Master

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ABSTRACT

This report presents the results of Phase I of a multi-phase program leading to the development of a High Reliability Gas Turbine. The engine would be used for the base load, combined cycle application starting in the mid-1980's. The design, with primary emphasis on reliability, recognizes the need on the part of the electric utility user for high operating availability together with low maintenance cost to take full advantage of the capital cost and efficiency advantage offered by the combined cycle plant.

Phase I was a conceptual design study of both the gas turbine itself and the full plant. Reliability for all component equipment was emphasized from the very outset.

In the case of the gas turbine, high reliability was reflected in a concept that features ruggedness and simplicity. Ease of maintenance was provided through structural simplicity and modularization.

Component designs for the gas turbine were selected on the basis of a review of past operational experience and an in-depth statistical reliability analysis. Design details were then determined using trade-off and parametric studies.

Reliability in the auxiliary and balance of plant waste heat recovery equipment was provided by redundancy of critical components as well as the use of proven equipment.

Careful attention was paid to developing a concept for a highly reliable control system that blends automatic operation with manual override for operator intervention. A condition monitoring system to provide alerts for incipient shut down conditions was also incorporated into the control design concept.

The final gas turbine design is a 107 MW, single-shaft engine. Projected availability for the final engine installed in a combined cycle plant is 92 percent. This design will serve as a firm point of departure for the preliminary design to follow during the Phase II program.

EPRI PERSPECTIVE

PROJECT DESCRIPTION

This final report of Phase I of RP1187, High-Reliability Gas Turbine Combined-Cycle Development, presents the results of a study conducted by United Technologies Corporation, one of three contractors that conducted studies in Phase I. The other two contractors were General Electric Company and Westinghouse Electric Corporation. The basic study was to determine the potential reliability of a combined-cycle plant using a "new centerline" design approach with the main design parameter being reliability rather than performance.

The project was outlined in seven phases from conceptual design to engine endurance testing. The specific phases are:

- Phase I: Conceptual designs--to produce innovative ideas around a new centerline design for equipment that employs the latest state-of-the-art technical data in a total plant

- Phase II: Preliminary design--to investigate in greater depth the more promising configurations uncovered in Phase I and to determine feasibility and comparative values in a closer look at system concepts

- Phase III: After a review and selective process has developed one or two models as the most beneficial plant configurations--to produce detailed designs of the plant components requiring full development and integration

- Phase IV: The fabrication activity--to manufacture, as needed, the components selected for the technology verification project

- Phase V: Test of technology verification model--consists of performance and technical test of breadboard-type models at the manufacturer's plant or special test sites

- Phase VI: The fabrication, delivery, and installation of the prototype plant--in preparation for the field test (Phase VII)

- Phase VII: Test of the prototype plant model at a host utility site for sufficient duration to establish the inherent reliability of the total system (total running approaching 10,000 hours)

Only Phase I and Phase II have been approved by EPRI. The design change efforts were to be mainly on the combustion-turbine system. Redundancy, where feasible, was to be used for equipment other than the combustion-turbine units to meet the goals. The three contractors took varying positions regarding the new centerline approach. In the case of Westinghouse, they based their work on a new redesign of their present market offerings of the 251 and 501 models rather than a complete new centerline design. The 501 model is near the 100-MW-size industrial engine that was the nominal size and type asked for by the EPRI request for proposal.

PROJECT OBJECTIVES

The major objective of the study was to determine through conceptual design methods the maximum feasible potential reliability of a gas turbine and a combined cycle for coal-derived fuels, both gas and liquid. A secondary objective was to define the research projects and technical data necessary to improve the present market offering of gas turbines to meet the goals as specified by EPRI. A third objective was to obtain the reliability performance of present-day installations of combined cycles.

The main goals at the outset were specified by EPRI and had the following values:

Availability	
Gas turbine unit	95%
Total plant	90%
Starting reliability	99%
Mean time between shutdowns	9000 hours
Maintenance costs	
Gas turbine unit	1 mill/kWh
Total plant	2 mills/kWh

PROJECT RESULTS

The results of the United Technologies study indicated that the availability goals could be met at all levels of equipment, from the combustion-turbine unit to the total plant, but that mean-time-between-failures (MTBF) goals could only partially be met. The combustion-turbine unit alone could approximately meet the MTBF goal (9000 hours), while the combustion-turbine system could only reach an MTBF of about 1000 hours. When the steam cycle and its electric generator, associated equipment, and balance-of-plant (BOP) items were added, the total plant MTBF was predicted to be no greater than roughly 500 hours.

The Phase II work will concentrate on improving this value as much as is practicable. It certainly will be necessary to be redundant in some of the components in the steam cycle and BOP. Further, in some cases BOP component upgrading or component improvements may be necessary.

Overall plant improvements could possibly call for technology support projects in order to meet the Phase II goals. The three contractors compiled R&D programs in Task 4 to define these technology support projects. However, it was not possible to delineate the specific projects by their potential contribution to the reliability improvements. For this reason the projects were outlined in fairly general and broad categories. In Phase II more direct and specific projects will be pursued, with an attempt to rank them by their degree of contribution to reliability.

The field-data analyses indicated that the number of forced outages in the combustion-turbine system were caused mainly by the ancillary equipment, while the major contributor to the number of forced outage hours was the combustion turbine, although it contributed to less than 10% of the number of forced shutdowns.

At the start of the program it was hoped that Task 1, field-data analysis, and Task 2, reliability trade-off analysis, would result in a clear picture of the described changes and their contribution to improved reliability. However, the field data were not specific enough, and the analytic effort contained sufficient compromises, so that a sharp list of reliability-improving changes was not feasible. It was feasible to determine some beneficial design parameters and to arrive at a design with a significant improvement in reliability. A good example is the trade of reliability with temperature--both firing temperature and "hot spot" metal temperature. This is discussed in the report.

