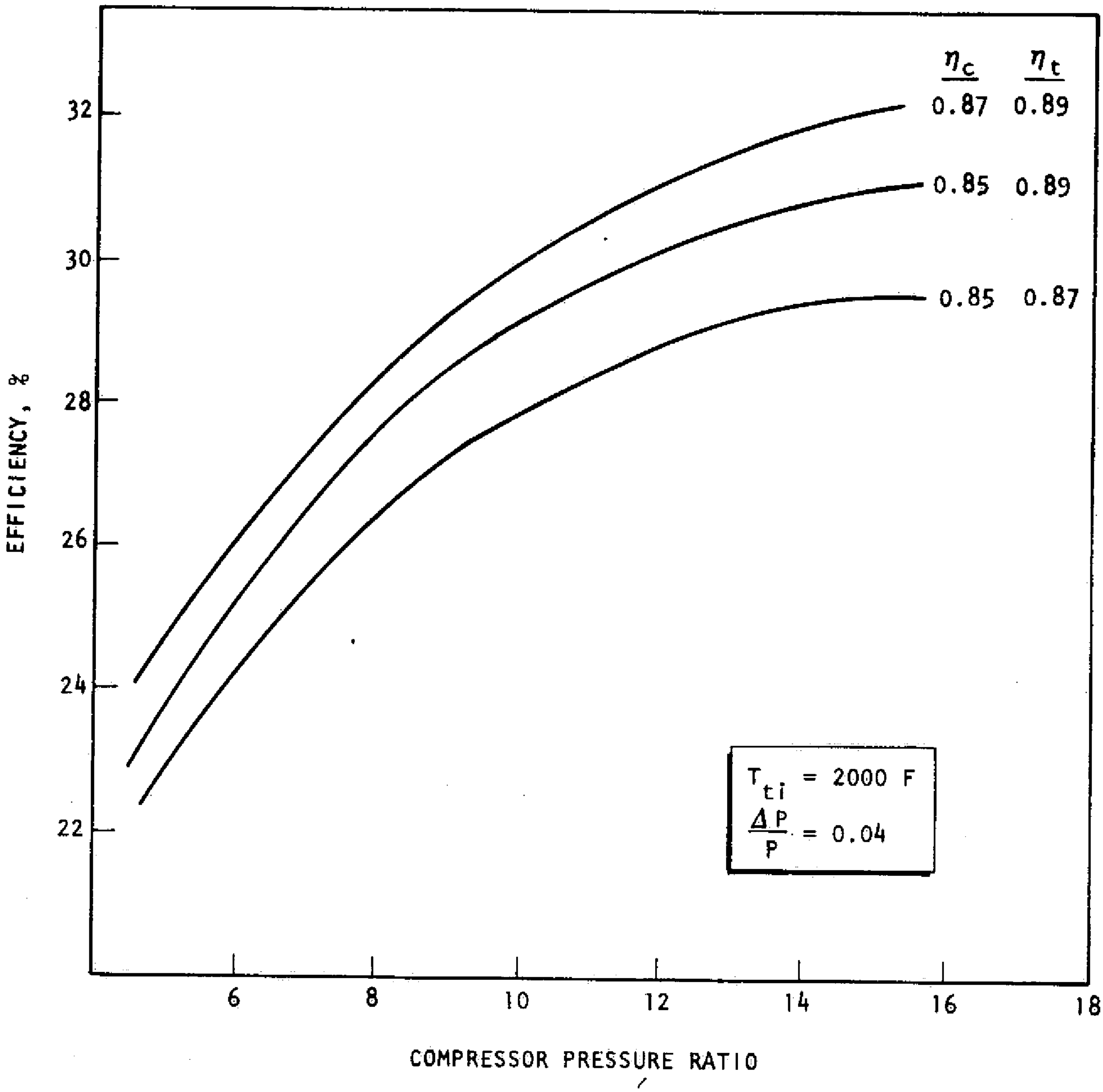

Figure 31. Effect of Turbine and Compressor Performance on Single Cycle Efficiency

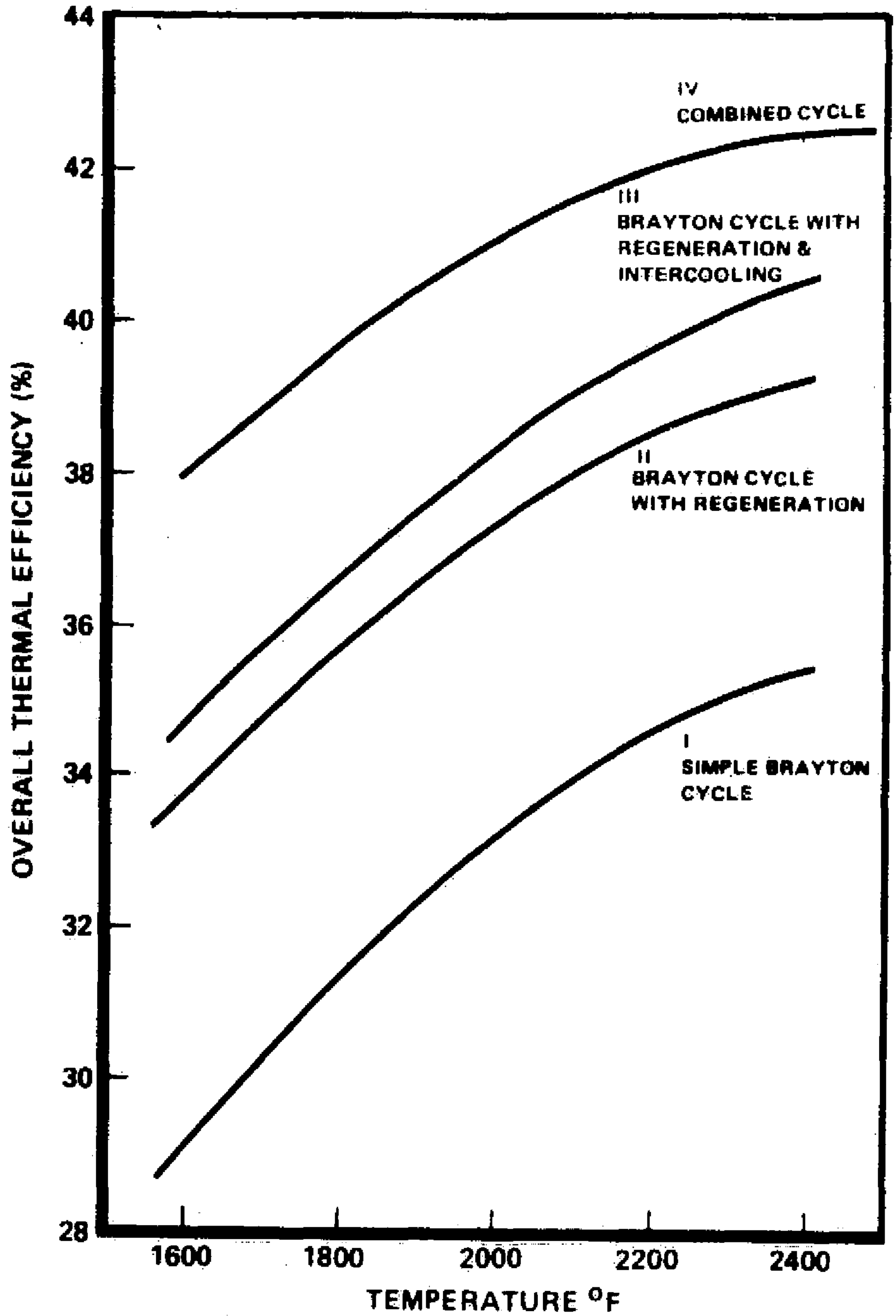

Figure 32. Temperature vs Thermal Efficiency at Optimum Pressure Ratio

Because of the high fuel costs, the low efficiency of the simple cycle produces a COE comparable to the more efficient combined cycle. The regenerated cycles produce a high COE due to the regenerator costs, and the efficiency is also below the combined system. Therefore, although the peaking capability of the combined cycle is reduced, this system was selected for further evaluation because of its efficient fuel utilization and competitive COE.

The combined cycle has been more readily accepted by utilities; however, the peaking capability is reduced because of the heatup time required of the steam system. Frequently, the bottoming cycle serves as the prime cycle with an exhaust-fired boiler. Gas turbine technology is improving and it is expected, expecially with clean fuels, that they can be designed with the turbine inlet temperature as high as 1589 K (2400 F). The cooling is accomplished by air bled from the compressor and precooled. The high temperature enables the use of a waste heat boiler. The overall result will be more efficient use of heat generated.

The combined cycle analyzed assumes a straight condensing bottoming steam cycle, which is estimated to have a heat rate of 10,705 Btu/kW-hr. The rating of the bottoming unit is approximately 30 to 50 MW. As the turbine exit temperature is a function of both the expansion ratio and the inlet temperature, the feedwater flowrate is throttled to provide a constant initial steam condition at 755 K (900 F) and 5.86×10^6 N/m^2 (850 psig). The flue gas temperature is computed to reflect a minimum of 272 K (30 F) pinch temperature between the hot gas and the feedwater steam mixture (Fig. 33). The contribution from the bottoming cycle is thus the energy available times the heat rate. The results from this analysis are summarized in Table 27. It should be noted that above a pressure ratio of 12 in Fig. 30, the turbine exit temperature is less than 283 K (50 F) above the superheated steam temperature. A superheater under this condition may be difficult to design. Note that high turbine temperatures are required to achieve the high cycle efficiencies. Reducing the turbine inlet temperature to the present state-of-the-art technology of 1200 K (1700 F) reduces the efficiency to about 36.8%. For the present, the 1200 K (1700 F) system is more realistic.

TABLE 27. COMBINED CYCLE

COMBINED CYCLE

Brayton Cycle

- Compression Pressure Ratio = 12
- Turbine Inlet Temperature = 1478 K (2200 F)
- Turbine Exit Temperature = 829 K (1032 F)
- Network Outlet = 75 MW
- Compression Efficiency = 0.87
- Turbine Efficiency = 0.89
    - Overall Heat Rate = 8522 Btu/kW-hr
    - Efficiency = 40.00%
- Flowrate:
    - Air 191 kg/s (1,519,140 lb/hr)
    - H_2 1.76 kg/s (13,954 lb/hr)
    - H_2O 23.1 kg/s (183,634 lb/hr)

Rankine Cycle

- Steam Pressure = 5.86×10^6 N/m^2 (850 psig)
- Steam Initial Temperature = 755 K (900 F)
- Stack Exit Temperature = 444 K (340 F)
- Network Output = 25 MW
- Heat Rate = 10,705 Btu/kW-hr

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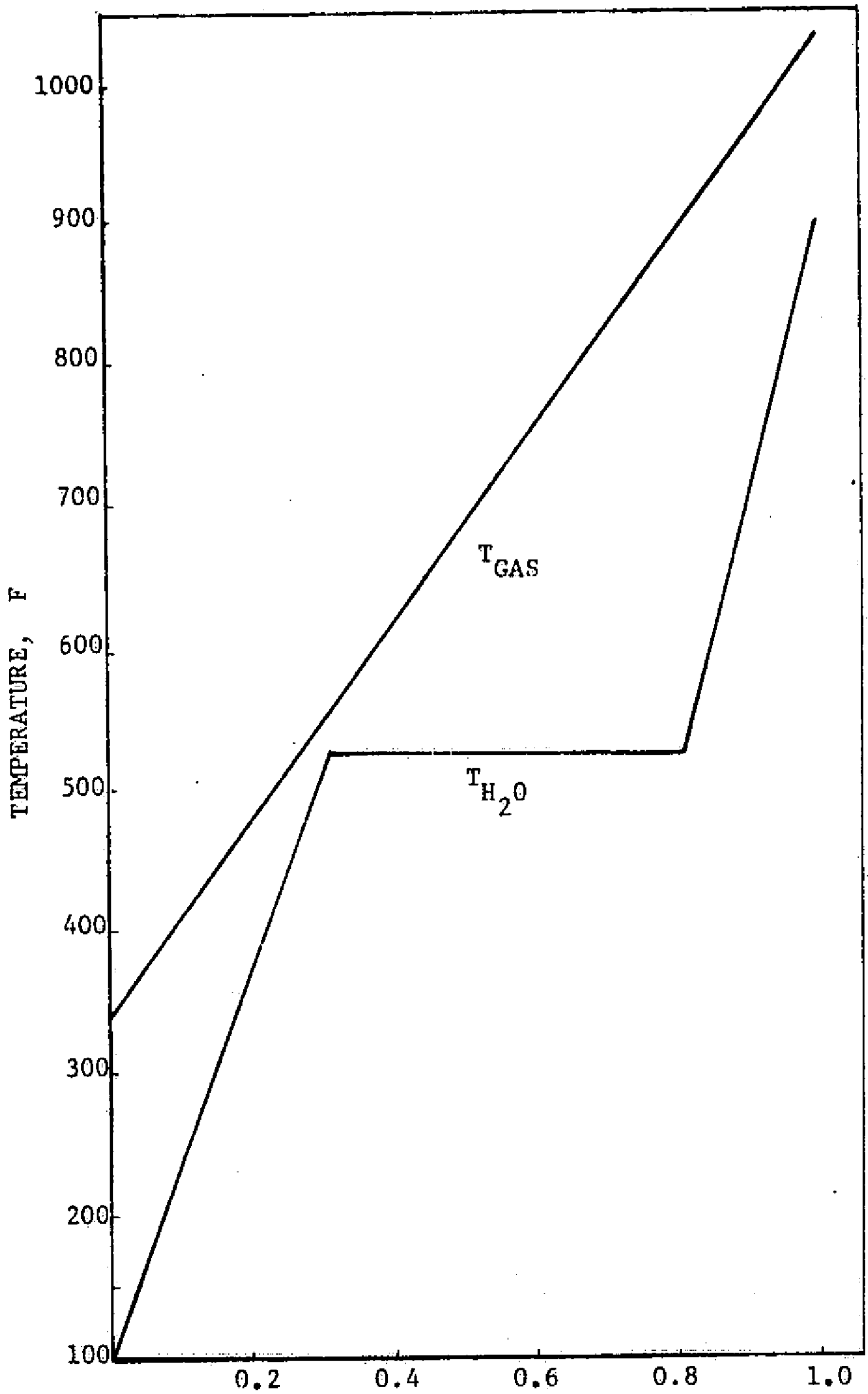

Figure 33. Linearized Boiler Temperature Profile

Advanced cycles were also investigated, including a pressurized "semi-closed cycle" with direct combustion heating of the working fluid (air). In this cycle (Fig. 34), the primary mass flow through the cycle was N_2 which resulted from combustion of air and H_2, with subsequent condensing of the water from the combustion products. Sufficient "makeup" air is compressed and added to the cycle for stoichiometric combustion with the fuel. Some N_2 is bled off from the cycle to keep the nominal mass flowrate through the cycle constant.

A truly closed-cycle gas turbine, with direct combustion heating, can be accomplished if both O_2 and H_2 are available as economically viable reactants. In this cycle, the combustion chamber is heated by the stoichiometric combustion of O_2 and H_2 directly in the stream of nonreactive working fluid. Cooling of the working fluid prior to compression results in condensation of the water produced by the combustion, returning only the working fluid to the cycle. Expensive noble gas working fluids may not be practicable because of eventual contamination from impurities in the O_2 and the H_2. Typical operating conditions for this type of cycle are shown in Fig. 35.

These cycles were judged to be too great a departure from current practice without a significantly large improvement in thermal efficiency.

System Design

The combustor "can" of an existing gas turbine assembly could be modified for H_2 fuel with relative ease. Virtually all of the design features of a liquid fuel combustor would actually constitute an overdesign when compared to the requirements of GH_2. A natural gas combustor configuration could be modified by merely reorificing injection systems to account for the density differences between the fuels. A H_2 burner designed specifically for the purpose would be significantly shorter than an equivalent design for liquid fuel. The mixing and dilution section of the burner would remain essentially the same, but the actual combustion section length could easily be reduced to half.

An optimized H_2 burner system could also be developed, with a more elaborate development program, to exploit the characteristics of H_2 for further emission control improvement. Unburned hydrocarbons and carbon monoxide emissions would be essentially totally eliminated by just the fuel substitution. Some hydrocarbon content would probably be present in so called "dirty" H_2 from coal conversion, but carbon compounds from these impurities would be expected to stay at very low levels.

Oxides of N_2 would remain in evidence with H_2 fuel, both from the combustion temperature involved and, to a lesser degree, from the impurities in coal-derived H_2. However, the wide flammability limits of H_2 would permit special lean burning combustor configurations, which would limit the maximum temperature even within the local combustion zone in the burner. This configuration could take the form of a premix section where the lean limit mixture is well mixed prior to combustion in a flame holder section where a well-mixed turbulent flame would be produced similar to the action of a bunsen or meeker burner. The temperature in this flame front would still undoubtedly be greater than allowable for turbine inlet, so a conventional dilution and mixing section would be utilized upstream of the turbine nozzle inlet section.

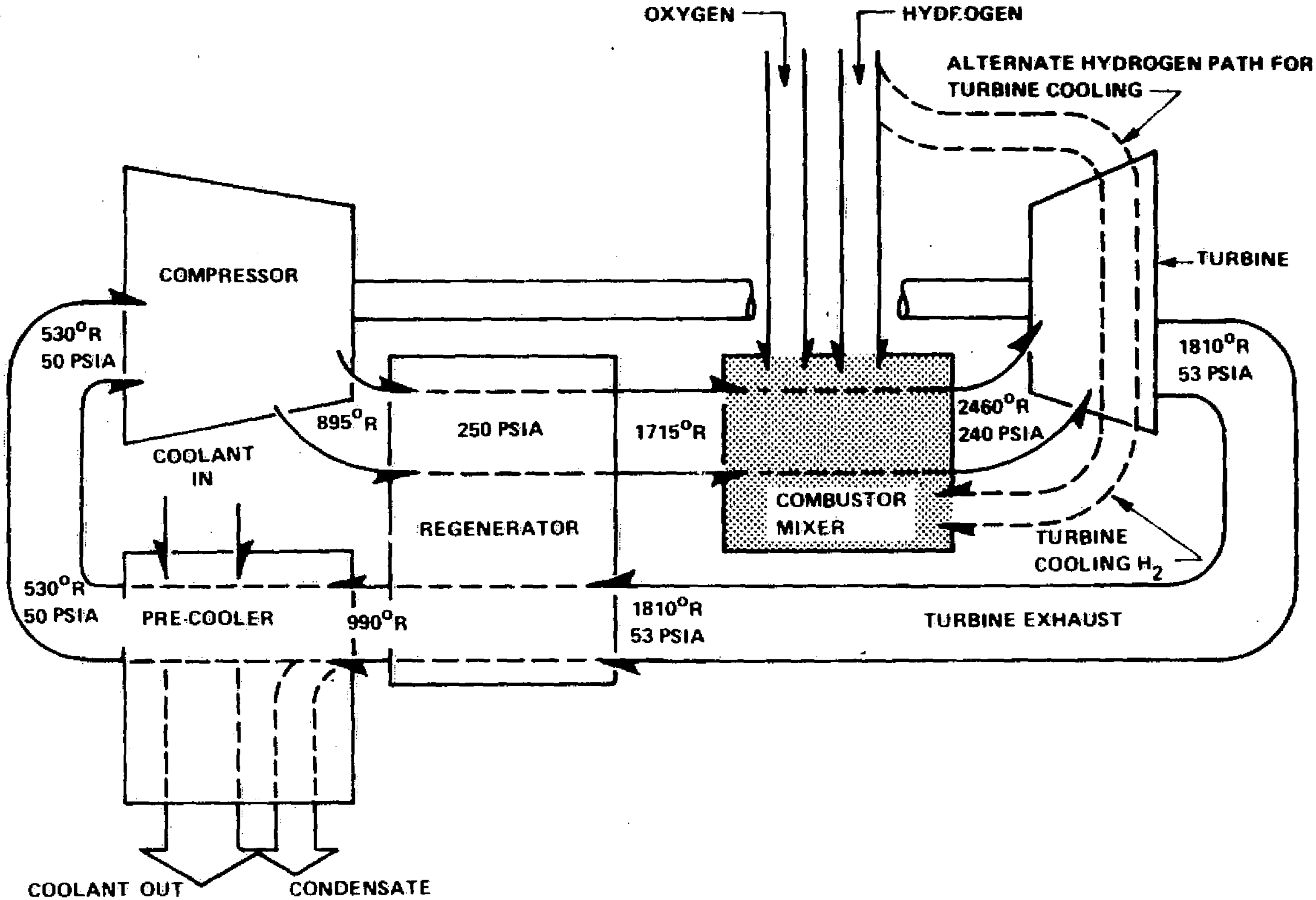

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Figure 34. Closed-Cycle Gas Turbine With Direct O_2/H_2 Combustion Heating