Analysis shows that the cost of electricity continues to lower as rotor inlet temperature is raised, but this higher temperature causes more unscheduled maintenance. The compromise solution was to use a lower metal temperature (obtained by an improved cooling scheme) of 1450°F and restrict the rotor inlet temperature to 2150°F. These values permitted meeting the goal for major maintenance intervals of 18,000 hours. Other improvements, such as the above, led to a projected MTBF for the new centerline design of 9000 hours for the combustion turbine. This value was the EPRI goal for Phase I.

Phase II will require very careful and skillful handling of plant details to improve the overall plant MTBF above 1000 hours. In the final analysis it may require raising the combustion-turbine goals in order to improve the overall plant MTBF to more nearly meet the Phase II goal of 3000 hours. Risk and failure modes and effects analyses will also be required in Phase II to give better visibility to the design results obtained.

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SECTION 1.0

SUMMARY

INTRODUCTION

United Technologies' Power Systems Division has completed Phase I of the High Reliability Gas Turbine Combined Cycle Power Plant Program. The objective of this program is to define the most practical gas turbine combined cycle power plant for intermediate and base load electric utility service.

An improved gas turbine engine must be defined, for it is the gas turbine which at present limits the combined cycle from achieving the reliability and durability requirements of a base load utility generator. For the gas turbine to find wide-spread use in intermediate and base load service, its present reliability, durability and maintainability deficiencies must be corrected, i.e., the gas turbine and its associated ancillaries must achieve higher availability.

As a result of our Phase I program, we have high confidence that the engine and system configurations presented herein will meet the availability and performance levels which utilities will demand for such units.

REQUIREMENTS

Combined cycle gas turbine power plants provide a flexible power generating option for intermediate and base load service. They have been identified as being one of the most economically attractive methods for utilizing coal in an environmentally acceptable manner. Utility experience with gas turbines to date has been primarily in peaking service. Deficiencies in starting and operating reliability and durability which utilities have experienced have been due to this type of cyclic service as well as to problems with the equipment itself.

Identifying and correcting these problems is the major requirement of this program. But future machines must meet new requirements as well. The world wide petroleum crisis dictates that reliable and durable operation be achieved on a

much broader range of gaseous and liquid fuels than in the past. New environmental standards dictate that these difficult fuels be used while achieving substantially lower emissions.

APPROACH

We structured the Phase I conceptual design study process to emphasize high reliability from the very outset. We also aimed to set the stage for the more refined preliminary design to take place in the subsequent Phase II follow-on. The Phase I program featured:

New Centerline Design Based on Available Technology

We started with an all-new design tailored especially for the base load combined cycle application. Starting fresh from the new centerline allowed us the freedom to correct reliability deficiencies intrinsic to the design concept itself. The new centerline approach also facilitated incorporating recently emerging gas turbine technology. But we have adopted only those proven technology advances which either directly enhance reliability or are dictated by future fuel flexibility requirements.

Current Product Reliability Analysis and Confirmation of Prediction Methods

Before beginning any actual conceptual design work, a reliability analysis was completed from both retrospective and prospective viewpoints. We assessed the field experience of our current product, the FT4. Since the FT4 has been employed principally in the peaking application, we expanded the analysis to use United Technologies' broad gas turbine experience data base in conjunction with established reliability projection methods. The expanded data base allowed us to assess plant availability for an initial base of reference and to substantiate the analytical methods we used for reliability predictions.

The base of reference was a necessary prerequisite for guiding the conceptual design. It was used both for establishing quantitative goals to be used to measure accomplishment of program objectives, and for providing trade-off factors in selecting alternative approaches.

Verification of our prediction methods was necessary to give confidence in the results of the reliability analysis we conducted to project future failure

rates. It was necessary to employ such projection methods for a design that is all new, although predicated on known, available gas turbine design technology. The resulting projections, when compared with program reliability/availability goals, were used to measure the level of success of the new design concept.

Later in the program, we visited four utility users of the FT4 to survey their operating experience in depth and their suggestions for a new centerline design. This survey provided an added dimension to the understanding of the true reliability problem.

Sequential Sifting Process to Evolve a New Centerline Gas Turbine Conceptual Design

Having established a base of reference from our current product experience, we conducted broad trade-off studies as a first step. This step was followed by aerodynamic parametrics to select promising overall compressor, turbine and combustor configurations. Utility cost and reliability factors were used to measure relative merit in the trade-off and parametric studies, Then, as a final step, mechanical studies of five different candidate concepts evaluated mechanical design, structural and maintenance considerations. From these candidates a final configuration was chosen, with reliability used as the foremost selection criteria, along with practicality, marketability, and competitive operating economics.

Emphasis on High Reliability in a Total Plant Concept

The Phase I program recognized that from the utility operator's viewpoint high availability entails reliability in the context of the complete powerplant with all its auxiliary and ancillary equipment. Drawing upon current FT4 combined cycle design experience, we used subcontractors who manufacture the major balance-of-plant equipment to help us develop the conceptual design of a complete high reliability combined cycle plant. We used the services of an architectural and engineering consultant (C. T. Main) to develop the overall plant layout. Selection was made of plant controls, heat recovery boilers, generator and electrical equipment, steam turbine, and auxiliary and gas turbine ancillary equipment as well. We evaluated starting systems to achieve satisfactory starting reliability. We conducted detailed computer analyses on the integrated plant using proven statistical reliability methods. Reliability analysis was extended by employing the "Monte Carlo" technique.

We based our analytical results on documented failure rate data for each item of plant equipment. We compared our projected results with program goals. The analysis identified critical components and helped identify spares and maintenance strategies which can maximize overall plant availability at minimum cost.

Subsidiary Tasks that Lay the Groundwork for Phase II Preliminary Design

Completion of the Phase I gas turbine conceptual design represents just the first step towards design of a high reliability combined cycle plant. Work was also undertaken during Phase I to pave the way for subsequent program steps. We prepared design criteria documents for both the gas turbine itself and its ancillaries. We laid out necessary technology R&D programs and investigated possibilities for technology retrofit; we developed a control system concept; we evaluated the ultimate impact on design of alternate service (peaking) and alternate fuels to include coal-derived liquids and gases; and we developed a condition monitoring strategy using the Hamilton Standard TRENDS^{R} system as a basis.

RESULTS

High Reliability Gas Turbine Design

Description - The high reliability gas turbine design is shown in Figure 1-1.

The final selected new centerline gas turbine design concept features:

- Single-shaft, front-end drive configuration with minimum numbers of compressor and turbine stages to enhance both cost and reliability.

- Simple, highly maintainable modularized structure with double-wall cases and rotor support using only front and rear bearings. The two-bearing configuration means not only high reliability in this potential problem area, but also easy access for maintenance as well.

- Firing temperature level (2,150^{o}F) which can provide combined cycle performance without sacrificing reliability. But metal temperatures are kept low through an advanced, but proven convective turbine airfoil

cooling method. The firing temperature will be higher than the current FT4, while average metal temperatures are kept much lower.

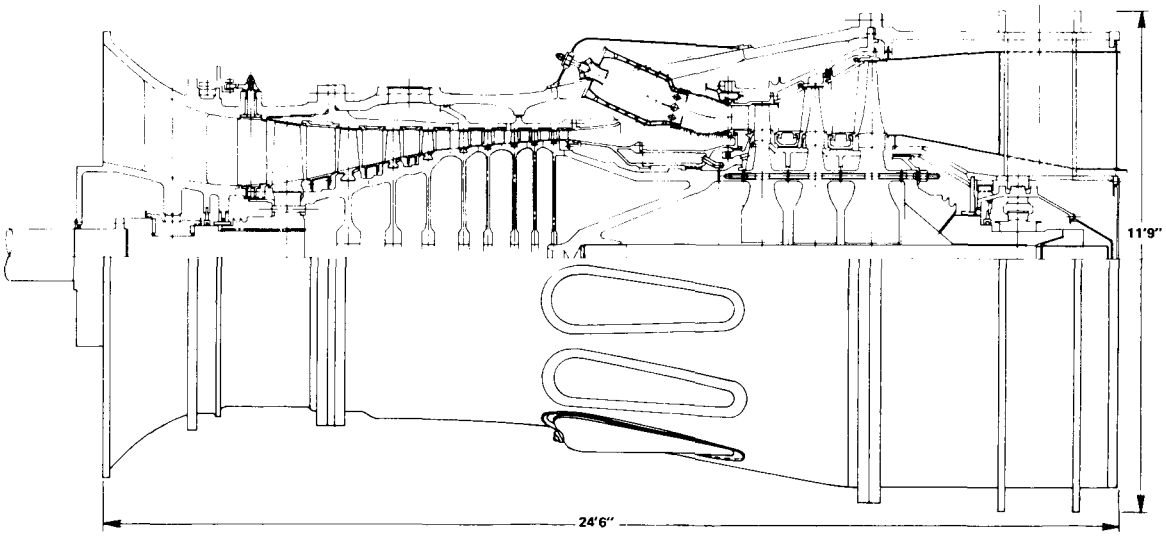


Figure 1-1 Cross-Section Layout of HRGT Conceptual Design

- Low emissions (including Dry NOx) Rich-Burn-Quick-Quench combustor concept which can handle a wide variety of fuels, including coal-derived liquids with substantial fuel-bound nitrogen content.

Performance

The gas turbine size and design cycle were selected to balance reliability and operating economics:

Power Output	107 MW (ISO)
Simple Cycle Heat Rate	10,700 BTU/KW-HR (HHV)
Pressure Ratio	14:1
Firing Temperature	2150°F

FT4 Field Experience and Verification of Reliability Methodology

The current FT4 gas generator (GG4C) has an overall failure rate (meantime between unscheduled maintenance action) of just under 5400 hours. The breakdown analysis of causes is shown in Figure 1-2. Also, broader insights into controls and ancillary operating experience was gained from personal interviews with and record surveys of four selected utility operators. This experience was incorporated into the final HRGT plant design.

The reliability projection methodology was substantiated by using the complete United Technologies' component failure rate data base and prediction system for new designs. This system and data base were used to establish an initial base of reference. The Single-Shaft-Base-Load reliability assessment was used to provide targets for the final HRGT plant design, as shown in Figure 1-3. Projected meantime between failure (unscheduled maintenance action) for the final selected HRGT design concept turned out to be 8264 hours, some 50 percent improvement over the current FT4.

Our reliability analysis of the complete powerplant, which used the baseline reference conceptual design, apportioned outages by major plant components. The results, summarized in Figures 1-4 and 1-5, appear to confirm what might be expected from operating experience with current gas turbine and balance-of-plant equipment. Plant availability is predominantly determined by outages associated with the gas turbine itself, but the unscheduled outage rate can be significantly influenced by minor, short duration shutdowns attributable to the gas turbine ancillary and steam heat recovery equipment (including sensors and controls).

UMA/10^6 EOH		PERCENT
50.00	INLET DUCT, CASE ASSEMBLIES AND BEARING SUPPORT STRUCTURES	26.9
42.84	MAIN ENGINE BEARINGS AND LUBE SYSTEM	23.1
35.70	COMPRESSOR COMPONENTS	19.2
32.14	UNKNOWN	17.3
10.71	TURBINE COMPONENTS	5.8
10.71	ACCESSORY DRIVE	5.8
3.57	COMBUSTOR	1.9
UMA RATE 185.67		100.0

MTBUMA = 5386 HOURS

DATA ARE FOR ALL FUELS – BASE, PEAK, MARINE, ELECTRICAL AND INDUSTRIAL APPLICATIONS

ACTUAL HOURS FOR ANY SPECIFIC APPLICATION CAN BE CONSIDERABLY DIFFERENT

TOTAL ENGINE OPERATING HOURS

GG4C-1D	193,000
GG4C-3F	87,000
	280,000

MTBUMA–MEANTIME BETWEEN UNSCHEDULED MAINTENANCE ACTION

Figure 1-2 GG4C Reliability

High Availability Through Maintenance Techniques and Hot Section Durability

Emphasis in the design was placed on minimizing total down time as well as unscheduled outage frequency.