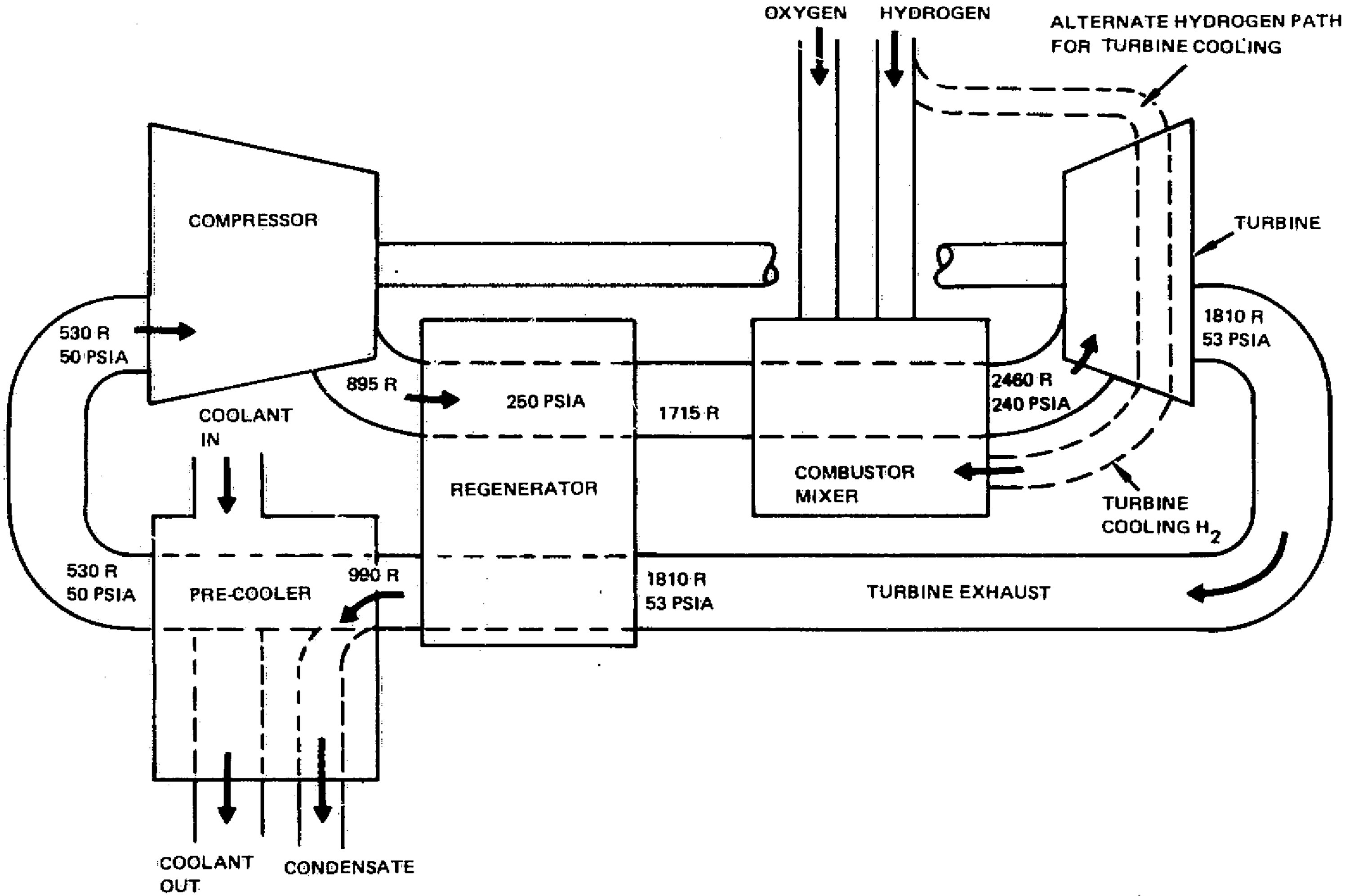

Figure 35. Closed-Cycle Gas Turbine With Direct O_2/H_2 Combustion Heating (Typical Working Fluid Temperatures)

The excellent cooling capability of H_2 lends itself to consideration for turbine cooling methods that would maintain high thermal efficiency by regeneration of the coolant flow, as well as permitting the improved cycle performance with increased turbine inlet temperature. The regenerative concept would encompass a closed cooling path through the turbine blades and the turbine nozzle. Each turbine blade would have a coolant path and a return path so that the heated H_2 could be recollected and ducted to the burner. The enthalpy of the heated H_2 would be returned to the cycle in the combustion chamber. A prototype of such a H_2-cooled turbine stage has been demonstrated in a component test rig by the General Electric Company (Ref. 4). With this cooling arrangement, turbine inlet temperatures as high as 2478 K (4000 F) have been demonstrated for durations of up to 1 hour.

High turbine inlet temperatures improve cycle efficiency, but increase the problems of nitrogen oxides. Lean burning with premixed fuel and air would probably reduce these emissions to an absolute minimum. The other noxious emissions normally associated with fossil fuel combustion will not be a problem; carbon, hydrocarbon, and sulpher compounds should not be present in measurable amounts unless the H_2 being used has a high level of these compounds as impurities.

The overall conclusion of this study indicates that H_2 and H_2/O_2 combustion provides high potential for application in several variations of the Brayton cycle. Development risks are relatively low with a high probability of success. The main requrement is for high-temperature turbines. Hydrogen is an excellent fuel for numerous variations of the Brayton cycle. Existing turbine systems could be converted to H_2 fuel with very little development effort, and few physical modifications. Advanced combustor design to exploit the characteristics of H_2 fuel could likewise be accomplished with relatively little development effort, and could provide significant improvement in emission control. Advanced cycles with regeneration, or "bottoming" supplementary cycles would also benefit from the clean burning and low pollution characteristics of H_2 combustion. Closed, and semiclosed cycles can be devised to operate with direct combustion heating, eliminating heat exchanger problems, because of the unique properties of H_2 as a fuel.

Gas Turbine Economics

The economic assessment performed for the gas turbine was based on the ground rules identified in the Objectives and Criteria section. The system costs were based on a 1200 K (1700 F) turbine inlet temperature rather than for the higher-temperature turbines, because this was judged to be midterm (i.e., to the year 2000) capability. Choosing the 1200 K (1700 F) temperature has the effect of reducing the heat rate and thus the efficiency to 9280 Btu and 36.8%, respectively.

The capital costs were determined as shown in Table 28. This cost was then factored by 18% to determine the fixed costs per year. Fuel costs were determined using $\$4.50/10^6$ Btu for hydrogen alone. Labor and maintenance costs were estimated depending on the amount of usage of the system. Table 29 summarizes the costs and shows a COE for various hours/year usage. This COE is comparable to the supplementary steam application through 3000 hr/yr (peaking and intermediate) at which time COE is slightly less than supplementary steam. However,

C-2

TABLE 28. COMBINED CYCLE COST

Structure and Improvement	600,000
Mechanical	
Gas Turbine (75 MW/1 Unit)	11,800,000
Steam Turbine	
+Pump + Boiler (25 MW/1 Unit) +Deaerator + Condenser	3,280,000
Piping, Controls and Instrumentation	2,000,000
Station Electrical	950,000
	$18,630,000
	($186/kW)

TABLE 29. OIL FIRED COMBINED CYCLE GAS TURBINE

- 100 MWe
- H_2 Fuel at $4.50/10^6 Btu
- Capital Cost: $186/kW, installed (including interest)
- 1200 K (1700 F) Turbine Inlet Temperature, 9280 Btu/kW-hr heat rate (combined)

Hours/Year	500	1000	2000	4000	7500
Fuel at $4.50/10^6 Btu, $ x 10^{-6}	2.09	4.18	8.4	16.7	31.3
Labor $ x 10^{-6}	0.5	0.7	1.0	1.3	1.5
Maintenance $ x 10^{-6}	0.3	0.5	0.8	1.2	1.5
Fixed Cost at 18%, $ x 10^{-6} ($186/kW)	3.3	3.3	3.3	3.3	3.3
Total $ x 10^{-6}	6.19	8.68	13.5	22.5	37.6
COE, Mil/kW-hr	124	86.8	67.5	56.3	50.1

combined cycle gas turbine COE is greater than a coal plant after approximately 3000 hr/yr intermediate and baseload as shown in Fig. 36.

On the basis of COE alone, the combined cycle gas turbine competes reasonably well with the supplementary steam generator through peaking applications, and somewhat more favorably for short intermediate loads, and does not compete with coal plants for long intermediate and baseload.

When one considers that the combined cycle loses its peaking capability because the steam components require signficiant heatup time, the supplementary steam generation appears more attractive.

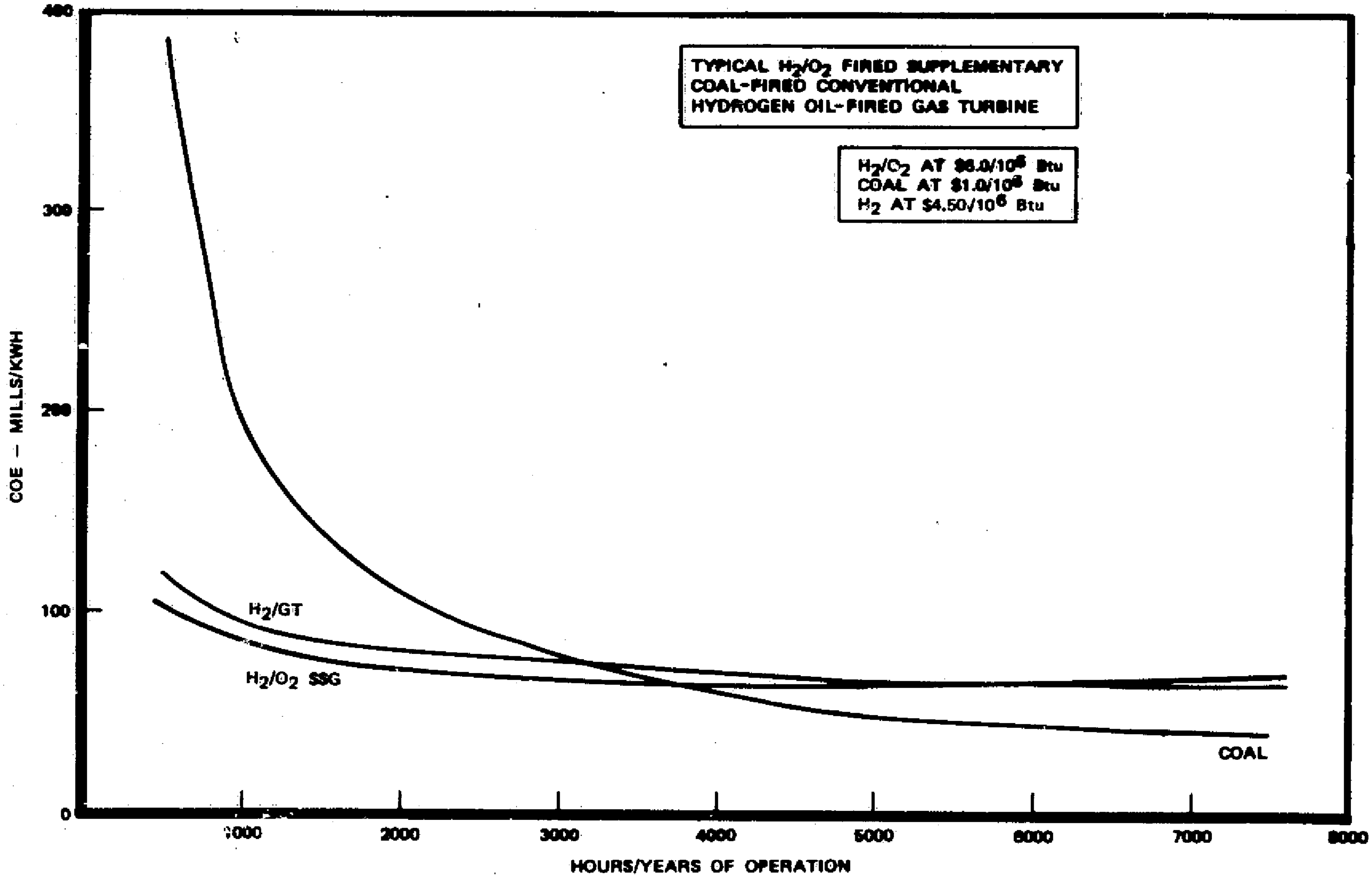

Figure 36. COE Comparison

PRESSURIZED WATER REACTOR - SUPERHEAT

System Description

A potential application of the H_2/O_2 combustor is the steam superheat and reheating in the steam power conversion cycles of pressurized water-cooled nuclear reactors. This concept is similar to the supplementary steam generation concept described in the Supplementary Steam Generation Section. The potential benefit from the steam superheating and reheating includes the improved cycle thermal efficiency caused by increased steam temperatures and the improved steam turbine performance and less erosion wear of the turbine buckets due to reduced moisture entrainment in the steam. A specific example of such an application is the use of a steam superheater, fired by fossil fuel (oil), at the Consolidated Edison Company's Indian Point pressurized-water-cooled reactor (PWR) steam powerplant. The Indian Point Station (2% mw capacity) is one of the earlier nuclear powerplants in commercial operation in this country (since 1962) and is the only plant that has employed a fossil-fuel-fired superheater. The main reasons for its not being widely incorporated in later-constructed nuclear powerplants have been: low nuclear fuel cost (until recently) to justify the use of a separate fossil-fuel-fired superheater; and development of special designed nuclear steam turbines to handle highly moisture-laden steam and the higher primary coolant pressure to give higer steam pressure and temperature 6.89×10^6 N/m^2/572 K as compared to 2.55×10^6 N/m^2/505 K (1000 psi/570 F as compared to 370 psi/450 F). In addition some operational and environmental problems may occur with this concept. These problems are discussed in the Environmental and Operational Constraints section.

There are about 130 light water nuclear steam powerplants (both BWR and PWR) in commercial operation or under construction in this country. Reference presents a list of these plants and their operating conditions and capacities. Most of the later units have capacities from about 800 to 1300 mw and operate at a steam pressure from about 6.20×10^6 to 7.24×10^6 N/m^2 (900 to 1050 psig) and at steam temperatures from saturation temperature to about 283 K (50 F) superheat, ranging from 544 to 572 K (520 to 570 F). In boiling-water reactors (BWR), steam to the turbine is generated directly in the primary cooling water circuit through the reactor core. Because of the large quantity of primary water circulation (15 to 20 times the steam flow) required for safety reasons, the steam flashed off is close to saturation. In pressurized-water reactors, the steam is produced in an indirect steam generator with the primary recirculating water temperature drop of about 286 to 289 K (55 to 60 F) and the steam temperature from saturation to 283 K (50 F) superheat. Though a small amount of superheat has an insignificant affect on the turbine performance, it is desirable from the standpoint of turbine control and partial load operations order that the inlet steam condition would not cross the saturation line. The primary water circuit pressure in PWR ranges from 12.41×10^6 N/m^2 to 15.5×10^6 N/m^2 (1800 to 2250 psig) which is about twice as high as that in BWR.

Technical Analysis

Indian Point PWR Station. The application of a fossil-fuel-fired (or oil-fired) superheater in the Indian Point PWR power station was discussed in Ref. 5 and 6. The reactor operates in a pressurized water loop at a pressure of 10.3 x 10^6 N/m^2 (1500 psia) and a total circulating flow of 8.1 m^3/s (128,000 gpm) at 525 K (486 F) inlet and 543 K (518 F) exit temperature. Steam leaves the secondary side of the vertical U-tube type steam boiler at a pressure of 2.90 x 10^6 N/m^2 (420 psig), a temperature of 505 K (450 F), and at a flow rate of 277 kg/s (2.2 x 10^6 lb/hr) and is superheated in a separately fired superheater to a steam temperature of 811 K (1000 F). Figure 37 shows a schematic diagram of the Indian Point PWR steam powerplant with the fossil-fuel-fired superheater and economizer. The plant arrangement and the operating data were taken from the Nucleonic's Reactor Field. A minor difference was noted later in the location of the economizer from the arrangement of Ref. 5. Because the heat load in the economizer was small, no attempt was made to revise the cycle analysis. Table 30 gives the heat balance of the Indian Point Plant with the oil-fired superheater. The computed thermal efficiencies are based on the sum total of the heat input from the nuclear reactor and the HHV of the oil consumed in the fired superheater with a furnace efficiency of 85%.

TABLE 30. HEAT BALANCE WITH FOSSIL-FUEL-FIRED SUPERHEATER

Heat Input From Nuclear Reactor	1975.094(10)^6 Btu		
Heat Input From Superheater	702.680(10)^6	Oil HHV Input (85%) = 827.847(10)^6	
Heat Input From Economizer	0.99(10)^6		
Feedwater Pump Energy Input	2.650(10)^6		
Condenser Loss			1729.918(10)^6
Power Output*			951.486(10)^6
	2681.404(10)^6		2681.404(10)^6
Combined (HHV) Gross Thermal Efficiency	= 33.95%		
Combined (HHV) Net Thermal Efficiency	= 33.85%		

*High-Pressure Turbine 308.396(10)^6
Low-Pressure Turbine 643.090(10)^6

Figure 38 and Table 31 show the cycle arrangement and heat balance of the Indian Point Plant using H_2/O_2 direct combustion as a superheater. Comparison with the results obtained for the case of oil-fired superheater indicates an improvement of about 1.3 points using the H_2/O_2 direct superheater. This improvement stems mainly from the fact that there is no furnace loss with the H_2/O_2 direct superheater as compared to 15% loss in the oil-fired superheater.