Targets for scheduled maintenance intervals are 9,000 operating hours between minor maintenance and 18,000 hours between major maintenance actions. The 9,000 hours represents the time needed to repair and/or refurbish turbine airfoils, and is achieved through new technology convective cooling techniques, long-life corrosion-resistant coatings, and judicious selection of the rated firing temperature.

The engine's structural design emphasizes ease of maintenance and accessibility. Major components can readily be separated into individual modules.

Turbine airfoils can be directly removed and replaced with the engine in place.

Redundant components are used in the gas turbine ancillary system to prevent nuisance outages caused by the failure of a minor part.

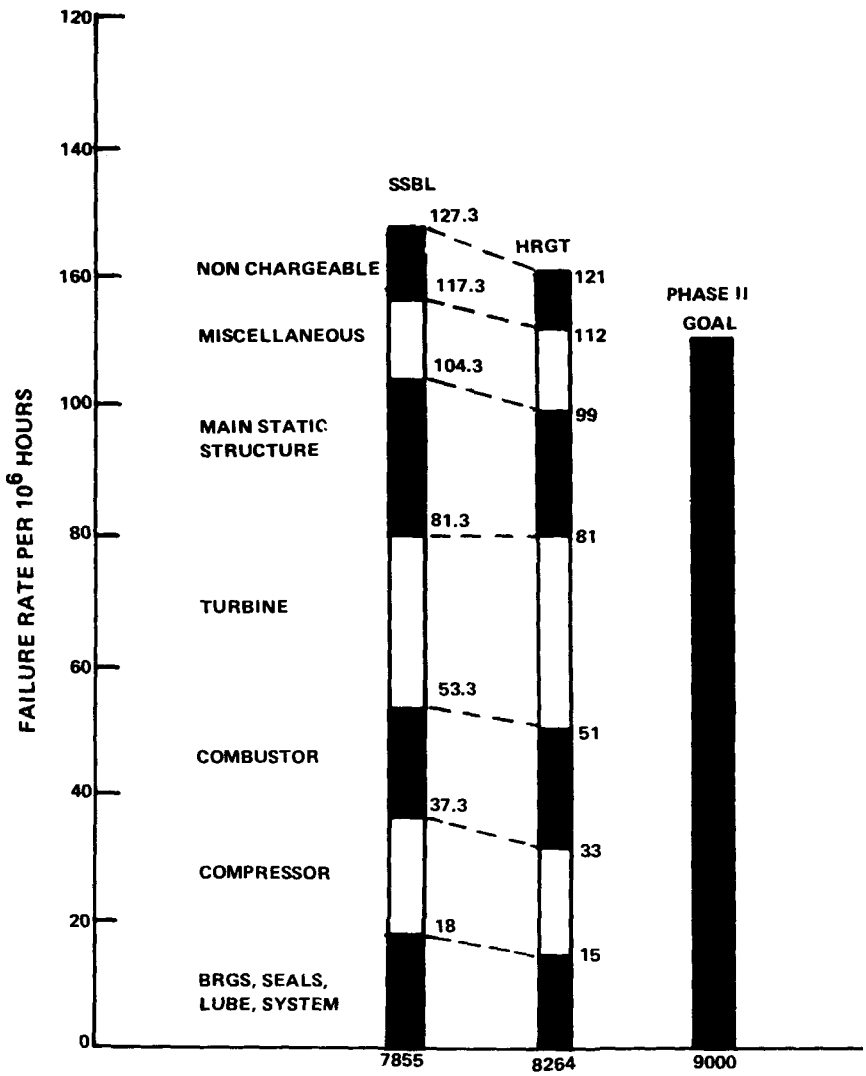


Figure 1-3 Analytical Reliability Results for Single-Shaft-Base-Load and HRGT Shown with Phase II Goal.

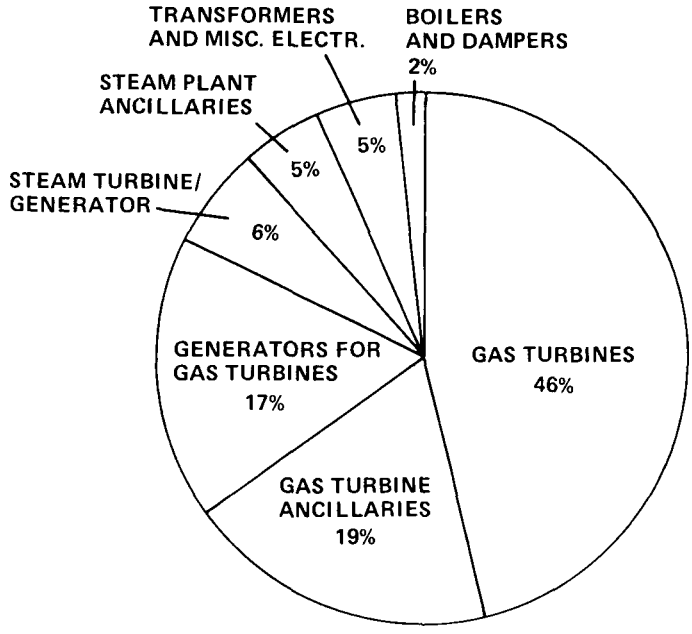


Figure 1-4 Contributions to Unavailability - The combustion turbine is the largest contributor, followed by the generators and ancillaries for the combustion turbine.