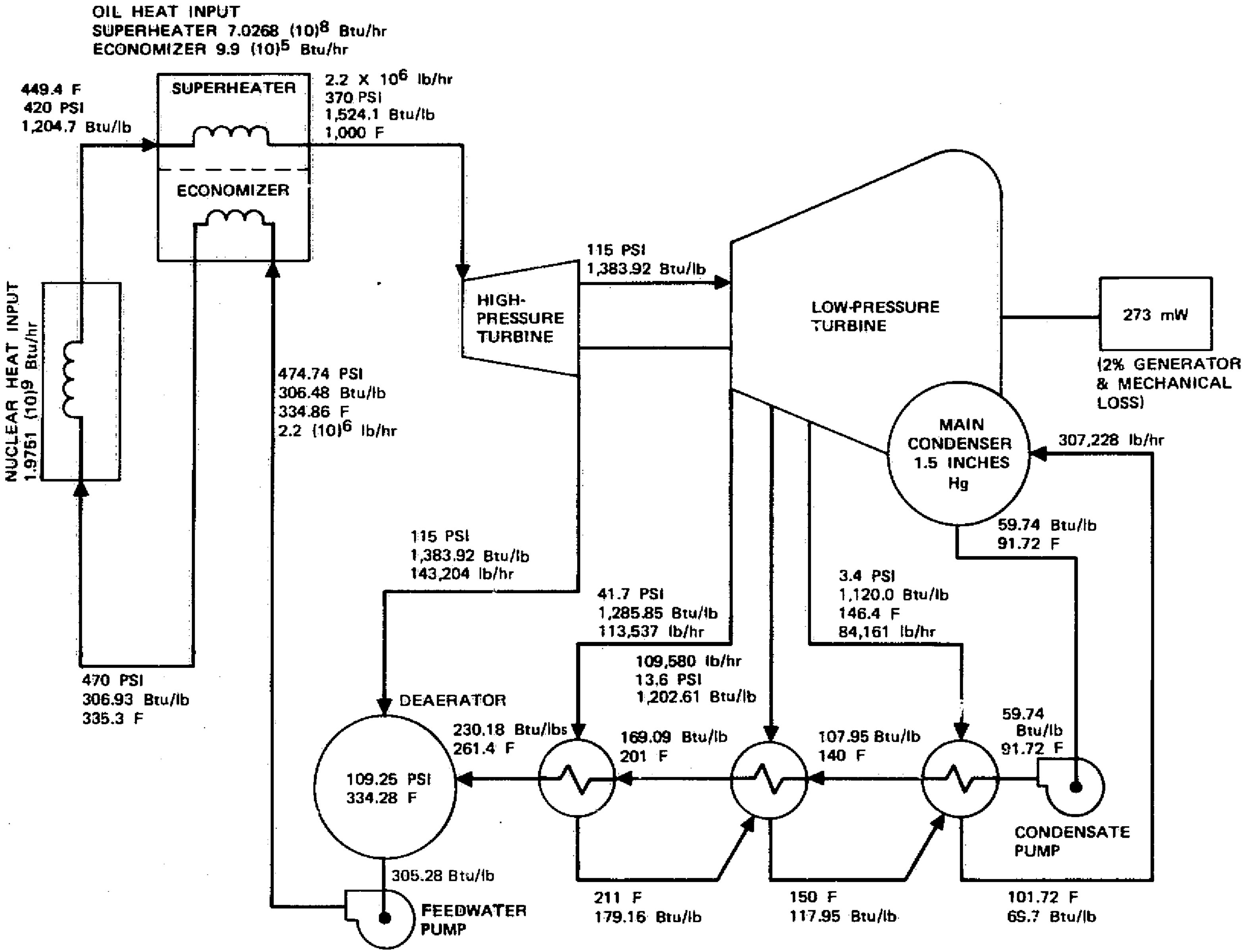

Figure 37. Indian Point Btu/lb PWR Steam Powerplant With Fossil-Fuel-Fired Superheater

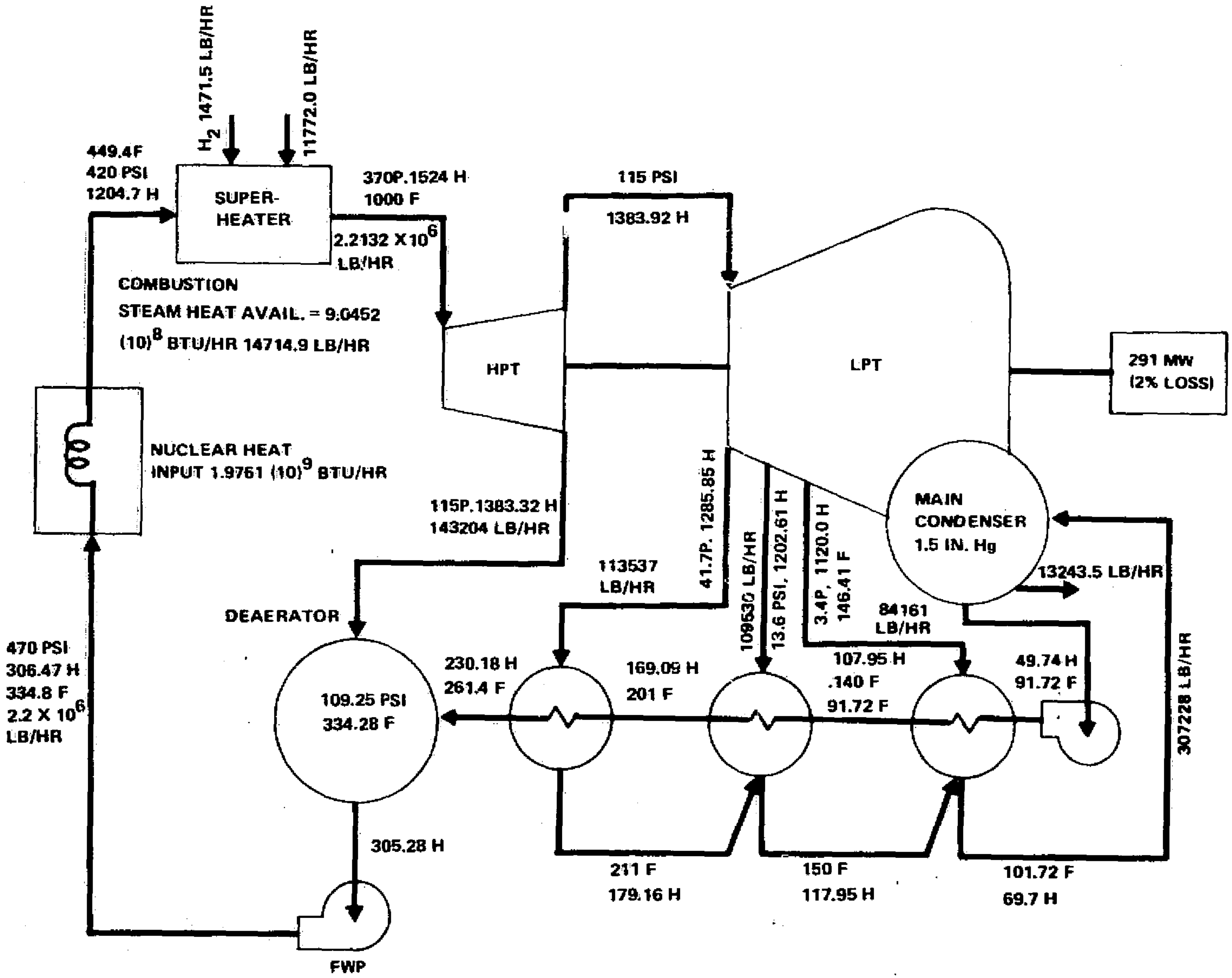

Figure 38. PWR Steam Powerplant

Some of this gain is offset by the higher latent heat loss, which is inherent to the H_2 fuel, and the less effective feedwater regeneration, because of mass injection (combustion steam) at the combustor and mass rejection (throwaway water) at the condenser outlet. It must be pointed out that the comparison being made here is really between performances of the oil-fired superheater and the H_2/O_2 direct superheater. If the comparison were between the PWR steam power cycles with and without a superheater, a larger difference in efficiencies would result. The most significant advantage of using the H_2/O_2 direct superheater lies, however, in the capital cost o the combustor, which is at least an order of magnitude lower than that of an oil-fired superheater.

TABLE 31. HEAT BALANCE WITH H_2/O_2 DIRECT SUPERHEATER INDIAN POINT PWR

Heat Input From Nuclear Reactor	$1976.106(10)^6$ Btu	
Heat Input (HHV) to Superheater	$904.524(10)^6$	
Feedwater Pump Energy Input	$2.618(10)^6$	
Condenser Loss (Includes Water Throwaway Loss)		$1868.547(10)^6$
Power Output*		$1014.702(10)^6$
	$2883.248(10)^6$	$2883.249(10)^6$

Combined (HHV) Gross Thermal Efficiency = 35.22%

Combined (HHV) Net Thermal Efficiency = 35.13%

*High-Pressure Turbine $326.961(10)^6$
Low-Pressure Turbine $687.741(10)^6$

David Besse Unit I PWR. The above case was analyzed with the assumption that the turbine adiabatic efficiencies are the same with or without the steam superheating. In practice, however, the turbine efficiency increases with decrease in moisture entrainment in the steam. Hence, superheating and reheating in the light water reactor steam cycles would improve not only the cycle efficiency with higher steam temperatures but also the turbine efficiencies because of lower moisture entrainment in the steam flow. Thus a more detailed steam turbine cycle analysis was performed on a typical recently completed PWR plant employing the performance prediction procedure given in Ref. 7, which takes into account all the factors affecting the turbine efficiency. The selected plant, David Besse Unit I station of the Toledo Edison Company, has a net capacity of 925 MW operating with throttle steam conditions of 6.10×10^6 N/m^2 (885 psia) and 583 K (590 F). The exhaust steam from the high-pressure turbine passes through a moisture separator and an indirect reheater and is heated in two stages by condensing the highpressure extraction steam and the throttle steam before it enters the low-pressure turbines, as shown in Fig. 39.

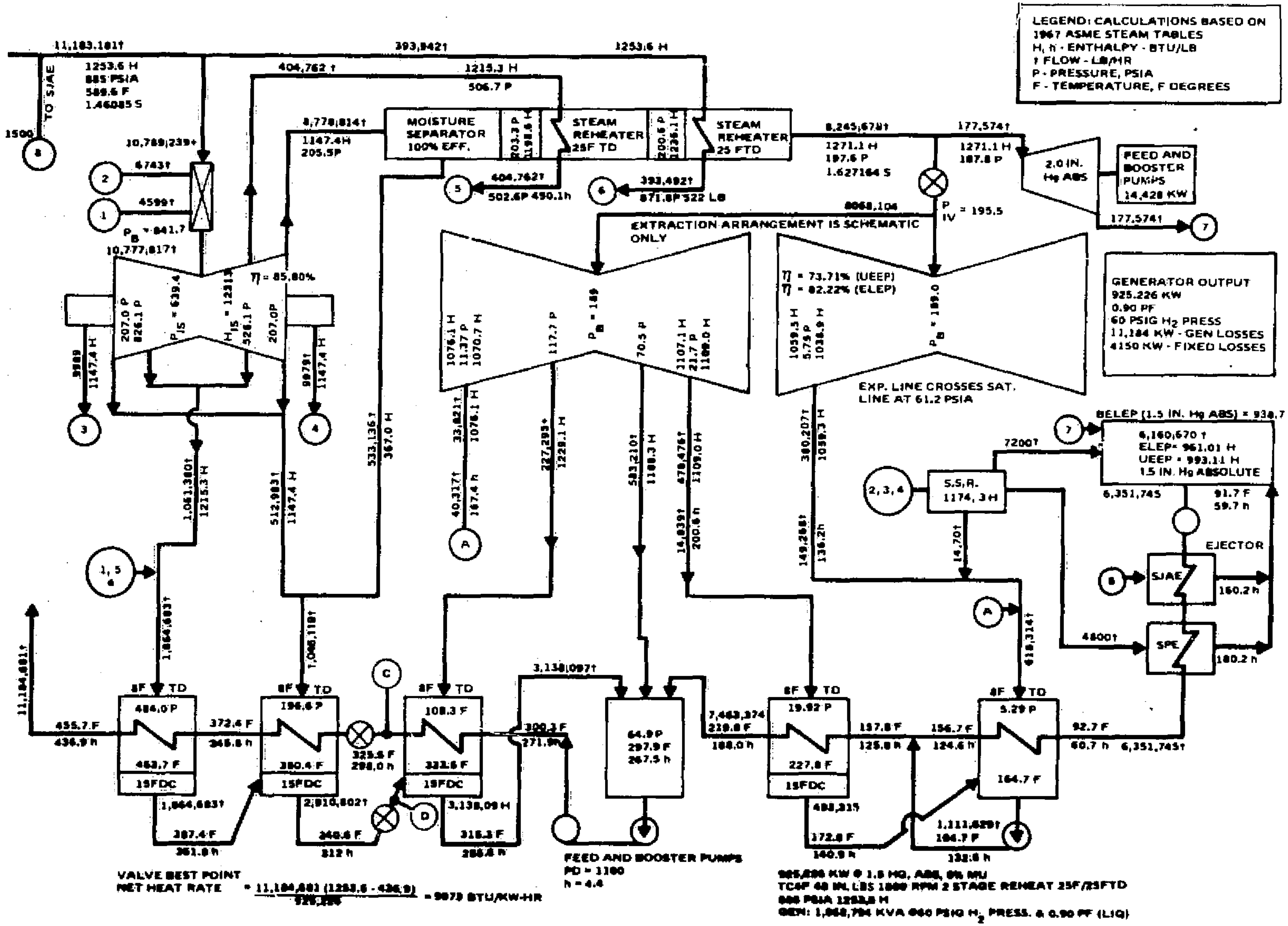

Figure 39. Mass and Heat Balance for Conventional PWR Steam Turbine

Figure 40 shows the corresponding H-S diagram. The low-pressure turbines are provided with moisture separating shrouds and cavities, and, at various stages, the moisture is extracted, mixed with some bleed steam and drained into the feedwater heaters. The zigzag lines in the H-S diagram indicate the changes in steam conditions after moisture separation in low-pressure turbines. • Table 32 gives the heat balance for the David Besse Unit I station.

TABLE 32. HEAT BALANCE FOR CONVENTIONAL POWERPLANT

Power Boiler	9.1345×10^9 Btu		
Feedwater Pump Heat Input	$.0492 \times 10^9$		
High-Pressure Turbine Output		1.0451×10^9	
Low-Pressure Turbine Output		2.1653×10^9	3.2104×10^9
Feedwater Turbine Outlet		$.04922 \times 10^9$	
Condenser Loss		5.9250×10^9	
Total	9.1837×10^9	9.1846×10^9	
Total Turbine Gross Output	3.2103×10^9		
Total Turbine Electrical Output (925,226 kW)	3.1570×10^9 (98.34%)		
Gross Plant Efficiency (Turbine Output)	$= 3.2103 \times 10^9/9.1333 \times 10^9 = 35.15\%$		
Net Plant Efficiency (Electrical Output)	$= 3.1510 \times 10^9/9.1333 \times 10^9 = 34.56\%$		

Figures 41 and 42 show the schematic and H-S diagram of the David Besse Unit I station equipped with an H_2/O_2 direct superheater and H_2/O_2 direct reheater. The reheat temperature, 644 K (700 F) in this case, was not raised to the maximum level in an attempt to utilize the same low-pressure turbines. Table 33 gives the heat balance of the plant with an H_2/O_2 superheater and reheater. An improvement of 4 points in efficiency is noted over that of the plant without the H_2/O_2 superheater and reheater. The equivalent efficiency of an H_2/O_2 combustor is 47.12% as compared to 34.56% for the PWR cycle only. As noted in the schematic diagrams a significant contribution to the gain is due to the increased turbine efficiencies.

System Design

Having established the heat balance from the reactor and the steam addition, the analysis was continued to establish an initial sizing for the superheater installation and an evaluation of the operation and the technology involved. An operational question addressed was the operating O_2/H_2 mixture ratio in the combustor. A nominal stoichiometric mixture ratio was chosen but with a bias on the H_2 rich side similar to that of the supplementary steam generation.

The minimization of noncondensible gases in the steam produced by direct combustion of H_2 and O_2 is important, because these gases will wind up in the extraction heater and the condenser, from which they will have to be removed to maintain correct pressure balances and heat transfer effects. Table 33 presents the quantity of noncondensible gases produced via deviation of the mixture ratio from stoichiometry and/or the presence of contaminant gases in either the

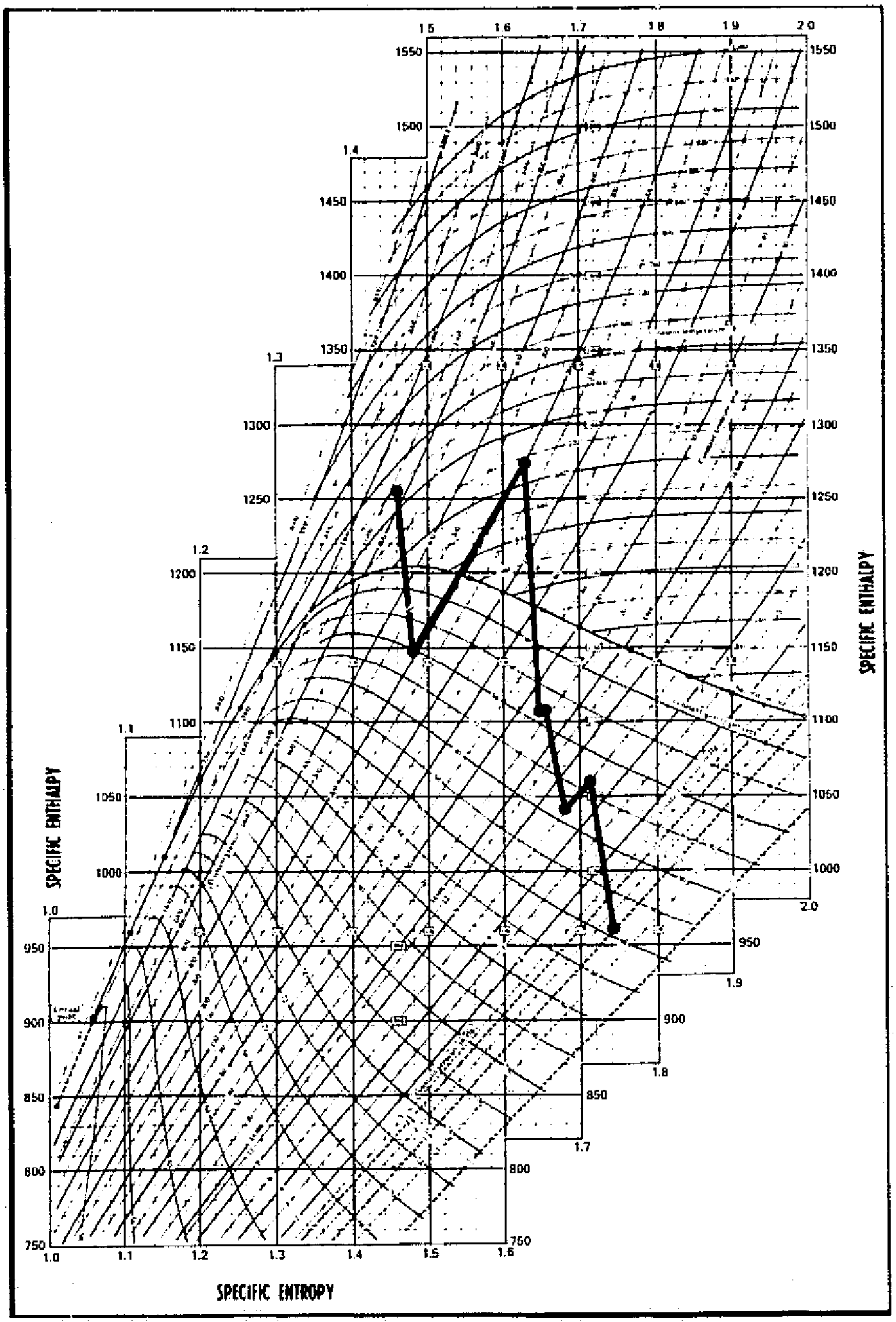

Figure 40. H-S Diagram for Conventional PWR Steam Turbines

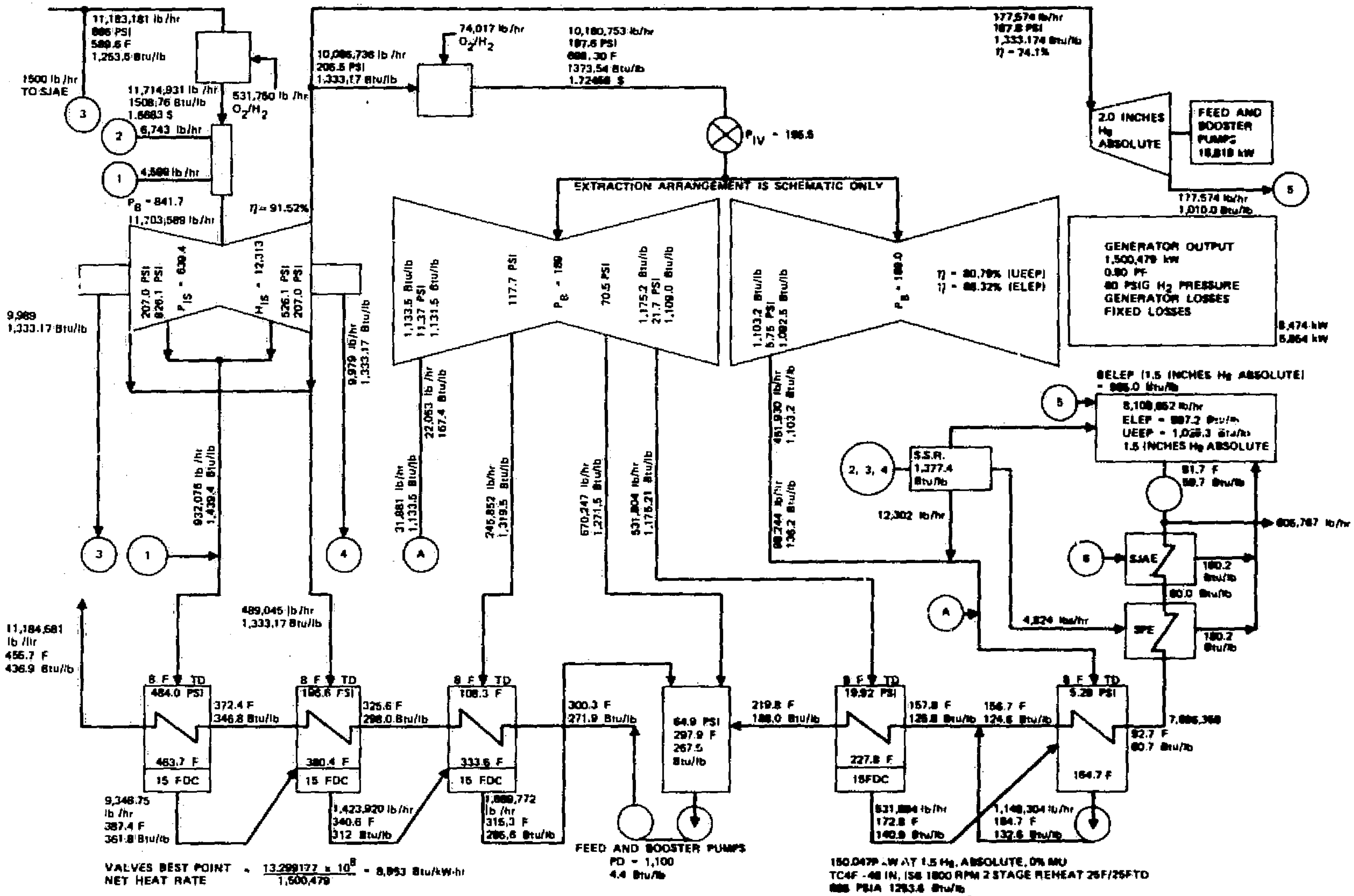

Figure 41. Mass and Heat Balance for PWR Steam Turbine With H_2/O_2 Superheater and Reheater