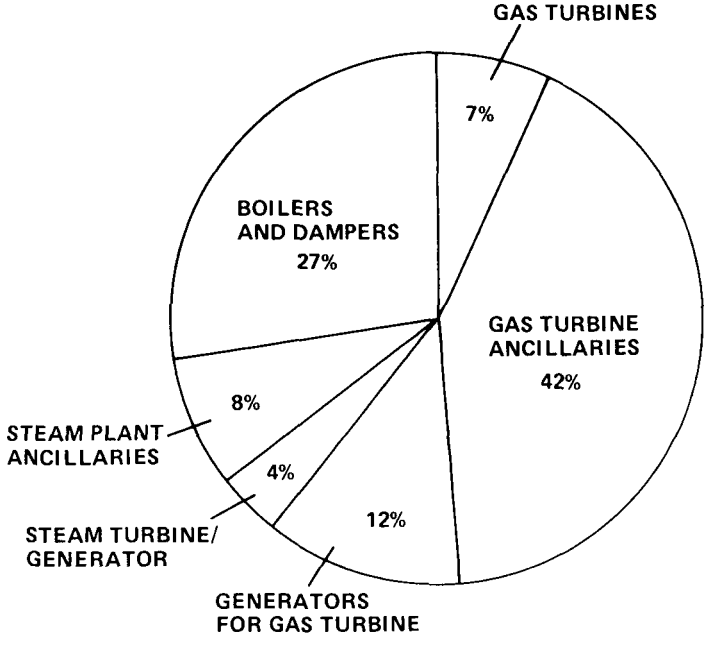


Figure 1-5 Component Contributors to Unscheduled Partial Shutdowns - Turbine ancillaries, boilers, and dampers are the major contributors.

High Reliability for the Total Powerplant

The overall power plant availability profile is shown in Figure 1-6 and results in an equivalent overall availability of 92 percent. The plant is shown in Figure 1-7.

Major features of the power plant are:

- A conventional three-pressure boiler system for maximum waste heat recovery. The high pressure boiler provides 300,000 lb/hr steam at 870 psig, 850°F conditions. With such modest steam conditions, natural convection can be employed to ensure high reliability.

- Bypass dampers to modulate turbine exhaust flow during system start-up and thereby minimize thermal transients in the steam system. Also, the bypass diverter valves provide extra flexibility by permitting continuous gas turbine operation when the steam system is temporarily shut down.

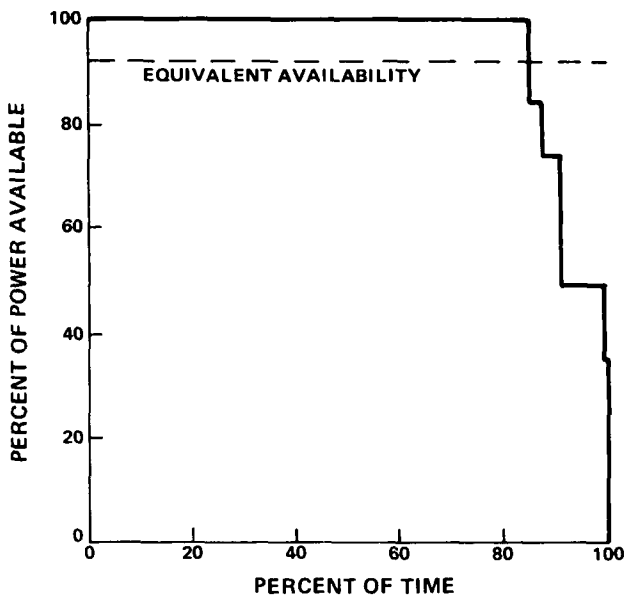


Figure 1-6 Availability Characteristics of Reliable Engine Combined Cycle Powerplant

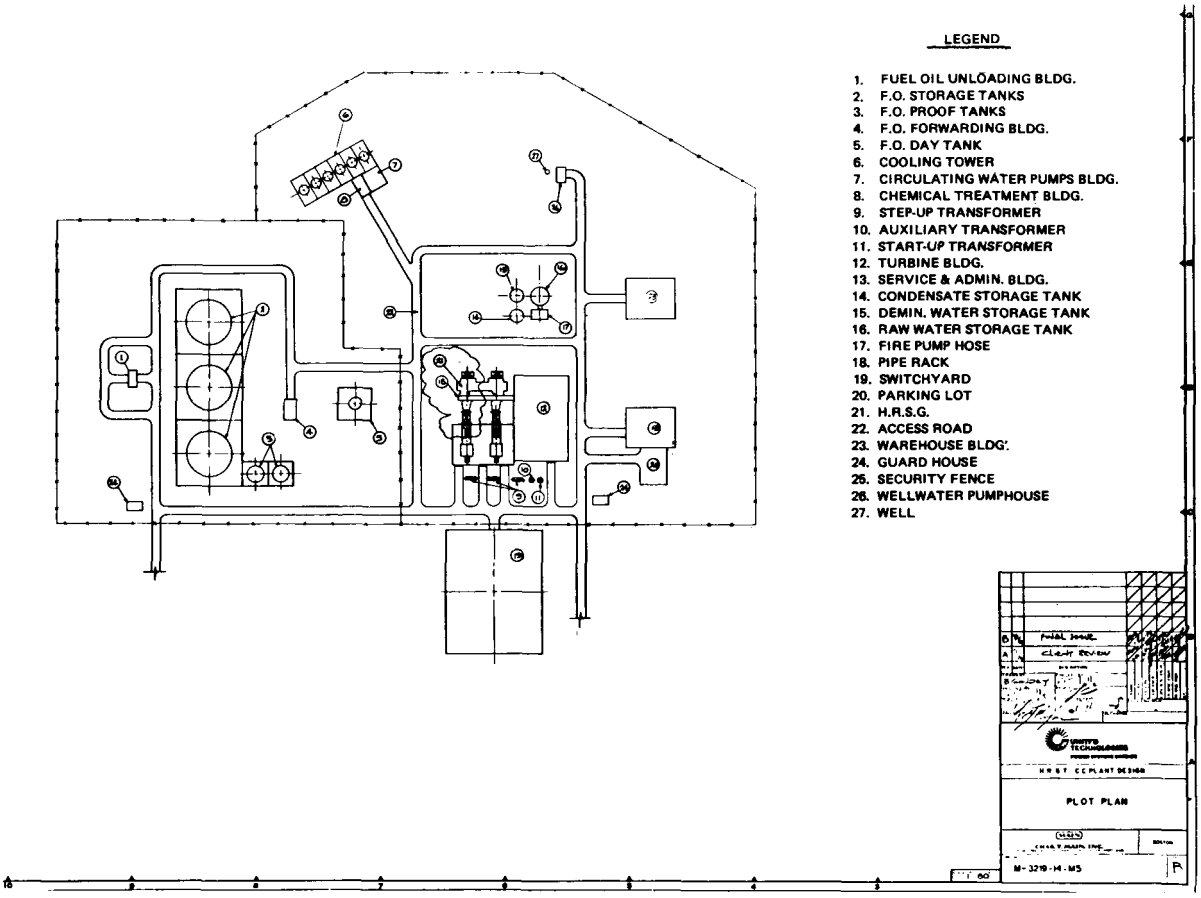


Figure 1-7 Site Arrangement of the HRGT Combined Cycle Powerplant

- An integrated control system which emphasizes reliability by using digital microelectronic supervisory controls with redundancy and in-place replacement of circuit boards. Manual override capability by the operator will avoid shutdowns due to spurious alarms. Manual operation is available as well for all major equipment components.

- Centralized turbomachinery building (Figure 1-8) that best uses maintenance space and equipment. To take full advantage of the gas turbines' modular maintenance concept, the plant layout is designed to service on-site maintenance with minimum down times. The adjacent aisleway space is sufficiently large to accommodate an engine and its maintenance equipment. Several ancillary equipment skids which service the turbomachinery are located in the building to minimize exposed plumbing and electrical conduits.

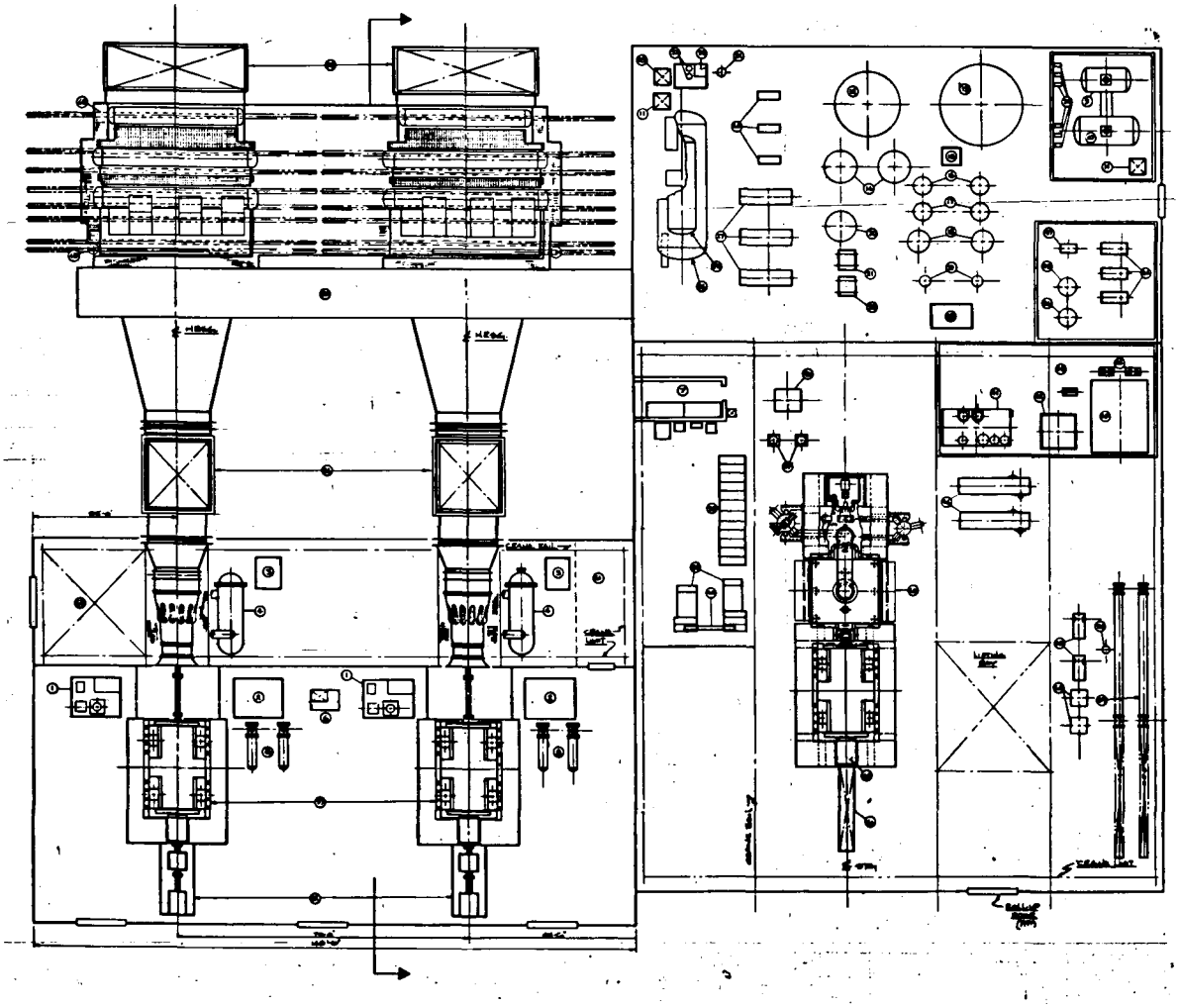


Figure 1-8 Gas Turbine Building - A prime design criteria was to provide facilities for easy maintenance of the system.