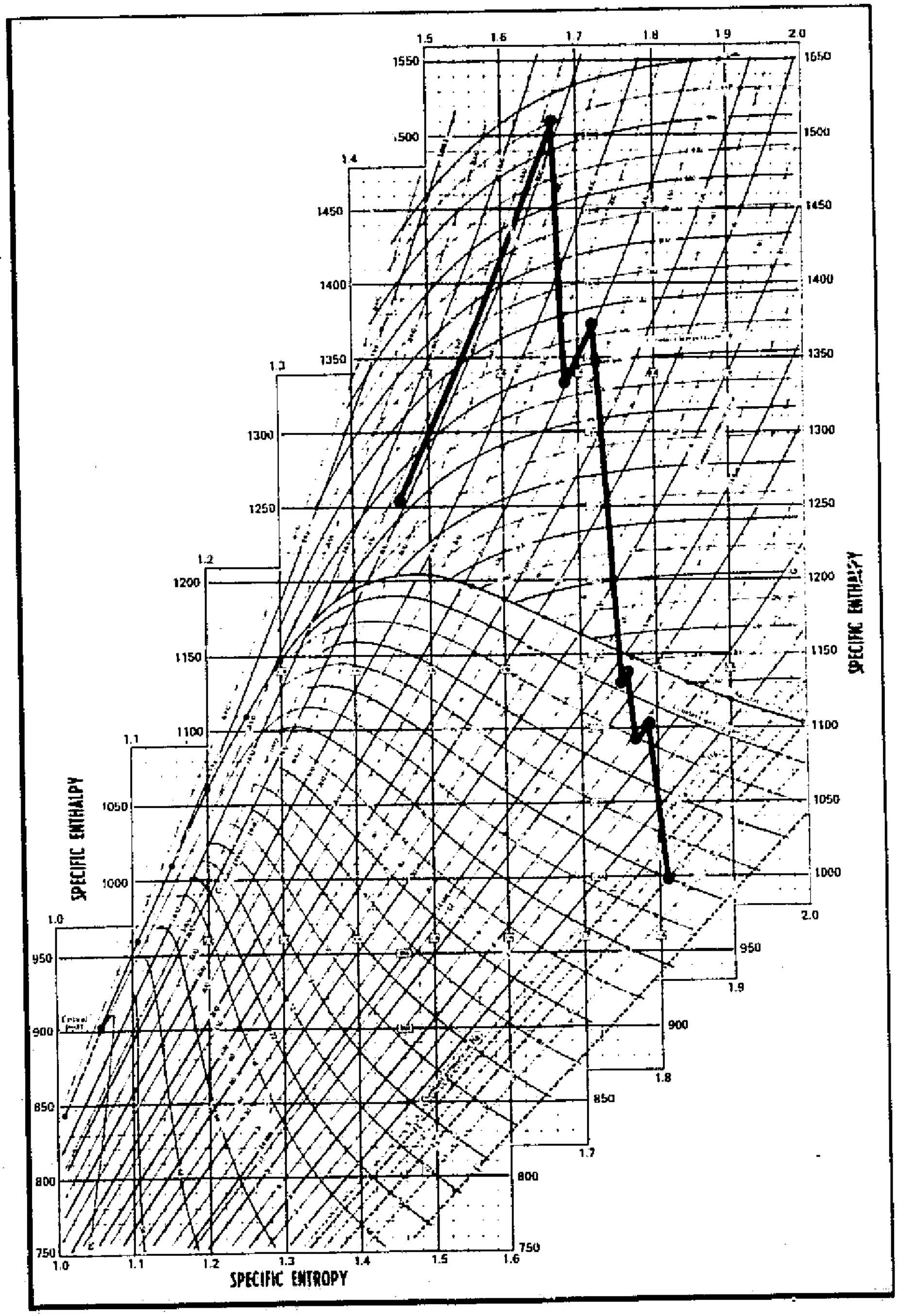

Figure 42. H-S Diagram for PWR Steam Turbines With H_2/O_2 Superheater and Reheater

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TABLE 33. HEAT BALANCE FOR POWER PLANT WITH H_2/O_2 SUPERHEATER

• Power Boiler	9.1345×10^9		
• H_2/O_2 Superheater	3.6558×10^9		
• H_2/O_2 Reheater	$.5089 \times 10^9$		
• Feedwater Pump Heat Input	$.0492 \times 10^9$		
• High-Pressure Turbine Output		1.9562×10^9	
• Low-Pressure Turbine Output		3.2499×10^9	5.2061×10^9
• Feedwater Turbine Output		$.0578 \times 10^9$	
• Condenser Loss		8.0443×10^9	
• Condensate Dump		$.0362 \times 10^9$	
Total	13.3483×10^9	13.3444×10^9	

- Total Turbine Gross Output 5.2061×10^9
- Total Electrical Output (1,500,479 kW) 5.1196×10^9 (98.34%)

- Total Gross Plant Efficiency (Turbine Output) = 39.15%
- Total Net Plant Efficiency (Electrical Output) = 38.50%

	Power Only	H_2/O_2 Combustor	Total
Heat Input Contributed	9.1345×10^9	4.1647×10^9	13.2992×10^9
Electrical Power Output Contributed	3.1570×10^9	1.9626×10^9	5.1196×10^9
Net Efficiency	34.56%	47.12%	38.50%

H_2 or O_2. The presence of noncondensible gases affects the overall system performance, because these gases must be pumped from the condenser against the differential pressure between the condenser and the outside atmosphere. Additionally some of these gases, such as O_2 and CO_2, may result in corrosion effects in the turbine and/or condenser, while the presence of hydrogen may present a safety hazard.

Analysis indicates that the stoichiometric ratio of O_2 to H_2 can probably be maintained via flow proportioning (trimmed from hydrogen concentration) to a deviation of approximately 1/2 of 1% from exact stoichiometry. Present thinking is that it will be most satisfactory to provide a slight excess of H_2 in the combustion products to avoid corrosion possibilities existing with free O_2 in the wet portions of the turbine and in the condenser. There will be no explosion hazard present so long as the H_2 is mixed with the steam; moreover, in the condenser the operating pressures will be so low that an explosion hazard with infiltrating air mixing with the H_2 will not exist. Once the noncondensible gases have been compressed from condenser operating pressure to atmospheric pressure and the exhauster steam condenser, combustible mixtures of hydrogen and air may exist and these will be vented under conditions guarding against accidental ignition.

The H_2 and O_2 flowrates (Table 34) calculated to supply 811 K (1000 F) steam to the turbine are found to be well within the present state-of-art for hydrogen/oxygen combustion. The design of a direct combustion superheater concept is the same as that for the supplemental steam generation shown in Fig. 15 of the Supplementary Steam Generation section . Dimensions for the PWR superheater are shown in Table 34. The technical challenge in this equipment will be to achieve long life and high reliability while yet attaining uniform mixing of the H_2/O_2 combustion products with the steam that is to be superheated. A design goal will be to achieve the mixing without having any of the material operating at a temperature much above the design steam temperature and especially without incurring rapid fluctuations in metal temperatures that could lead to fatigue. However, attainment of these goals does appear to be feasible, and the required investment in equipment directly associated with the combustor would be expected to be quite modest.

A preliminary evaluation of the technology involved in the superheating of steam via direct combustion of O_2 and H_2 resulted in the list of possible technical problems presented in Table 35.

Pressurized Water Reactor Economics

The cost of electricity (COE) for the superheater addition to the pressurized water reactor was based on baseload use. The turbines used in nuclear systems without superheaters are wet turbines. Applications involving less than baseload operation will require two sets of turbogenerating equipment, one when the superheater is operating and one when it is not. It was considered that this added complexity for peaking operation would not be cost-effective. The non-superheat turbine also should be available in the event of downtime of the superheater. However, this contingency was not included in the COE of the PWR superheater application.

TABLE 34. COMBUSTOR COMPONENT FOR SUPER HEAT ON TYPICAL PRESSURIZED WATER REACTOR (600 MWe) - DESIGN CHARACTERISTICS

	Final Steam Temperature		
	600 F	800 F	1000 F
Inlet Diameter*	36 in. ID	36 in. ID	36 in. ID
Outlet Diameter*	38 in. ID	40 in. ID	42 in. ID
Length**	96 in.	96 in.	96 in.
Weight, Hydrogen	3.06 lb/sec	7.4 lb/sec	13.2 lb/sec at 1200 psia
Weight, Oxygen	24.48 lb/sec	59.2 lb/sec	105 lb/sec at 1200 psia

NOTE: Inlet Steam Conditions 550 F - 1045 psia - 8.25×10^6 lb/hr

*Can Double Installation With 70% of These Diameters

**Add 20 ft Mixing Length Down Stream (Min.)

TABLE 35. POSSIBLE TECHNICAL PROBLEMS - O_2/H_2 SUPERHEAT FOR PWR STEAM

1. Noncondensible Gases in Heaters and Condensers.
2. Flame Retention, Flame Detection, Flame Safety.
3. Adds a Cause for Turbine Trips.
4. Mixing of Steam and Combustion Products, Avoidance of Hot Spots, Coring, Metal Temperature Fluctuations.
5. Sophisticated, Reliable, Rapid Acting Control System Will be Required.

The PWR superheater capital cost was estimated as shown in Table 36. Cost of various elements relating to the superheater were estimated and added to the base load plant. As explained earlier in this document, nuclear plants are generally of large capacity. Therefore, a 1000-MWe plant was priced in which 600 MWe is produced by the nuclear reactor and 400 MWe by the superheater addition. The cost of the 600 MWe portion was determined by scaling down from a known 1000 MWe nuclear plant using a scaling factor of approximately 0.75 to compensate for the higher relative cost of smaller plants.

TABLE 36. SUPERHEATER ADDITION TO PWR CYCLE

• 600 MW Nuclear Capacity	
• 1000 psia, Saturated Base Cost at \$500/kW = \$300,000,000	
• 400 MW Added Capacity Via Superheating	
• Additions for H_2/O_2 Direct-Fired Superheater at 1000 F	
• Superheaters and Controls	2,500,000
• Auxiliary Equipment	3,500,000
• Larger Turbine, Cond, etc	50,000,000
• Larger Electrical	30,000,000
• More Complex Startup	2,500,000
Total	88,500,000
• Cost of Added Capacity \$221/kW	
• Design and Construction Period, 9 Years	

In performing these calculations, it can be seen from Table 37 that the installed cost of the combined system is less than for an all-nuclear system (i.e., \$777/kW vs \$900/kW). However, when other factors such as fuel costs and operating and maintenance costs are included, the COE for the combined system is substantially higher than for the all-nuclear system. The primary driving factor again is the cost of fuel.

It seems unlikely then that the addition of superheat to a nuclear system would ever be competitive to an all-nuclear plant until fuel costs for the H_2 system can be reduced substantially to approximately \$1.60/10^6 Btu. This is highly unlikely. When the COE is added to the other complexities such as radioactive water disposal (described in Environmental and Operational Constraints section), the addition of superheat to a PWR system by direct combustion of H_2 and O_2 does not appear to be promising.

TABLE 37. COST OF ELECTRICITY-PWR SUPERHEATER
(Basis - 1000 MW Capacity, 7500 hr/yr)

	O_2/H_2 Superheater		All Nuclear
• Installed Cost x 10^{-6} (Including Interest and Escalation)	600 177 777	Nuclear Superheater Total	900 0 900
• Cycle Efficiency, %			
• Nuclear	34		34
• H_2/O_2	47		
• Plant Work Force	132		120
• Fuel Cost, \$/10^6 Btu	— —	Nuclear 0.25 Hydrogen 6.00	— —
• Yearly Fuel Cost x 10^{-6}			
• Nuclear	11		19
• H_2	130		0
• Total	141		19
• Fixed Costs at 18%, x 10^{-6}	140		162
• Labor at 25 \$/hr, x 10^{-6}	7		6
• Maintenance, 2%, x 10^{-6}	25		30
• Total, x 10^{-6}	313		217
• COE Mil/kW-hr	41.7		30

LIGHT WATER REACTORS - SUPERHEATER FOR BOILING WATER REACTOR (BWR)

System Description

The concept of indirect superheater for boiling water reactors (BWR) was examined for technical feasibility. This candidate PCS involves the indirect superheating of the steam produced by boiling water reactors, either pressurized or boiling water style. In this concept, H_2 combustion would supply the input energy for the superheater of the BWR system. This concept is, in effect, a variation on the fossil-fired superheaters which have been utilized in conjunction with some boiling water reactors.

As indicated earlier, the history of the nuclear generation industry indicates that both PWR and BWR installations are economically competitive only in relatively large sizes. Therefore, it was concluced that it would be reasonable to base this evaluation of a separately fired H_2/O_2 superheater on an installation containing a nuclear reactor of approximately 3400 MW thermal output. Further, it was decided to base the steam output conditions of the BWR reactor on the industry's standard of approximately 6.89×10^6 N/m^2 (1000 psig) steam pressure.

The technical feasibility studies conducted on the indirect-fired superheater were limited to burning the $H_2$with air rather than with either pure O_2 or air enriched with O_2. Thus, the present study was limited to combustion at close to atmospheric pressure, and utilizing air exclusively. It was then possible to establish the baseline operating conditions for the BWR with an indirect H_2-fired superheater.

Two concepts for the heat transfer train of the indirect superheater were examined. The more conventional concept is shown in Fig.43 and its superheater operating conditions are listed in the left-hand column of Table 38. In this concept, the air heater and economizer are proportioned to provide a flue gas temperature to the stack of K (300 F). This is consistent with the conditions utilized in conventional boiler practice, where condensation of moisture on the heat exchanger surfaces is carefully avoided to minimize air heater and economizer corrosion. The relatively low heat exchanger efficiency of 78.7% indicated in column 1 of Table 38 is caused by the nonrecovery of the latent heat of the large amount of water vapor in the gases going to the stack.

An alternative heat exchanger arrangement was examined in which the exhaust gases would be cooled to $\sim$335 K (140 F) and approximately one-half of the moisture in the exhaust gases condensed. For this arrangement, the feedwater flow leaving the condenser at 295 to 305 K (72 to 100 F) is routed through an economizer located in the flue gas stream between the air heater exit and the stack. The operating conditions are shown in Fig. 44 and the efficiency calculations in the right-hand column of Table 38. While the thermal efficiency of this arrangement is $\sim$10% higher than the previous, this section of the economizer will be bathed in dilute nitric acid and would be expected to require special corrosion-resistant materials. Considering the extra economizer expense, and the cycle efficiency losses due to nonregenerative heating of the feedwater, it appears unlikely that the overall economic feasibility of the indirect H_2-fired superheater for BWR stations would depend on which of these two schemes is utilized; neither would it depend on other relatively minor variations in the equipment train provided.

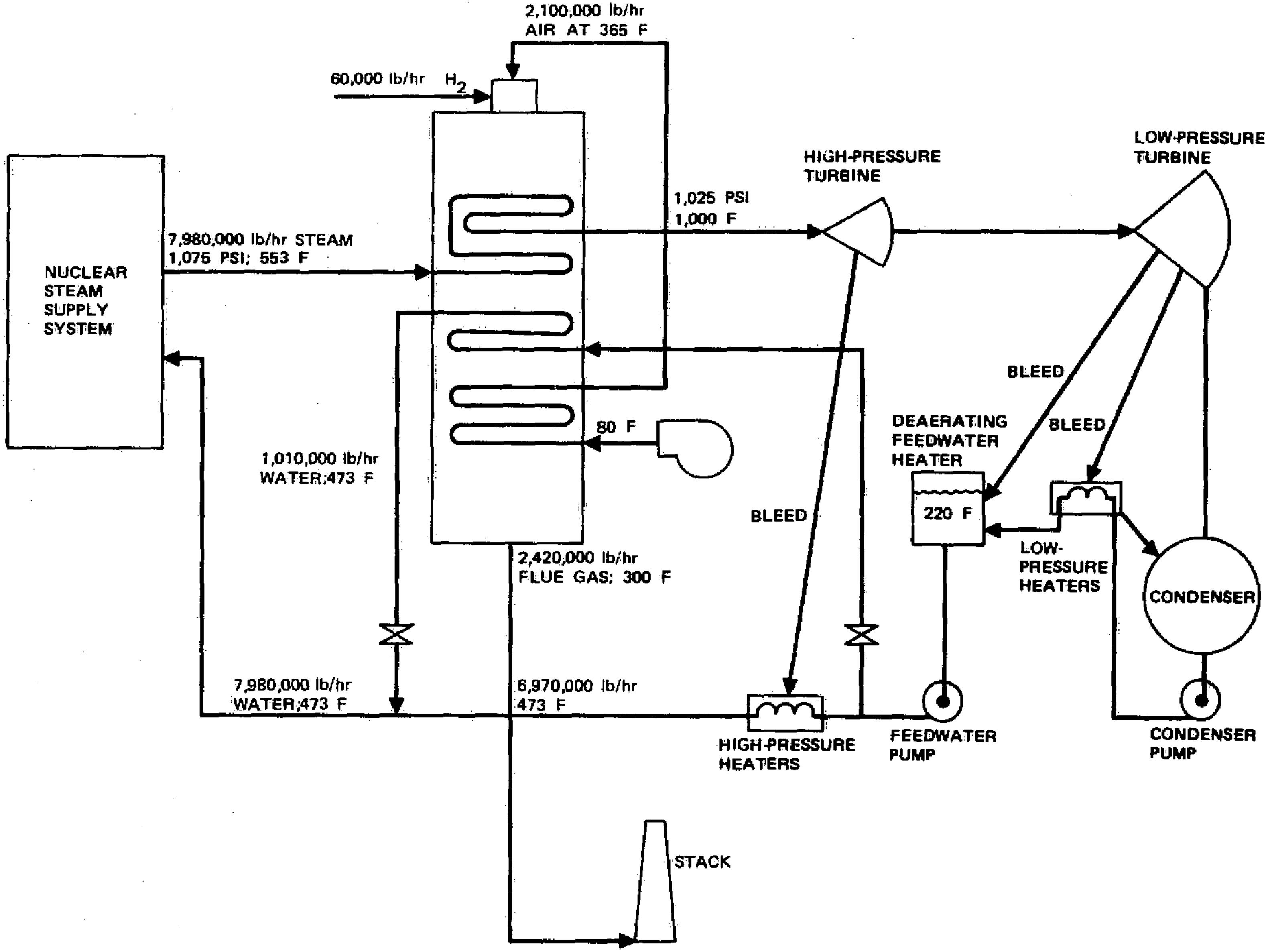

Figure 43. Simplified Flow Sheet - 1000 F Indirect H_2-Fired Superheater

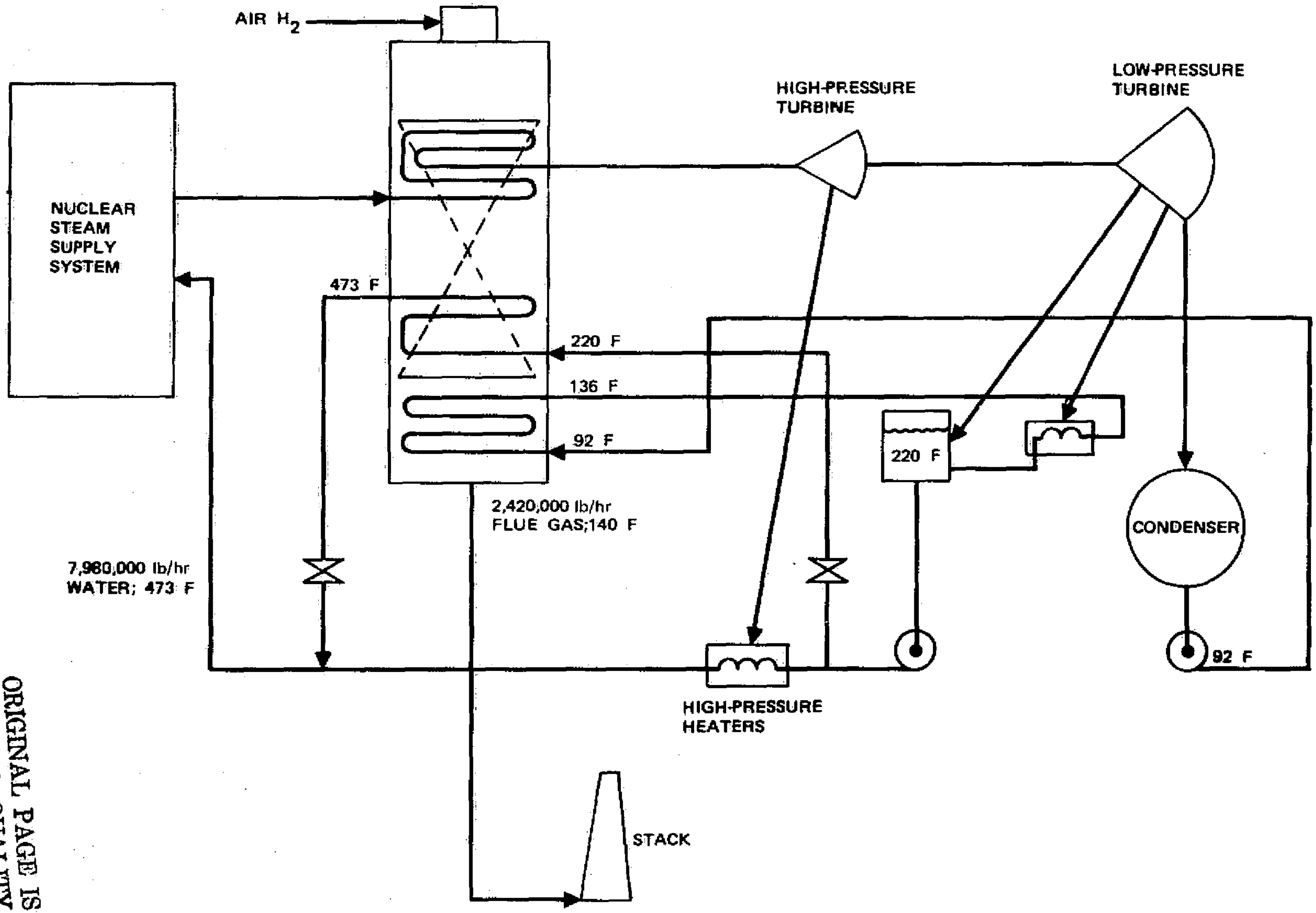

Figure 44. Simplified Flow Sheet - Alternate With Condensing Economizer

TABLE 38. INDIRECT-FIRED SUPERHEATER EFFICIENCY DATA
(1000 F final steam temperature)

	Flue Gas Temperature to Stack	300 F (Fig. 1)	140 F (Fig. 2)
Excess air to burners, %		5	
Excess air at stack, %		15	
Per pound of H_2 burned	Air to burners, pounds	36	
	Air at stack, pounds	39.4	
	Flue gas at stack, pounds	40.4	
	H_2O of combustion, pounds	8.94	
	H_2O in stack gas (by weight), %	22.1	
	Sensible heat to stack, Btu	2500	600
	Latent heat to stack, Btu	9300	5150
	Unburned combustible loss, Btu	0	0
	Radiation loss (0.5%), Btu	305	305
	Unaccounted for loss (1.5%), Btu	915	915
	Total losses, Btu	13,020	7030
	Total available, Btu	61,070	61,070
	% of heat input recovered	78.7	88.5
	% of heat input to superheater	68.6	68.6
	% of heat input to economizer	10.1	19.9

The duty cycle for this equipment would be expected to be that associated with baseloading, consistent with the duty cycle of the necessarily large and expensive nuclear power installation.

No severe environmental operational constraints on the utilization of the H_2-air for the indirect-fired superheater are foreseen. The combustion temperature of H_2 and preheated air is sufficiently high that it may tend to produce an unacceptably high level of NO_x in the exhaust gases unless adequate techniques are employed in the combustion process to prevent this occurrence. However, it is believed that an acceptably low level of NO_x production can be attained by the utilization of staged combustion of the H_2 and air. While staged combustion of hydrogen fuels such as natural gas, oil, and coal is presently not developed to complete satisfaction, it is believed that the absence of carbon in the H2 fuel will make the goal of acceptable NO_x formation (without producing objectionable soot and smoke) more readily attainable. The absence of carbon will also avoid the production of another contaminant: carbon monoxide. Table 39 is an assessment of the status of the technology of H_2/air in indirect-fired superheaters. It is believed that there are no insuperable technical difficulties. The problems will be those of scale, which is very large, and of reliability.

This system was not analyzed further, instead, the BWR system was evaluated because it was determined to be the more efficient of the two systems and, also, lower in cost.

TABLE 39. STATUS OF TECHNOLOGY

1. Combustion: readily developed
2. Separately Fired Superheater
   Scale: very large
   Technical difficulty: modest
3. Environmental
   NO_x readily controlled
   No SO_x or CO

Technical Analysis

Dresden 1 BWR Station. Dresden 1 Station of the Commonwealth Edison Company is one of the earlier BWR steam powerplants; it has been in commercial operation since 1960 with an initial net capacity of 210 MW. The steam cycle shown in Fig. 41 consists of two loops: one flowing through the reactor core produces the high-pressure steam of 6.83×10^6 N/m^2 (990 psia) and 555 K (540 F) for the high-pressure turbine inlet, and the other going through a secondary steam generator produces steam at 3.28×10^6 N/m^2 (476 psia), which is injected into the corresponding intermediate stage of the high-pressure turbine. Th purpose of this secondary steam loop is for regulation of the nuclear core heat output at partial loads by varying the temperature of the inlet water into the core or the amount of secondary steam generated. In later-designed BWR steam cycles, the regulation of the core heat output is accomplished by varying the quantity of recirculating water flow.

Figures 45 and 46 show the schematic diagrams of the Dresden 1 BWR steam cycle without and with H_2/O_2 indirect superheater and reheater, respectively. The reheater is incorporated downstream of the secondary steam generator to heat the steam to 811 K (1000 F) entering the intermediate stage of the high-pressure turbine. To minimize the latent heat loss of the combustion steam, a portion of the feedwater is diverted from the feedwater pump discharge to the indirect superheater and the indirect reheater where the water is evaporated by condensing the combustion steam. Tables 40 and 41 give the heat balances of the two cycles. An improvement of about 3 points in overall efficiency is obtained when based on the sum total of the reactor heat input and the HHV of the H_2 gas consumed (61,036 Btu/lb H_2 or 6830 Btu/lb combustion steam at 770 F). The equivalent efficiencies (42.90 and 42.85%) of the H_2/O_2 combustor were obtained

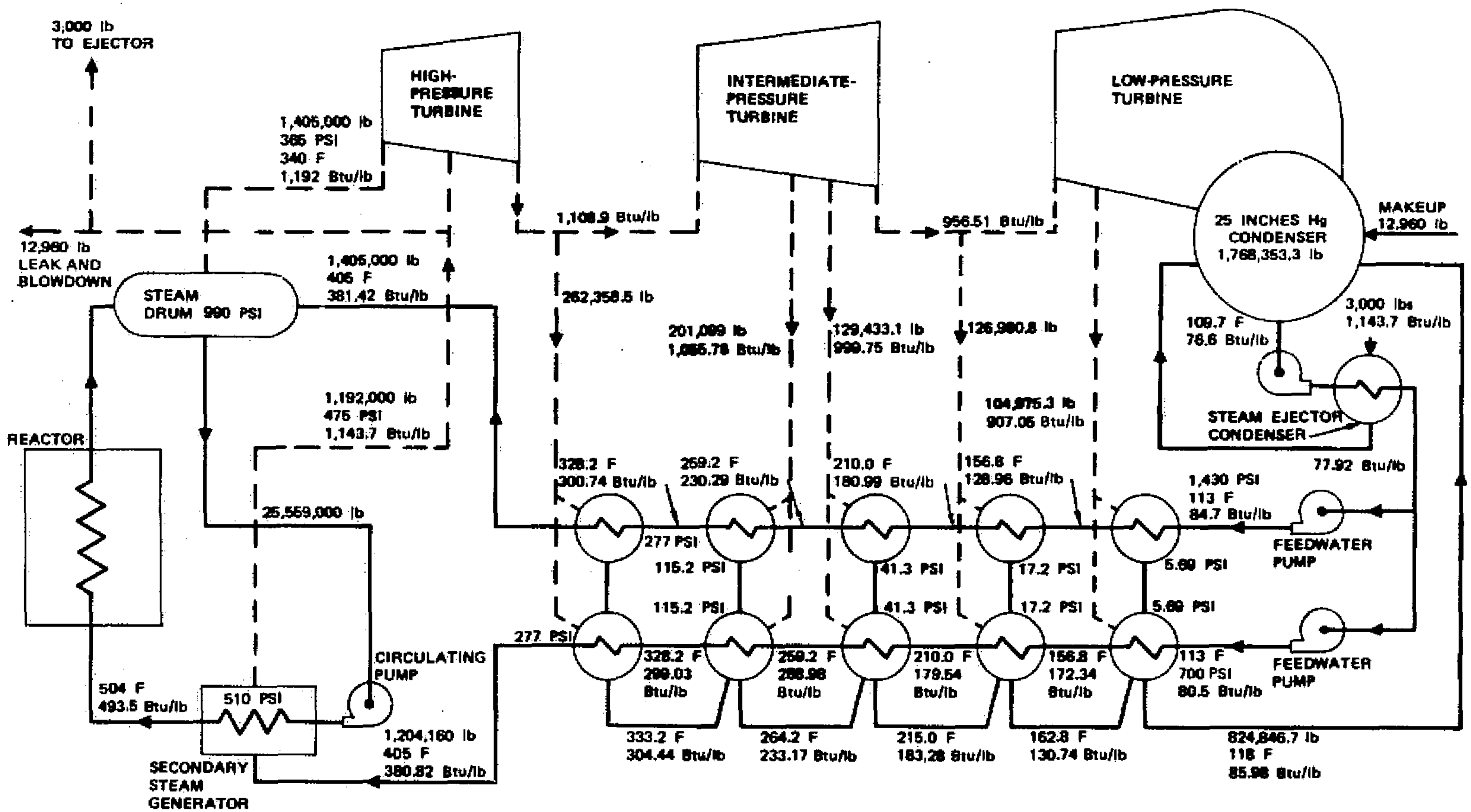

Figure 45. Dresden BWR Steam Powerplant (No Superheater)

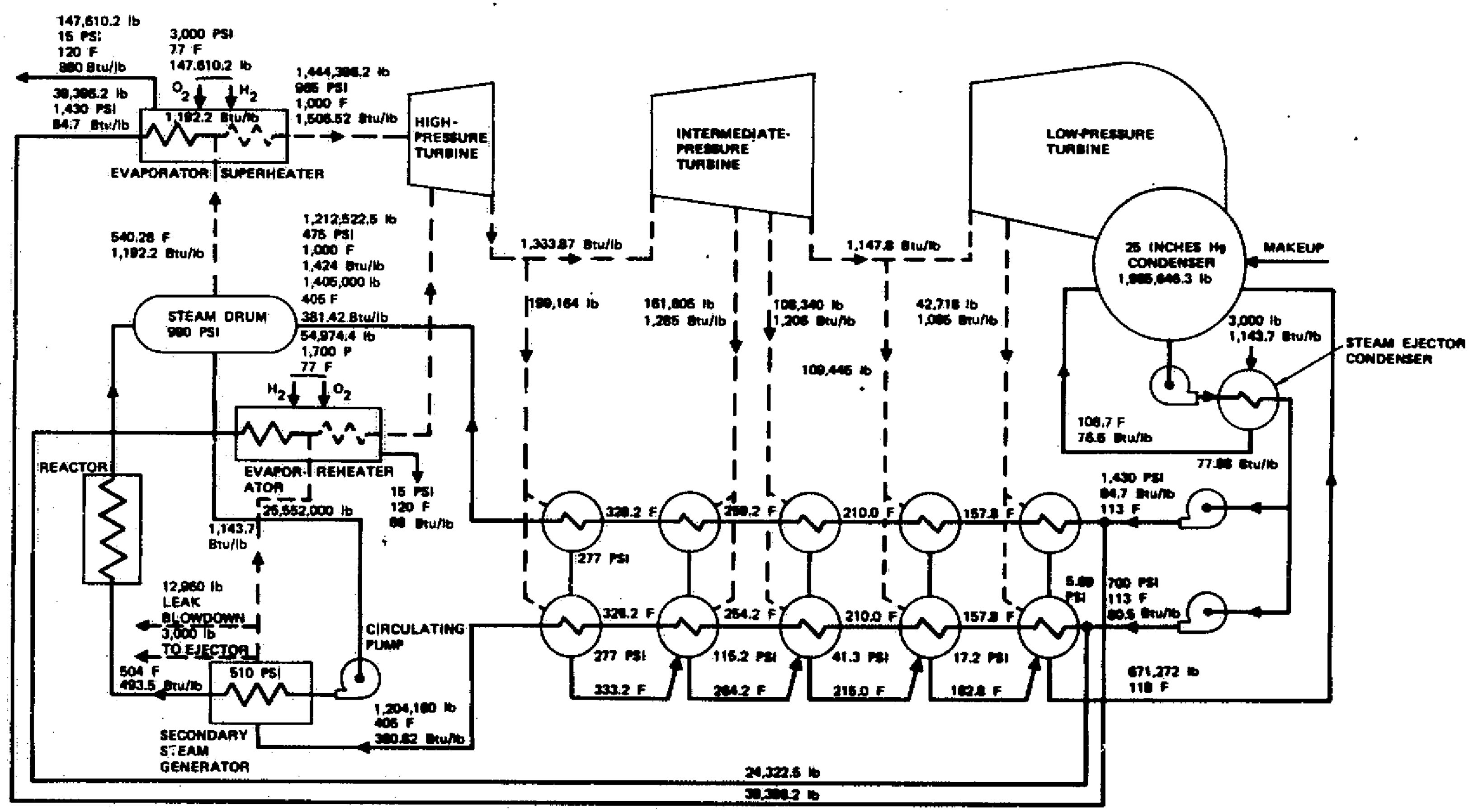

Figure 46. Dresden BWR Steam Powerplant With Indirect H_2/O_2 Superheater, Reheater, and Evaporator

TABLE 40. DRESDEN BWR HEAT BALANCE

Reactor Heat Input to Primary Feedwater	$1139.146(10)^6$ Btu	
Reactor Heat Input to Secondary Feedwater	$918.630(10)^6$	
Feed Pump Energy Input	$12.759(10)^6$	
Condenser Loss		$1369.988(10)^6$
Leakage Loss		$13.830(10)^6$
Power output*		$686.960(10)^6$
	$2070.535(10)^6$	$2070.778(10)^6$
Gross Thermal Efficiency = 33.38%		
Net Thermal Efficiency = 32.76%		

*High-Pressure Turbine	$158.386(10)^6$
Intermediate-Pressure Turbine	$329.641(10)^6$
Low-Pressure Turbine	$198.933(10)^6$
Power Output	$686.960(10)^6$
Less Pump Power	$674.201(10)^6$

TABLE 41. DRESDEN BWR WITH INDIRECT H_2/O_2 SUPERHEATER, REHEATER, AND EVAPORATOR HEAT BALANCE

Reactor Heat Input to Primary Flow		$1139.146(10)^6$ Btu	
Reactor Heat Input to Second Flow		$918.630(10)^6$	
Feedwater Pump Energy Input		$13.177(10)^6$	
Heat Input of Combustion Steam (HHV)		$874.625(10)^6$	
Condenser Loss			$1858.468(10)^6$
Leakage Loss			$13.830(10)^6$
Throwaway Combustion Steam Loss			$11.269(10)^6$
Power Output*			$1062.188(10)^6$
		$2945.578(10)^6$	$2945.755(10)^6$
		Reactor Only	H_2/O_2 Combustor
Gross Turbine Efficiency	= 36.22%	33.38%	42.09%
Net Turbine Efficiency	= 35.77%	32.76%	42.85%

		Increase due to H_2/O_2
*High-Pressure Turbine	$278.952(10)^6$	
Intermediate-Pressure Turbine	$502.373(10)^6$	
Low-Pressure Turbine	$280.863(10)^6$	
Power Output	$1062.188(10)^6$	$375.228(10)^6$
Less Pump	$1049.011(10)^6$	$374.810(10)^6$

by taking the differences of power outputs and heat inputs for the two cases with and without the H_2/O_2 superheater and reheater, as shown in Table 41.

The advantages of the indirect superheater and reheater include the gain in efficiency by the utilization of the waste latent heat of the combustion steam for feedwater heating and the isolation of the radiation-contaminated steam loop which eliminates the contaminated water disposal. However, the requirement of large amounts of heat transfer surface in indirect superheater and reheater will greatly increase the capital cost of the superheater and the reheater.

REPLACEMENT SYSTEM

System Description and Technical Analysis

A preliminary technical evaluation was also conducted on a concept involving the retrofit of existing steam powerplants with boilers that produce steam by direct combustion of H_2/O_2. Such boilers would replace existing boilers that are too old, or environmentally unacceptable, or are unable to be operated in the rapidly responsive mode required in a peaking powerplant installation. The H_2 and O_2 would be combusted at essentially stoichiometric conditions, and a water diluent would be utilized to lower the combustion temperature to within the specification of the existing steam turbines. It is understood that the mission of the plant so retrofitted would be to operate for short periods in the peaking power mode, i.e., for perhaps 4 to 6 hours a day, not necessarily every day, and for a probable total of less than 1000 hours of operation per year.