- A centrally located control room allows operating personnel to stay near the equipment for sensing equipment status and to minimize time devoted to manually correcting minor problems.

Starting reliability is projected to be 89 percent for the Phase I plant. Ninety-five percent could be achieved by refinements during Phase II.

Starting reliability is enhanced by a reliable control system with redundant sensor and manual override capability.

The best overall start system in terms of reliability and cost appears to be an ac electric motor drive .

RECOMMENDATIONS

To further the achievement of the High Reliability Gas Turbine Combined Cycle Development Program objectives, Phase II preliminary design can proceed with the Phase I conceptual design study results as the spring-board. Specifically, based on the Phase I study results, we recommend:

- Phase I results indicate the EPRI program goals are realistic and can be applied to the Phase II program.

- Gas turbine design refinements should carry our selected Phase I HRGT concept through more detailed design analysis. This would include: structural analysis of component parts - both static and dynamic-aerodynamic analysis of individual turbine and compressor blade rows; heat transfer and geometric definition of cooled turbine airfoils; design and off-design performance analysis.

- The integrated control system concept should be carried forward into more refined design and analysis to include a powerplant dynamic simulation.

- Further detailed analysis should be conducted on the TRENDS[R] condition monitoring system to further evaluate its impact on reliability.

- Design and selection of auxiliary, ancillary, and steam bottoming equipment should be reviewed and refined in more depth to achieve the best balance of capital and operating costs and operating availability.

- The gas turbine and ancillary design criteria documents have been prepared from the gas turbine manufacturer's perspective. EPRI, representing the utility operator's viewpoint, should review these documents and recommend revisions as appropriate to the needs of the utility baseload application.

- To achieve, ultimately, complete fuel flexibility, including coal-derived liquids, development of the Rich Burn Quick Quench (RBQQ) combustor technology should be fostered. The Phase II HRGT design work should, in the meantime, investigate as a back-up, a low-emissions combustor concept representing a more near term capability.

- Critical to the achievement of high reliability turbine airfoils is the development of our convectively cooled two-piece blade. This technology, together with a maximum of 2,150°F (combustor exit) firing temperature, should serve as basic elements of the HRGT design as the program continues into Phase II.

GAS TURBINE DESCRIPTION

The gas turbine engine design concept which we judge best meets the needs of the high reliability combined cycle development program is described below. This design results from work performed thus far during Phase I of the program; we propose to use it as the basis for more detailed design studies during Phase II, our goal being to arrive at a practical and feasible engine design which can set high standards of availability with a reasonable compromise of all other parameters.

Overall Characteristics

The key features of this baseload, high availability machine are:

- Modular Construction -- to maximize availability by reducing maintenance time; rapid replacement of components.

- Single Shaft Direct Drive -- Simplest configuration to minimize number of parts. Single speed operation reduces vibrating failure modes.

- Cold End Drive -- Simple, direct hot gas ducting to heat recovery boilers.

- Factory Assembled -- Simple, direct connection for reliable, rapid installation on site.

Compressor

The compressor uses nine stages of low aspect airfoils to provide a short, rugged unit with a minimum number of parts. Recently developed technology has made possible a compressor with this low number of stages. This technology has been selected because it provides the greatest potential for high reliability and durability.

- Stiff, thick airfoils -- these blades and vanes are highly resistent to erosion, foreign object damage, and vibration.

- Rapid access for maintenance -- horizontally split case is used to provide rapid access.

- No multiple stage variable geometry -- only a single variable inlet vane is needed to provide surge stability.

- Rugged welded drum rotor -- rotor shaft is short, stiff, and easily balanced for vibration-free operation.

Combustor

The combustor uses the Rich Burn-Quick Quench concept to meet emission requirements with coal-derived as well as petroleum-based fuels.

The Reliable Engine must incorporate new combustor technology to be able to meet fuels and emissions requirements. The United Technologies-developed Rich Burn-Quick Quench Combustor (EPA Contract No. 68-02-2136) is the only technology capable of meeting fuels/emissions requirements. The results of United Technologies-supported as well as Government-funded work has been used to define the most reliable design of this combustor which features:

- Readily accessible rich burn combustor cans and fuel nozzle assemblies.

- A common annular lean burn combustor and transition section to provide more uniform turbine inlet temperatures.

- Shrouded, regeneratively cooled rich burn combustor.

- o Adaptability for distillate, residual, and coal-derived gaseous and liquid fuels.

Turbine Section

The turbine uses existing technology high work aerodynamics to minimize the number of parts and provide a short, stiff rotor. Key features include:

- Only three stages required to provided satisfactory efficiency.
- A compact rotor directly coupled to the compressor.
- A double wall case for durability and reliable tip clearance control.
- Rapid maintenance -- front stage vanes and blades replaceable without engine disassembly; entire turbine module removable as a unit.

Turbine Airfoils

Metal temperatures are maintained several hundred degrees below those of present engines for long, corrosion resistant life. Airfoil cooling is accomplished without film cooling, i.e. there are no cooling holes susceptible to plugging by particles or deposits from dirty fuels.

- Airfoil metal temperatures are held below 1450°F hot and 1350°F average. This maximum temperature is approximately 200°F lower than in present engines.
- Airfoil cooling is provided by all-convective designs; no film cooling holes are used.
- Cooling air from the compressor is cooled external to the engine before flowing to the airfoils.
- Airfoils are constructed of conventional turbine alloys with hot corrosion resistant coatings applied.

Rotating Elements

The short, stiff compressor and turbine rotors allow use of a two-bearing rotor support system. A rugged, reliable rotor system results.

- Rotor critical vibration modes are far from operating speed.
- Journal bearings with high damping are used for long life.
- Bearing/seal compartments are located in the lowest temperature regions; directly accessible for maintenance.