The technical requirements for the H_2/O_2 steam generator are dictated by the design conditions of pressure, temperature, and flowrate of the existing boiler plant equipment, and also by the duty cycle required in a peaking installation. It is anticipated that the steam flowrates may range up to 1 million pounds of steam per hour at pressure up to 13.8×10^6 N/m^2 (200 psig), steam temperatures up to 811 K (1000 F), and possibly with reheat up to 811 K (1000 F).

The design requirements visualized for the retrofit Rankine steam boiler are listed in Table 42. Of particular significance on this list is a requirement for rapid startup and shutdown to provide economical peaking service. The cycle diagram meeting the design requirements is presented in Fig. 47. The proposed cycle meets the requirements by being capable of a "hot bank" in which it is possible to maintain the direct-fired boiler and separator at an elevated pressure so that the steam supplied to the high- and low-pressure turbines can be matched relatively closely to the temperatures existing in the turbine at the time load buildup is required. Additionally, it is possible to provide steam periodically to the high- and low-pressure turbines at a temperature that will maintain them in a hot condition and thus minimize the load buildup interval.

The water separator is provided in the cycle for two reasons: it will permit the generation of saturated steam of low quality during the standby period and thus the maintenance of the hot bank, and it will provide a safety factor preventing the admission of slugs of water into the high-pressure turbine in the event that, despite all precautions, the control system goes awry and low-quality steam is produced during normal operations.

The reheater is not provided with a water separator because it is expected to be fed only with steam from the high-pressure turbine and with GO_2 and GH_2. In assessing the practicality of the retrofit boiler for conventional Rankine steam plants, it must be recognized that the H_2 and O_2 burned in the reheater will add to the weight of the steam flow leaving the reheater as compared to the practice that existed with a conventional boiler. However, it is estimated that the added steam flow leaving the reheater will lie between 3 and 5% of the

TABLE 42. DESIGN REQUIREMENTS FOR A DIRECT-FIRED H_2/O_2 Boiler

<u>Duty Cycle</u>

Peaking Power Supply

<1 hour from start to on-line
~4 to 6 hours running time/day
<1000 hours/year service

<u>Steam Temperature Regulation</u>

± 5.6 K
(±10 F)

<u>Steam Pressure Regulation</u>

±1%

<u>Steam Quality</u>

<50 ppb of solids in steam

<u>Pressure Safety</u>

- Steam relief provisions per ASME Boiler Code plus H_2/O_2 explosion prevention

<u>Pressure Part Design</u>

- Per ASME Boiler Code
- Cyclic temperature fluctuations avoided

<u>Protection Against Water Admission to Turbine</u>

- Positive

<u>Boiler and Reheater Implications</u>

- Light off under pressure
- Throttle to about 5% load
- Control H_2/O_2 MR to ±1%
- Maintain a hot, high-pressure standby
- Light-gage liners for thermal fatigue protection
- Fast-acting, super-reliable, control system
- Full-flow demineralizer
- Inertial water droplet separator
- Start and shutoff sequences determined by turbine startup capabilities

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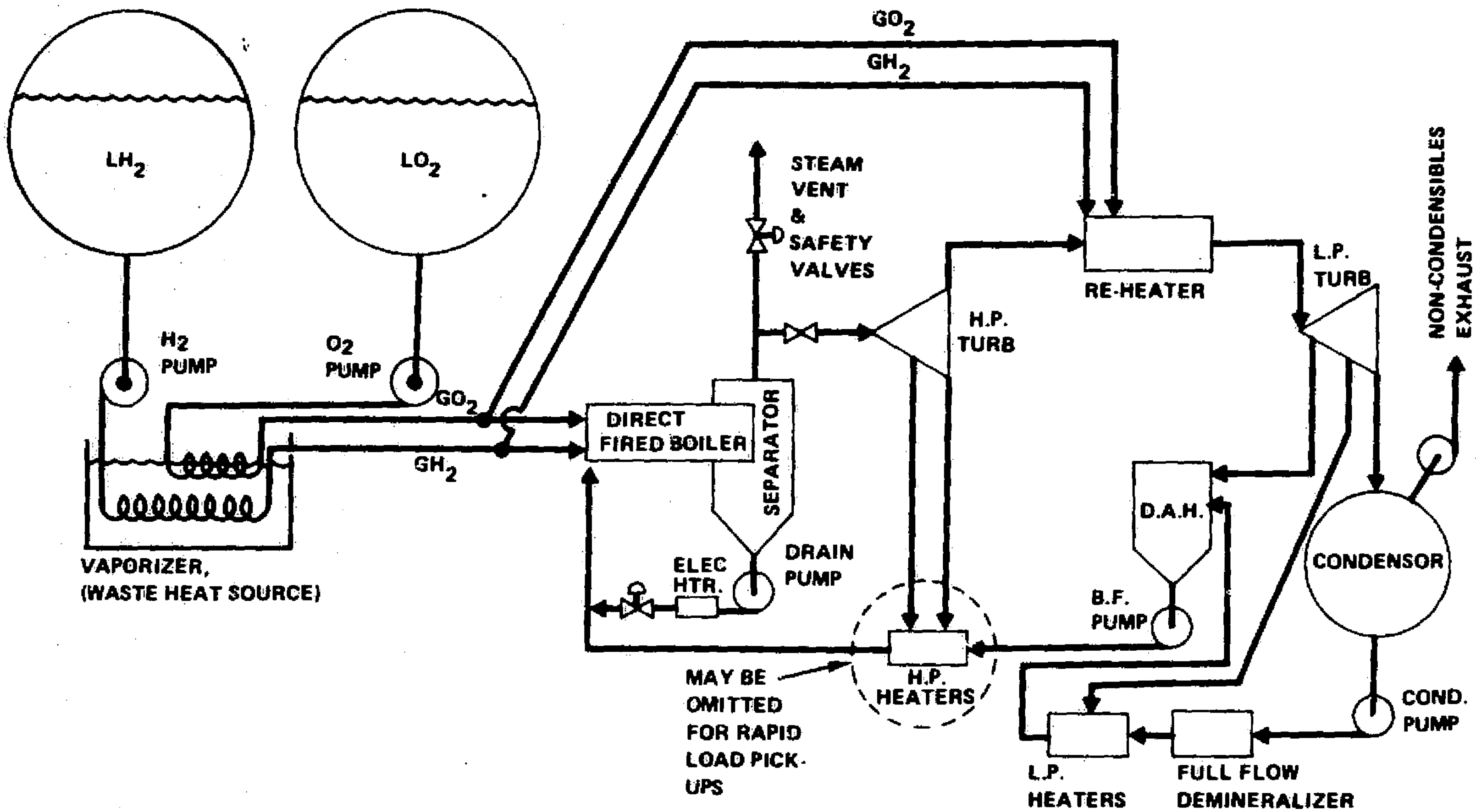

Figure 47. Boiler Replacement Cycle System Schematic

steam flow entering the reheater, and that it will be possible to accommodate this increased flow in the low-pressure turbine.

The cycle diagram shows a dotted circle around the high-pressure heaters and a notation that these may be omitted to facilitate rapid pickup of electrical load.

The exact efficiency of an existing turbine installation supplied with steam from the H_2/O_2 direct combustion boiler, (and direct combustion reheater, if used), will depend upon the exact details of the existing turbine and heater installation. As a rough approximation, the heat input by the H_2/O_2 system is expected to be on the order of 93% of the heat input of the coal-fired system it replaces. This is based on the heat for vaporization of the LH_2 and LO_2 (if used) being supplied from waste heat and not charged to the cycle, and the H_2 and O_2 being pumped as liquids.

The major components of the direct fired boiler-superheater-reheater replacement are listed in Table 43. This table addresses only the combustion equipment and does not cover the installations for storage of hydrogen and oxygen.

No environmental or operational constraints are visualized on the utilization of H_2/O_2 for the direct-fired boiler and reheater.

Table 44 is an assessment of the status of the technology of H_2/O_2 for direct-fired boilers and reheater replacements for Rankine cycle equipment. It is believed that there are no impossible technical difficulties. The areas requiring special attention are specified in Table 44.

This system was not analyzed further in favor of the evaluation of the supplementary steam generation, which was considered to be the better of the two applications.

TABLE 43. MAJOR COMPONENTS OF THE DIRECT-FIRED, H_2/O_2 BOILER FOR RANKINE CYCLE RETROFIT

<u>Boiler</u>
• Igniters, flame safety system
• Burners, high and low capacity
• Combustion chamber, thermal shock liners
• Water injectors
• Water separator, droplets and slugs
<u>Start System</u>
• Recirculating pump
• Heater, electric or steam
<u>Vent System</u>
• Pressure control during startup
• Safety relief
<u>Control System</u>
• H_2 and O_2 ratio control
• H_2 and O_2 final trim, based on sampling
• Steam pressure control
• Steam temperature control
• Start and shutdown transient control
• Safety control (pressure, temperature, water carryover, flameout)
<u>Reheater</u>
• Same as boiler except no water injection
<u>Feedwater Treatment</u>
• Full-flow dimineralizer, <50 ppb solids
<u>Condenser</u>
• Added noncondensible pump or ejector

TABLE 44. STATUS OF TECHNOLOGY OF THE DIRECT-FIRED, H_2/O_2 BOILER FOR RANKINE CYCLE RETROFIT

<u>H_2/O_2 Combustion</u> Straightforward development for: ● Completeness ● Minimum of excess O_2 ● Deep throttling
<u>Water Injection and Evaporation</u> Straightforward development for: ● Uniform mixing ● Evaporation of <u>all</u> drops
<u>Water Droplet and Water Slug Separation</u> Straightforward development for: ● Performance to present boiler standards ● Ability to handle sudden slugs
<u>Control Technology</u> Available
<u>Water Treatment Technology</u> Available
<u>Turbine and Condenser Corrosion</u> Serious problem with O_2-rich steam

ADVANCED STEAM CYCLES

With direct steam operation from combustion of H_2 and O_2, the steam temperature of 839 K (1050 F) imposed by the allowable boiler tube metal temperature is eliminated. If a high-temperature steam turbine with operating temperatures higher than the present limit could be developed, a substantial gain in thermal efficiency could be achieved. Based on a midterm technology assessment, it is believed that, with proper cooling of the turbine blades, nozzles, and casings, maximum steam throttle temperature could conceivably be raised to 1089 K (1500 F) from the present limit of 839 K (1050 F). With the development of steam or gas turbines using ceramic blades, the temperature limit may be extended to beyond 1366 K (2000 F).

Two high-temperature steam turbine cycles with steam throttle temperature of 1366 K (2000 F) were studied in addition to the high-temperature, multiple reheat, steam Ericsson cycles, which is discussed in a separate section. The high-temperature steam Rankine cycle as shown in the schematic diagram of Fig. 48 and T-S diagram of Fig. 49 is essentially a special case of the steam Ericsson cycle, with the number of reheats reduced to one. The steam conditions assumed for this case are 24.13×10^6 N/m^2/1366 K/1366 K (3500 psi/2000 F/2000 F), although the steam inlet pressure could be reduced to 6.89×10^6 N/m^2 (1000 psia) or lower because of the smaller number of reheats. This can possibly reduce the cost of the high-temepratuге steam turbines.

As in the case of the steam Ericsson cycle, the high-temperature steam Rankine cycle will require an indirect surface-type heat exchanger for heat recuperation between the high-temperature, intermediate-pressure exhaust steam with the feedwater. Because of the high steam temperatures, expansions through the turbines will not carry the exhaust steam into the two-phase dome unless indirect recuperation is used between the exhaust steam and the feedwater. The pressure of the exhaust steam through such a recuperator is also limited by the considerations of the steam density and pressure drop across the recuperator. Location of the recuperator at the intermediate-pressure turbine exhaust was selected. The steam, after passing through the recuperator, is given additional expansion to condensing pressure through the low-pressure turbine where further feedwater regeneration is provided by steam extractions from selected low-pressure turbine stages, as in conventional steam power cycles.

Table 45 gives the heat balance of the high-temperature steam Rankine cycle. The net cycle thermal efficiency is 50.8%, which is about 4.3 points lower than that of the steam Ericsson cycle with corresponding steam conditions of 24.13×10^6 N/m^2/1366 K/1366 K/1366 K/1366 K (3500 psi/2000 F/2000 F/2000 F/2000 F). the difference can be attributed mainly to the difference of two reheats.

Indirect heat recuperation with equal mass flows of superheated steam and feedwater is not as efficient as that in a Brayton cycle employing perfect working gases such as helium or air. The real gas effect of widely varying heat capacity in and near the two-phase dome region results in much lower recuperated feedwater temperature than the exhaust steam temperature, and a substantial entropy increase.

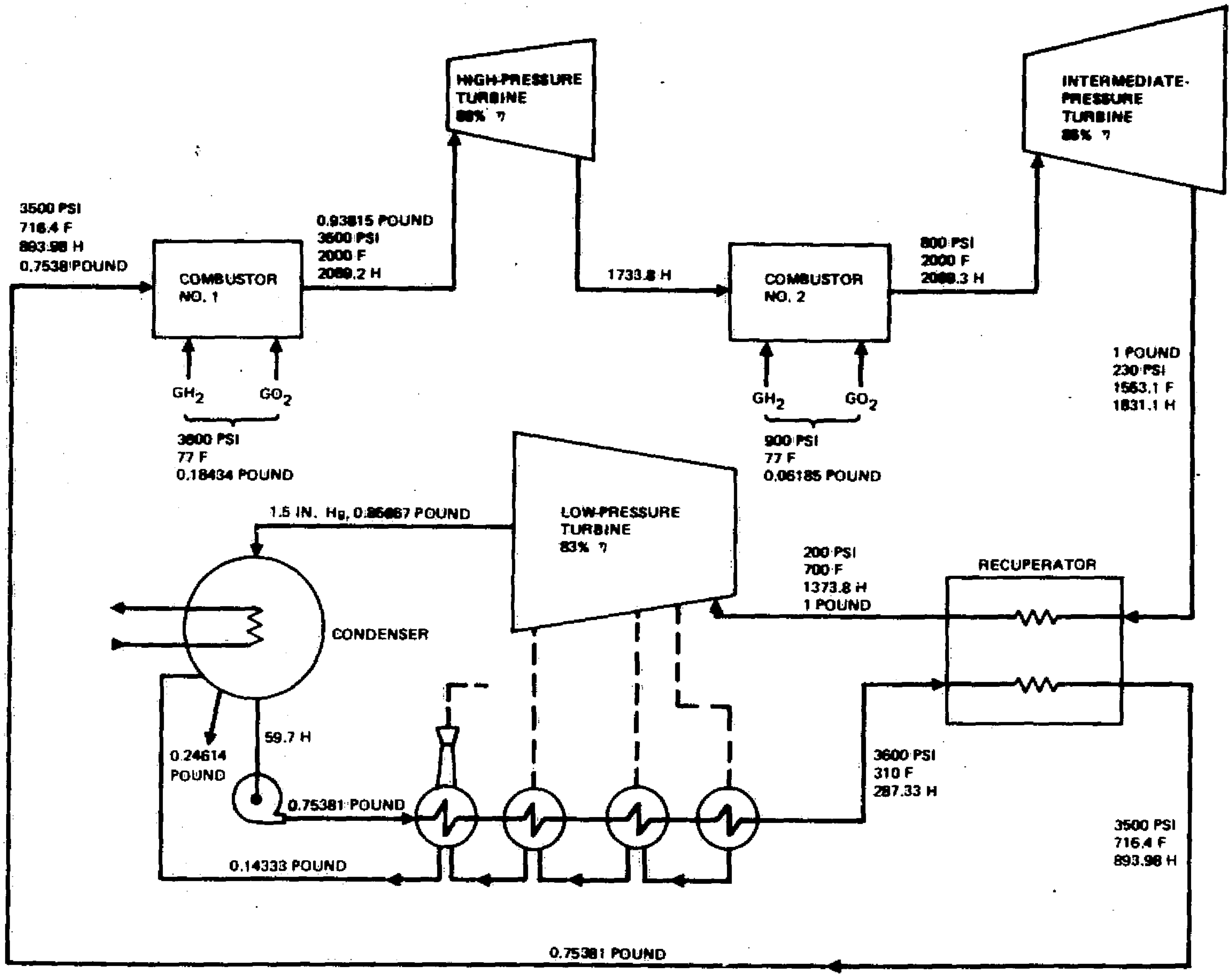

Figure 48. High-Temperature Steam Rankine Cycle
(GH_2/GO_2 Stoichiometric Combustion)

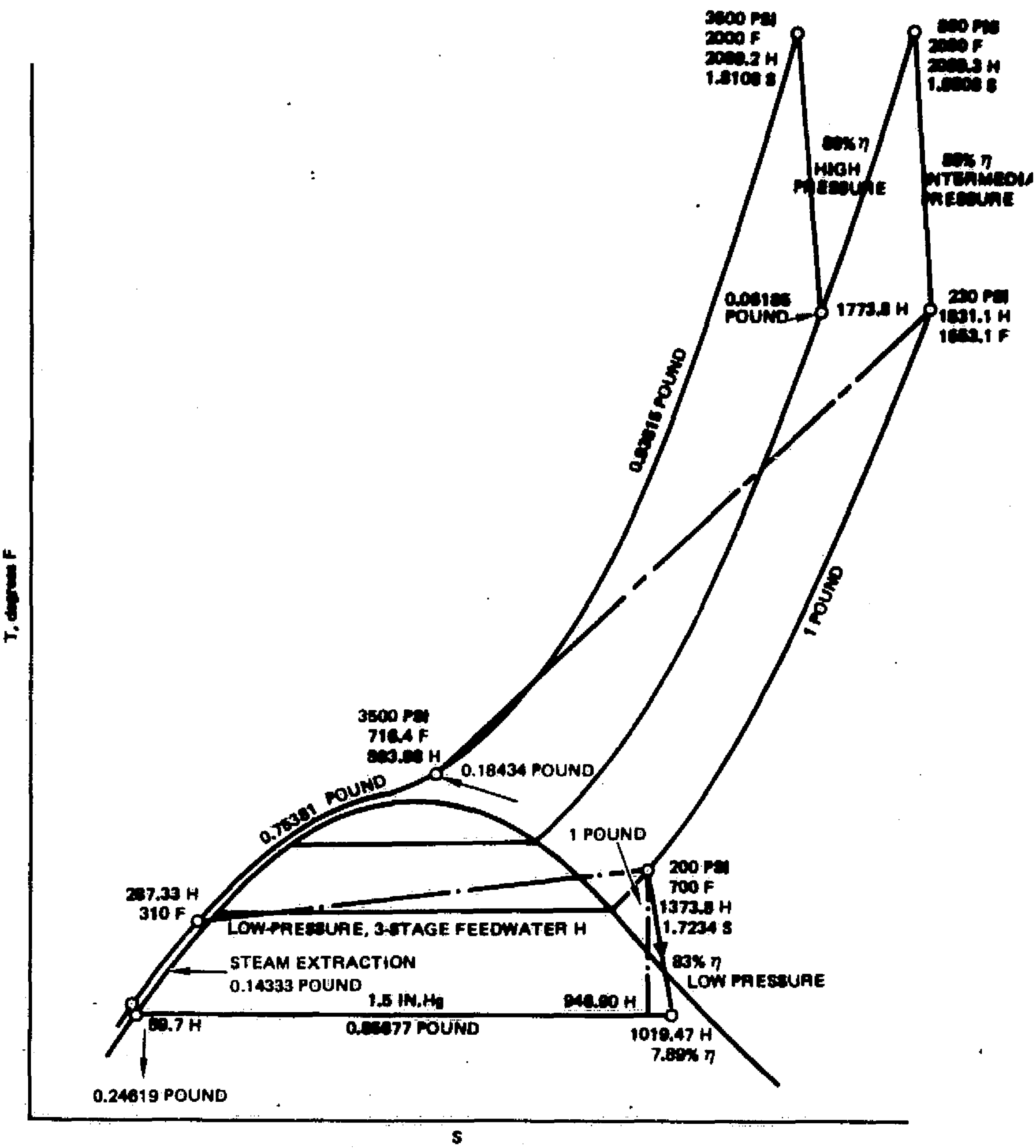

Figure 49. High-Temperature Steam Rankine Cycle With Intermediate Pressure Recuperation and Low-Pressure Steam Extraction for Feedwater Entholpy

TABLE 45. HIGH-TEMPERATURE STEAM RANKINE CYCLE*
HEAT BALANCE

Heat Input to Combustor No. 1	1259.042 (HHV)	1681.478	
Heat Input to Combustor No. 2	422.436 (HHV)		
Feedwater Pump Energy	10.553		
Condenser Loss			823.631
Water Throwaway Loss			3.621
Power Output**			864.779
	1691.031		1691.031
Gross Thermal Efficiency = 51.43%			
Net Thermal Efficiency = 50.80%			

* N/m²/ K/ K
(3500 psi/2000 F/2000 F)

**High-Pressure Turbine	277.130
Intermediate-Pressure Turbine	258.200
Low-Pressure Turbine	329.449
	864.779

To circumvent this adverse effect of indirect recuperation of superheated steam, the concept of partial condensing and partial compression was incorporated in the advanced high-temperature steam cycle. This cycle is essentially a hybrid between the Rankine and the Brayton cycles in the sense that a compressor is utilized to raise the pressure of a portion of the exhaust steam from the recuperator to increase the effectiveness of the temperature approach in the recuperator and to minimize the entropy gain due to temperature degradation. Figures 50 and 51 show the schematic and T-S diagrams of the cycle; the exhaust steam leaving the recuperator is split into two streams. One stream goes to the low-pressure turbine, is condensed in the condenser and the condensate water is pumped through the various feedwater heaters and the recuperator to the combustor, as in the case of the high-temperature steam Rankine cycle. The other stream, however, of near-saturated steam enters the compressor and is compressed directly to the inlet (or throttle) steam pressure without the phase change condensing. The recuperator, the ratio of flow split, and the compressor capacity are designed so that the compressor outlet steam condition would correspond approximately to that of the other high-pressure recuperated stream. Both streams then merge with approximately equal conditions at this point and enter the combustor No. 1.

As evidenced in the high-temperature steam Rankine cycle in Fig. 48, a large temperature degradation occurs across the recuperator between the exhaust steam inlet and feedwater outlet. This is mainly due to the larger heat capacity of the feedwater compared to that of the exhaust steam, which results in a large temperature degradation and gain in entropy if the recuperator has equal flows

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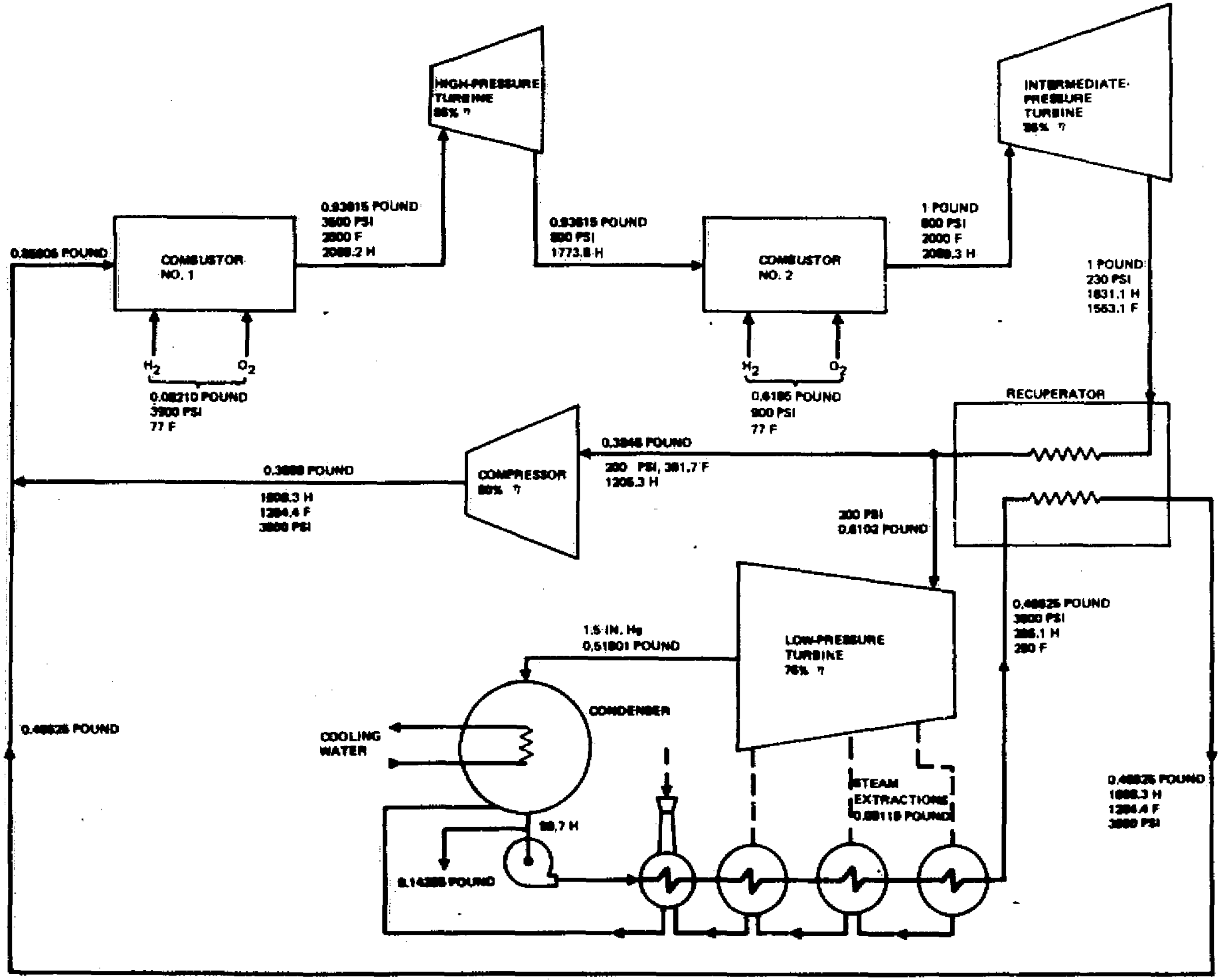

Figure 50. Partial-Condensing Steam Rankine-Brayton Cycle (GH_2/GO_2 Stoichiometric Combustion)

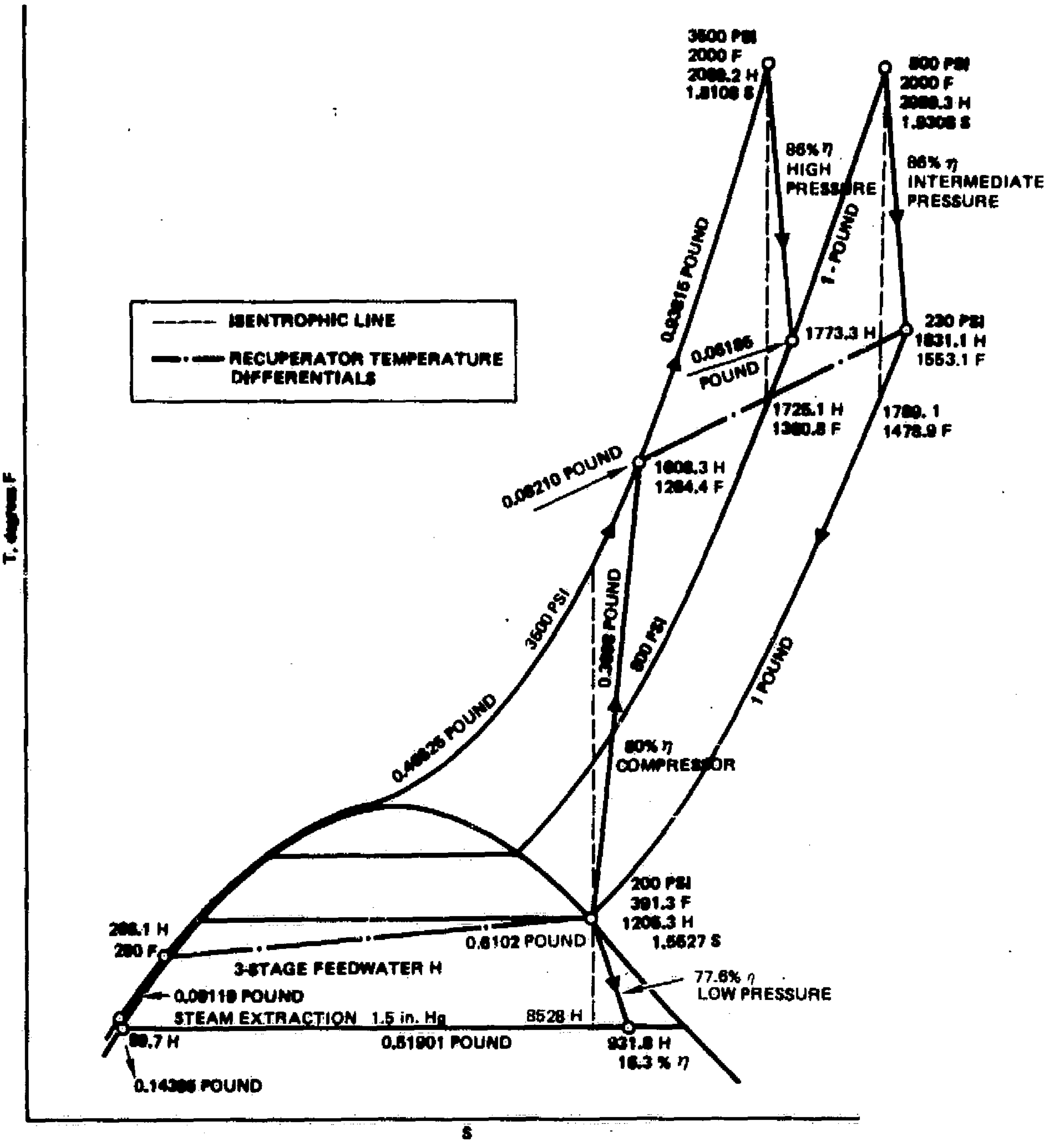

Figure 51. Partial-Condensing Rankine-Brayton Cycle for High-Temperature Steam Turbines (GH_2/GO_2 Combustion)

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through both of its legs. In the partial-condensing cycle, however, only a portion of the total flow goes through the feedwater leg of the recuperator, and the total flow goes through the exhaust steam leg of the recuperator because the other portion is compressed directly to the throttle inlet pressure. Hence, the feedwater outlet temperature from the recuperator is raised by virtue of the unequal flows through the two legs of the recuperator, resulting in less heat input required in the combustor and higher thermal efficiency. As shown in the heat balance (Table 46) the net thermal efficiency of the partial-condensing cycle is 53.66%, which is 2.8 points higher than that of the high-temperature steam Rankine cycle, but still lower than the Ericsson cycle.

While the partial-condensing steam Rankine-Brayton cycle gives substantial performance improvement over the full-condensing, high-temperature steam Rankine cycle, there is also the added complexity of a steam compressor. Furthermore, the requirement of matching the compressor and the recuperator performances in partial-condensing cycles does not provide operational flexiblity of the system.

Neither of these cycles shows as much promise as the Ericsson cycle; therefore, no further evaluation of these cycles was made and the Ericsson cycle was selected for continued assessment.

TABLE 46. PARTIAL-CONDENSING, RANKINE-BRAYTON CYCLE* HEAT BALANCE WITH GH_2/GO_2 COMBUSTION

Heat Input (HHV) to Combustor No. 1	560.743	
Heat Input (HHV) to Combustor No. 2	422.436	
Feedwater Pump Energy Input	6.528	
Condenser Loss		453.541
Water Throwaway Loss		2.116
Power Output**		534.049
	989.707	989.706

Gross Thermal Efficiency = 54.32%

Net Thermal Efficiency (Less FWP Power) = 53.66%

* N/m²/ K/ K
(3500 psi/2000 F/2000 F)

**High-Pressure Turbine	277.130		
Intermediate-Pressure Turbine	258.200		
Low-Pressure Turbine	155.798		
Compressor		157.089	
Power Output		534.039	527.522 (Less Feedwater Pump Power)
	691.128	691.128	

ENVIRONMENTAL AND OPERATIONAL CONSTRAINTS

All systems for the generation of central station power are required to accommodate a rather complex set of operational and environmental constraints. Some of these constraints derive from the service being furnished, i.e., the production of electric power for consumption by industry and the general public. Such constraints are not legally defined but rather inherent in the service. They are summarized in Table 47.

The operational constraints of Table 47 are concerned primarily with the capabilities of the H_2 PCS to meet the load demands imposed upon typical central station power generating equipment. The constraints arise from the nature of the load which, in general, must be accepted by the power system, i.e., accommodating the peaks and valleys. The capabilities listed are not necessarily hard and fast requirements, and deviations may be traded off against other desirable characteristics of a specific PCS. A peaking installation for instance might be permitted a reduced load range in exchange for reduced capital cost and/or particularly rapid startup and shutdown sequences.

A second set of constraints on design and operation are either legally defined or requirements of insurance carriers, and relate to the physical safety of both plant personnel and the public in the immediate vicinity of the operating machinery. Such constraints are typically contained in the ASME Boiler and Pressure Vessel Code and in the local laws of the legal jurisdictions in which the equipment is installed, i.e., states, countries, or cities. A brief summation of the legal safety requirement on powerplant equipment is contained in Table 48. It is recognized that this summation is by no means complete and that the requirements may vary substantially with the locality of the plant.

The legal constraints of Table 48 are binding upon installations of H_2 PCS's. However, new H_2 PCS's that do not meet the applicable codes and regulations as they presently exist may possibly be installed and operated under several mechanisms for change. Initial installations may be permitted to operate as a special case provided the regulatory government agency can be convinced that, while the PCS does not meet the applicable codes, its safety is consistent with coded requirements. Procedural mechanisms are provided to modify code requirements, and these may be pursued to extend the code to cover some H_2 PCS condition that is not initially acceptable. It should be understood, however, that those H_2 PCS systems concepts that are not acceptable under present codes are exposed to some hazard that they will ultimately fail acceptance by that code and be nonusable.

A third set of constraints on design and operation which has become increasingly important recently are the legal requirements on the plant effluents, i.e., its environmental or pollution constraints. A summary of these constraints as presently applicable to fossil fuel-fired stations is presented in Table 49. The nationally recognized standards are those of Table 49, and one may find local variations in which much stricter requirements are enforced. It is to be understood that the environmental requirements imposed upon central station power generation are still in a state of flux, and it is possible that additional requirements and/or a tightening of existing rules will occur. As some

TABLE 47. TYPICAL DESIRABLE OPERATIONAL CONSTRAINTS

<u>LOAD-FOLLOWING CAPABILITY</u>

- Normal - approximately 1%/min load change
- Emerging trips - shed load in seconds
- Minimum load - approximately 1/3 rated
- Maximum load - approximately 110% rated
- Hot bank - desirable

<u>STARTUP AND SHUTDOWN</u>

- Base loading - up to 24 hours
  - self-consistent, i.e., combustion procedures compatible with turbomachinery
- Peaking operation - rapid, 10 minutes or less is desirable

<u>AVAILABILITY FOR OPERATION</u>

- Maximized - 90 to 95% desirable

TABLF 48. FREQUENT LEGAL CONSTRAINTS

<u>SAFETY</u>

- ASME Boiler and Pressure Vessel Codes

  Section 1 - Boilers
  Section 3 - Nuclear Power Plant Components
  Section 8 - Pressure Vessels
- ANSI Power Piping Code, B31.2
- National Electrical Code
- Nuclear Regulatory Commission - Nuclear Safety
- OSHA Related - personnel health and safety

<u>ENVIRONMENTAL</u>

- Environmental Protection Agency
  - Water
  - Air
  - Solids

<u>ZONING AND BUILDING CODES</u>

TABLE 49. TYPICAL ENVIRONMENTAL CONSTRAINTS, COMBUSTION-RELATED EMISSIONS

Emission	Fuel	Maximum (lb/M Btu heat input)
SO_x	Solid	1.2
	Liquid	0.8
	Gaseous	0.2
NO_x	Solid	0.7
	Liquid	0.3
	Gaseous	0.2
Particulates	All Fuels	0.1

of the H_2 PCS evaluated in Task 1 would be installed in nuclear stations, environmental constraints upon such stations are also of interest and are complex and strict.

The influence of the environmental and operational constraints upon the economic viability of the H_2 PCS candidates evaluated in Task 1 is presented for each system in the material which follows.

Two concepts were considered for the application of H_2 PCS's to improve the economic position of nuclear generating stations. Direct combustion of H_2 and O_2 in the steam line from the nuclear supply system to the turbine for the purpose of superheating the steam was considered to be applicable to nuclear stations using the pressurized water reactor system. Combustion of H_2 with air in an indirect-fired heat exchanger, i.e., one in which the heat resulting from the combustion is transferred through metal walls to the steam, was considered to be applicable to either pressurized water reactor systems or boiling water reactor systems. The indirect H2-fired superheater avoids the production of excess condensate through the H_2/O_2 combustion.

The factors considered in evaluating the environmental constraints upon H_2 PCS applications to nuclear steam systems were: (1) noncondensible gas entrainment, (2) effects of H_2 purity, (3) nitrogen oxide (NO_x) emissions, and (4) possible radioactive emissions. An evaluation of possible ratioactive contamination of the condensate indicated that, even with the separation nominally achieved between the nuclear loop and the steam loop of the PWR system, the steam loop may become radioactive. This may result either from direct contamination from leaks in the boiler or from contaminants in the nuclear steam supply system. With the direct combustion concept, the steam formed from H_2/O_2 combustion mixes directly with the possibly radioactive nuclear steam supply system steam, and both the quantity and the disposal of the combustion-produced condensate must be considered. This problem can be reduced by use of a deaerating feed water heater to remove gaseous impurities and by routing the excess condensate through a demineralizer before discharge. This will require a larger demineralizer to handle the greater flows, and larger-capacity holding tanks to contain the excess condensate produced until the radioactive levels are down to an acceptable degree. It is expected that existing nuclear plant equipment is adaptable to this purpose, but that larger-size equipment will be necessary. The indirectly fired superheater concept does not encounter the problem of producing excess condensate that may be radioactive and which must be disposed. Since minimal sulfer is expected to be present in the H_2 supplied, there should be no sulfer oxide emissions problem. The wide flammability limit of H_2 would be expected to permit control of nitrogen oxides formation to remain below legal limits by allowing staging of the combustion to provide low flame temperatures. Additional tools available for NO_x control include flue gas recirculation, premixing of air and H_2 and lean burning, and specific burner designs.

The major problem with the application of H_2 PCS's to nuclear systems via superheating of the steam appears to lie in the operational constraints. Most nuclear steam installations operate at about 6.89×10^6 N/m^2 (1000 psia) steam pressure, and the economy of nuclear stations is such that large units, on the order of 1000 MWe/unit are typical. Substantial improvements in station heat rate are possible by raising the steam temperature from the saturation or near-saturation conditions typical of today's nuclear stations to the 811 K (1000 F) steam temperature typical of today's fossil-fueled steam stations. The operational problems arise in that such superheating vis H_2 combustion is economically justifiable only for a peaking duty cycle. The cost of H_2 (and of O_2 if utilized) relative to the costs of nuclear fuel are such that continuous production of power by the combustion of H_2 cannot be justified economically, even if the incremental capital cost of the H_2-fueled capacity is very low. Thus, an economically competitive combined nuclear/H_2 PCS can be justified only for peaking service from the H_2-fueled capacity. However, the realities of turbine construction, steam piping engineering, and thermal strains make it impractical to operate the combined nuclear/H_2 PCS at high steam temperature during the peaking periods and at saturation during the sustaining periods. Thus, this operational constraint serves to make the combined nuclear/H_2 PCS economically unfeasible.

A series of H_2 PCS concepts were evaluated which may be categorized as "advanced steam" systems which rely on direct combustion of H_2 and O_2 to produce and/or to superheat steam to a variety of pressures and temperatures to attain higher thermodynamic efficiencies than are typical of today's fossil fuel-fired steam stations. Such advanced steam systems are typified by the cycles discussed in earlier sections of this report, i.e., the Ericsson steam cycle, the partial condensing cycle. The environmental effects of such systems are expected to be very modest and easily controlled to meet the legal constraints. In the ideal situation where pure H_2 and O_2 are combusted at exactly the stoichiometric ratio, the only product is distilled water, which is environmentally completely acceptable. Electrolytically produced H_2 and O_2 are very nearly the equivalent of pure H_2 and O_2, and their combustion products would be expected to be unobjectionable. When the H_2 is produced from various hydrocarbon feed stocks (and one would expect the only practical feed stock for the H_2 PCS under consideration to be coal), one may expect a variety of impurities to be present in the H_2. Table 50 lists some impurities and the range of quantities in which one would expect them to be present. Oxygen that is produced by cryogenic separation from the air is also not entirely pure; argon and nitrogen may be found in quantities on the order of 0.5%. When this nonelectrolytic H_2 and O_2 is combusted, the noncondensible gases produced will be pumped from the condenser

TABLE 50. HYDROGEN FROM COAL GASIFICATION, TYPICAL COMPOSITIONS - % BY VOLUME

	Process	
	Koppers/Totzik	V-Gas
CO	0.1	0.1
H_2	93.1	94.3
CH_4	5.5	4.8
N_2 and Argon	1.3	0.8

and exhausted to atmosphere. As the quantity of sulfer present in the H_2 and O_2 is essentially zero, it is anticipated that there will be no difficulty with sulfer dioxide emissions. Any N_2 content of the H_2 and O_2, however, may be partially converted to nitrogen oxides in the combustion process. It is expected that NO_x emissions can be controlled by either modifying the combustion process to premix the H_2 and steam, and thus lower the flame temperature, or by catalytically treating the gases as they are exhausted from the condenser. Catalytic treatment is practical in this situation because the volume of gases that will be exhausted from the condenser of a hydrogen combustion system is much less than from a boiler. Additionally, the condenser exhaust gases will be clean and low temperature, and not contain either sulfer or particulate matter.

It is the operational constraints on the "advanced steam concepts which tend to make them less attractive than some others for H_2 PCS applications. These systems are capable of providing substantially improved overall thermodynamic efficiencies as compared to existing fossil stations. The costs of H_2 and O_2 relative to alternative fuels, however, is so high that, not withstanding the improved thermodynamic efficiency, a H_2 PCS is economically competitive only in peaking power application, i.e., those situations where its low capital cost relative to competitive energy systems offsets the high fuel cost. Unfortunately, the high pressure and temperature of the advanced steam cycles makes it impractical to utilize them in the peaking mode. They could not be brought on the line quickly, run for a relatively short period of time, say 2 to 4 hours, then taken off the line. Technically, they would be more suitable for continuous operation.

A third class of H_2 fuel PCS examined may be categorized as "supplementary steam". These systems may consist for instance of the replacement of an existing boiler which is no longer operable by direct combustion of H_2 and O_2. Hydrogen and oxygen may be utilized to supplement the steam produced by an existing boiler which has been derated for environmental or fuel switching reasons. Entirely new systems can be visualized in which additional capacity is provided in the turbogenerator train and that capacity is supplied during peaking periods with directly generated steam from O_2 and H_2. Environmental constraints on these systems are similar in most respects to those discussed above for the advanced steam systems. It is believed that environmental constraints would pose no problems that could not be resolved. It is with respect to operational constraints that the supplementary steam system has advantages over the systems previously discussed. The same economic constraints apply here as for the H_2 PCS systems, i.e., only peaking power via the H_2 PCS is economically viable. However, peaking-type operation is more readily attainable with the supplementary system that it appears to be with either the nuclear or advanced steam systems. Consider a strictly supplementary system in which steam is normally supplied continuously by a fossil fuel-fired system, and extra steam and/or extra reheating is provided by the H_2 PCS only during peaking hours. One can visualize very rapid response to load in that the entire system operates at all times with time-constant temperatures throughout the system. The challenge for the H_2 PCS is to respond to peaking load demands by providing extra steam, and extra energy to the reheater if one exists, without significant changes to the temperatures throughout the system. Such H_2 PCS's appear technically feasible with respect to both environmental and operational constraints.

Another class of candidate H_2 PCS's are the gas turbines and the various gas turbine/Rankine cycle combinations. These systems would typically burn H_2 with air, and their environmental problems would be expected to be similar in many respects to those in existing gas turbines burning natural gas and air. Such natural gas burning turbines are typically hard put to meet the more stringent local requirements for NO_x emissions. Water or steam injection into the flame is frequently employed to reduce flame temperature and control those emissions. Such injection of course is wasteful of energy. The very wide flammability limits of H_2 with air are expected to make it possible to control NO_x formation by premixing and lean burning of the H_2 and air mixture, thus controlling the flame temperature and avoiding NO_x formation. As a backup, water and/or steam injection could be used similarly to present gas turbine practice with natural gas. The low sulfur content of the H_2 supplies anticipated would be expected to avoid excessive SO_x discharge. Smoke production would be entirely absent as significant carbon products are not present. Noise and heat conditions would be similar to those in existing gas turbine installations. With respect to operational constraints, it is anticipated that a H_2 gas turbine fuel will permit operating practices not attainable with present-day distillate oil or natural gas. Hydrogen is such an excellent coolant that it may well be applied to blade cooling, thus permitting higher operating temperatures. The wide range of flammabilities and the low heat radiation from a H_2 flame, compared to flames of carbonaceous fuels, should simplify burner can configurations and perhaps further permit the utilization of higher flame temperatures with consequent higher thermodynamic efficiencies.

TECHNOLOGY ASSESSMENT

In conventional steam powerplants, the main constraint to higher steam throttle temperatures than the present limit of 839 K (1050 F) is the steam superheater tube metal temperature. With the H_2/O_2 direct steam generator, the combustor wall cooling can be effectively provided by injection of either the diluent water (as used in the evaporator and superheater) or the low-temperature steam (as used in the reheater). In these cases, the temperature constraint then falls on the steam turbine.

Although a few experimental steam turbines have operated at 922 K (1200 F) steam temperatures, no attempt has been made to operate the commercial steam turbines beyond the present temperature limit of 839 K (1050 F). This is in sharp contrast to the high-temperature capability of the commercial gas turbines. Stationary gas turbine power units presently are being operated at around 1089 K (1500 F) turbine inlet temperature, while the commercial turbojets are being run at around 1255 K (1800 F), turbine inlet temperature. Operating temperatures as high as 1422 K (2100 F) have been reported in some advanced jet engine gas turbines. While there is no fundamental reason why a steam turbine cannot have the high-temperature capability of the gas turbine, there are technological and economic justifications that have deterred its advancement beyond its present temperature limits. These deterrents are discussed in the following paragraphs:

1. <u>High Steam Pressure</u> - Steam turbines generally are operated at much higher inlet pressures of $>16.5 \times 10^6$ N/m^2 (>2400 psia) and pressure ratios of about 3000 than are gas turbines ($<1.7 \times 10^6$ N/m^2 or <250 psia) and pressure ratio <15. Hence, there are more stages required in a steam turbine (30 to 40 stages) than are required in a gas turbine (3 to 5 stages).

2. <u>Size and Cost</u> - Because of the large number of stages and thicker casings required as well as the large size and capacities of steam turbines (>200 MW compared to 75 MW maximum gas turbine capacity), the cost becomes a main factor of consideration. Therefore, low-cost materials of construction (alloy steels) invariably are used in steam turbines as contrast to superalloys (such as IN-706, M-21, MAR-M-240, 246 Cast) commonly used in gas turbines. Alloys steels (such/as 0.37 C/1.25 Cr/ 1.50 Mo/0.30 V steel forings for rotors, 0.06 C/11.5 Cr/ 0.40 Mo/0.5 Ni hot-rolled stainless steel for buckets and 0.08 C/16.00 Cr/1300 Ni/ 2.00 Mo/0.8 Cb austenitic cast steel for casings used in steam turbines) have only about onetenth the cost of the superalloys.

3. <u>Cooling</u> - All high-temperature gas turbines require a substantial amount of bleed-off from the compressor for cooling the buckets, the rotor, and the casing. This bleed-off loss is the main reason for the relatively lower efficiency of high-temperature gas turbine compared to steam turbines. Because of the large number of stages, and large surface area subject to intense convective heating by the high-temperature and high-pressure steam, cooling in steam turbines would significantly degrade the turbine efficiency.

4. Thermal Expansion - Longitudinal thermal expansion is more critical in steam turbines because of the larger number of stages and longer rotor when operating at higher temperatures.

5. Thermal Distortion in Governing Stage and Nozzle Block - Multiple admission governing stages are invariably used in steam turbines to maintain high partial load efficiencies. Thermal distortion due to uneven temperature distribution has been a factor limiting the steam temperature and startup time.

On a near-term basis, it appears that the economic and technological justifications mentioned previously will prevail, and it is expected that the current steam turbine practice will continue at least until 1980. However, projected future energy shortage and high cost will undoubtedly exert a pronounced impact on the need for high-efficiency and high-temperature steam turbines (or even alkaline metal Rankine cycle turbines). On a midterm basis; it is envisioned that the cooling techniques will be developed for high-temperature steam turbines. The amount of cooling required would be minimum if steam temperatures no higher than 1033 K (1400 F) and superalloys were used. The steam temperature is expected to be raised to 1366 K (2000 F) after the year 2000. It is projected that high-temperature materials including ceramics will be developed and can be suitably used in steam turbines with minimum cooling. Table 51 shows the general turbine technology growth guidelines for steam and gas turbines.

TABLE 51. TURBINE TECHNOLOGY GROWTH GUIDELINES

Now to 1980	1980-2000	After 2000
Steam Turbines		
24.1×10^6 N/m^2 (3500 psia) 839 K (1050 F)	24.1×10^6 N/m^2 (3500 psia)* 1033 K (1400 F)	24.1×10^6 N/m^2 (3500 psia)* 1366 K (2000 F)
Alloys Steels, Reheats, and Steam Extractions for Feedwater Heating	Superalloys, Multiple Reheats, and Alloy Steel Recuperator	Superalloys, Blade and Casing Cooling, Multiple Reheats, and Superalloy Recuperator
Gas Turbines		
1.7×10^6 N/m^2 (250 psia) 1255 K (1800 F)	2.1×10^6 N/m^2 (300 psia) 1644 K (2500 F)	2.1×10^6 N/m^2 (300 psia) 1922 K (3000 F)
Superalloys, Blade and Casing Cooling, Alloy Steel Recuperator	Superalloys, Blade and Casing Cooling, Reheating, and Compressor Intercooling, Superalloy Recuperator	Ceramic Blades, Blade and Casing Cooling, Reheating and Compressor Intercooling, Superalloy Recuperator

*Listed maximum temperatures are for reheat steam. High-pressure throttled steam at lower temperatures.

DEVELOPMENT PLAN

PROGRAM APPROACH

The full-scale supplementary steam generation combustor system will be developed in stages to allow the most time to develop the more difficult components and still maintain a short and low-cost schedule to plant proof-of-concept testing. Figure 52 provides an estimate of the scheduling required to complete the program outlined. A still shorter schedule can be realized if the proof-of-concept combustor design is started during combustor system development test effort. Compression of this type can result in pilot plant proof-of-concept testing in the third year of the program. However, this results in higher costs early in the program although the overall program cost should not be affected.

Initial development effort will concentrate on the combustor, ignition, and cooling systems. During this time, the control system will be designed so that development of the complete combustor system can begin at an early time. The schedule for proof-of-concept combustor design is delayed until after combustor system development testing is complete. This was done as a conservative planning approach. It is possible to begin this design effort earlier at little or no risk and thus arrive earier at proof-of-concept testing. By advancing, this design effort, the proof-of-concept plant testing can begin as early as the third year of the program.

COMBUSTOR DEVELOPMENT

The first phase of this program will be to develop the combustor, ignition. and cooling systems. This will be followed by performance tests. The purpose of this effort is to obtain the design information needed for the combustor system. Small-scale component tests will be conducted to evaluate various design concepts.

Initial burner and ignition testing will be accomplished in a water-cooled, facility-type combustor prior to operation with steam flow. Ignition testing will be an important part of the initial development testing, since an external combustion-wave ignition system is to be used. Significant variation in operating procedures, sequencing, and even hardware configuration will be evaluated during the ignition test series. These tests will be conducted with the burner injector mounted in a water-cooled combustion chamber. Burner operation and performance also will be evaluated in this same hardware without steam flow.

Performance as a steam reheater or superheater will be evaluated after the basic burner characteristics have been evaluated. The burner assembly will be mounted in a burner-mixer assembly in a throughflow duct section. A source of superheated steam will be provided, flowing through this duct and surrounding the burner assembly. The ignition characteristics of the burner submerged in this steam flow will be evaluated. This testing probably can be accomplished with lower-than-scale steam flowrate, and still simulate the steam conditions in the burner area under both ignition and steady-state operation.

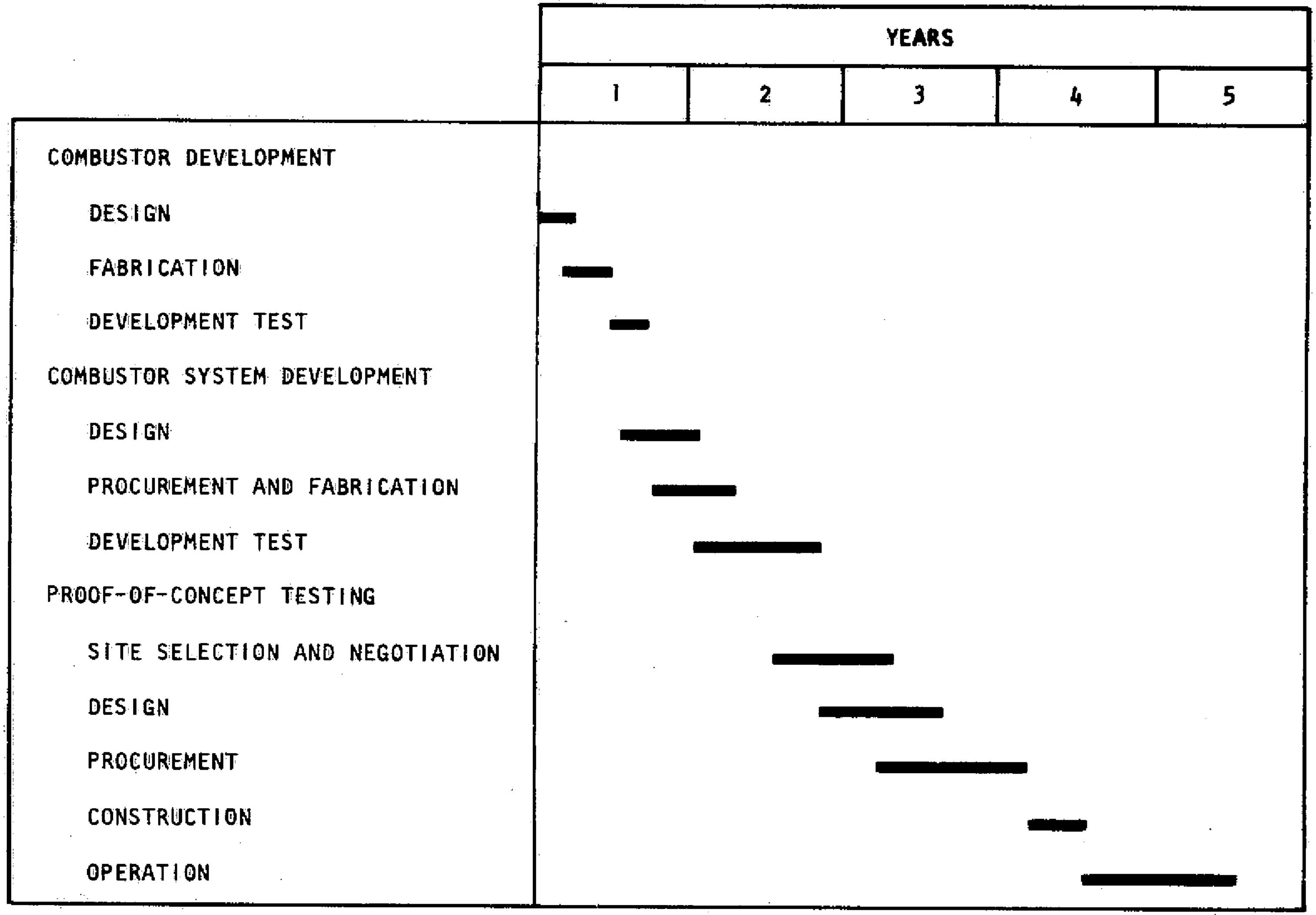

Figure 52. R&D Plan for Supplementary Steam Generation

SYSTEM DEVELOPMENT

Based on the results of the combustor component testing, a complete combustor system, including all the necessary controls, will be designed and tested.

Operation as a supplementary steam generator can be simulated with upstream and downstream water spray in addition to the throughflow of steam. Downstream gas properties will be monitored to determine the quantities of gaseous reactants remaining in the stream. This will permit the determination of any hazard of buildup of combustable mixture in the steam flow. Some gaseous products can be handled in the supplementary steam ejectors frequently used on steam condensers to remove noncondensibles.

Operation of this hardware will require a facility with full-scale capability. In addition to reactant (H_2 and O_2) flowrates, the full-scale operation requires a large steam flowrate. A hyperflow type of steam generator is one approach to this testing with minimum facility requirements. This approach would limit test durations, and would produce steam with a high proportion of noncondensibles.

The system hardware will be provided with extensive disassembly and instrumentation provisions in keeping with the developmental nature of the proposed testing. Initial burner and ignition evaluation will be conducted in a water-cooled combustion chamber similar to the earlier component testing. This will permit the evaluation of the basic burner characteristics without the expense of steam generation.

Ignition and burner operation will also be evaluated with the steam flow to resolve any quenching or blowout problems that might be encountered. Also, the ability to throttle the burner rate will be experimentally determined. An upstream spray de-superheater will be utilized to demonstrate operation with upstream "cold" mass addition.

The combustor system will be tested throughout the range of operation expected from a powerplant. This will be done to determine system performance and to verify proper operation of the control system. Some testing will be done beyond the expected plant range to assess operating limits and as a means of identifying potential problems early.

PROOF-OF-CONCEPT TESTING

The proof-of-concept effort will start with the identification of a suitable existing power plant for installation of a pilot plant supplemental steam generation system. A study will be conducted to evaluate the potential of such an installation and to determine the operating conditions and size of apparatus required. The basic installation will be designed and the equipment optimized for the specific application.

With this background design complete, the supplementary steam generation combustor is designed for eventual installation in the pilot plant. The equipment will be checked out in the same facility setup as the full-scale development combustor. Ignition characteristics, performance, and operational procedures will be evaluated in the Rocketdyne Test Facility.

This checkout will be followed by installation at the selected plant test site, and building the supplementary steam equipment and associated support equipment into the existing steam powerplant. Initial checkout testing will be accomplished to leak test and otherwise prove the integrity of the installation.

The initial checkout will be followed by a test period with steam flow from the existing boiler system for additional verification of the integrity of the installation.

First hot-firing tests will be conducted under engineering supervision for performance evaluation and development of operating procedures. The normal powerplant staff will be familiarized with the operation of the supplementary steam equipment during this period of engineering test, so that the system could be turned over to the regular operating personnel at the end of the engineering test operation period.

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