Ancillaries

Gas turbine subsystems such as the fuel and lubrication supply systems, the ignition and starting systems, etc., have been a major source of system unavailability. Our approach is to solve by system design as many of the reliability/durability/maintainability problems possible short of modifying the design of the ancillary components themselves. We feel substantial improvements can be made by:

- System design guided by Failure Mode and Effect Analysis plus Fault Tree Analysis.
- Selection of approaches with high inherent reliability.
- On-line condition monitoring and diagnostics.
- Skid-packaged ancillaries with ease of access for inspection and replacement.

GAS TURBINE COMBINED CYCLE SYSTEM DESCRIPTION

Two gas turbine engines are used; the exhaust gases of each feed separate heat recovery steam generators. The energy thus recovered is extracted by a single steam turbine driven generator to provide a total plant output of 300 MW. The heat rate is 7700 Btu/kWh (HHV).

The more significant system design features incorporated to create the high availability combined cycle power plant are summarized below:

Generators

Both the gas turbine and steam turbine generators are hydrogen cooled, brushless excited units. A condition monitoring and diagnostic system is used to track the health of these units, (and for all other major components of the system) to allow maintenance to be performed on an as-required rather than a scheduled basis. The design incorporates:

- Lubrication system common with the turbines.
- Hydrogen cooling to provide a longer-life environment within the generator.
- Condition monitoring to sense stator temperature, vibration, gas conditions and show trends needed for corrective action.

Heat Recovery Steam Generators

A standard design, three-pressure-level boiler is used. The increased power and efficiency of this three-level design were judged to significantly outweigh the increase in complexity compared to single-pressure-level types. Industrial experience with similar units indicates the basic steam tubes and drums have satisfactory reliability and durability. Care must be taken in design of hot gas ducting and controls to achieve the reliability goals of this program. Steam conditions are 870 psig, 850°F, 300,000 lb/hour per boiler.

Key features are:

- Steam generator has natural convection water circulation for high reliability.
- Bypass dampers -- simple, field proven diverter valves are used to isolate a steam generator from its turbine to permit continued gas turbine operation if the steam system is out of service. The dampers modulate turbine exhaust gas flow during system starting to eliminate thermal transients in the steam system.

- Stack temperature of 300°F eliminates possibility of corrosive condensation.

Steam Turbine

A Toshiba International steam turbine with extensive field experience is used as the bottoming cycle prime mover. Seven of the units have been in service for durations from 2 to 20 years accumulating nearly 700,000 operating hours with only four forced outage incidents.

System Control

A single control center is used for the combined cycle power plant. Four important features have been included in the design to maximize availability:

- Integrated gas turbine, steam system, and ancillary controls.

- Condition monitoring and diagnostic prediction system integrated into the plant design.

- Digital microelectronics supervisory controls with redundancy, in-place replacement of circuit boards, and overide capability.

- Manual operation capability for all pieces of the system.

We base these design concepts on the experience we and our balance-of-plant subcontractors have had with gas turbine/steam turbine power systems and the results of the on-site survey of utility systems performed during Phase I of this program.

We recognize that the introduction of "automated", "microprocessor controlled", "fully integrated" equipment has some break-in problems, as usually associated with new equiment. However, in consideration of the economic performance of the plant, our control system recommendations can: 1) reduce the severity of operating conditions on the components; 2) maximize intervals between maintenance actions; and 3) minimize downtime for maintenance. In addition, provisions made for personnel to manually operate the system can further safeguard against unnecessary plant shutdown.

Turbomachinery Building

All turbomachinery is housed in a single building to centralize control of the plant and provide better utilization of maintenance space and equipment. Partitions are used to isolate generators from the turbines and to provide gas turbine noise enclosures. There are sufficient aisleway areas beside each machine to provide a convenient sheltered "lay down" space when the unit is being maintained.

The several ancillary equipment skids which service the turbomachinery are located in this building to minimize plumbing and electrical line lengths and exposures. A centrally located control room allows operating personnel to be close to the equipment for sensing of equipment status and to minimize time for manually correcting problems.

RECAPITULATION

A summary of the key reliability, availability, and performance parameters of the High Reliability Gas Turbine Combined Cycle Power Plant is shown in Table 1-1. These values are a result of judgements made in our Phase I program to arrive at the best balance of designs that lead to high reliability, durability, availability, and economically competitive performance.

TABLE 1-1. RELIABLE ENGINE COMBINED CYCLE SYSTEM CHARACTERISTICS PREDICTED FROM PHASE I

	Phase I Results	EPRI Phase II Goals
Output Power, MW	300	300
Heat Rate, Btu/KWH (HHV)	7700	7200
Availability (Equivalent) %	92	95
Reliability (Equivalent) %	96	98
Combustion Turbine Start Reliability %	95	95

The table shows our status at the end of Phase I. We have approached but have not yet met the 95 percent equivalent availability goal. We have identified specific improvements to be taken during Phase II to meet the goal. In the interest of maximizing conditions leading to high availability, we have selected conservative operating conditions. This has resulted in a heat rate of 7700

Btu/kWh vs. the EPRI goal of 7200 Btu/kWh. However, we judged that durability and reliability considerations override those of performance.

For example, we reduced turbine metal temperatures several hundred degrees from those of today's engines. We were able to do this without suffering unwarranted performance penalties or introducing unreliability due to untried turbine cooling technology by drawing on achievements in airfoil fabrication processes developed by and in use at United Technologies' aircraft gas turbine divisions. This results in a lower turbine inlet temperature and hence lower efficiency and power output than could be achieved if advanced turbine cooling technology now in the early stages of development were to be used.

This is but one example of the trades and judgements that will continue during Phase II. Our goal in Phase II remains the same as during Phase I:

> To produce a reasonable engine and system design relative to performance, first cost, and development program cost and schedule while striving for highest reliability, durability and maintainability.

SECTION 2.0

INTRODUCTION

The High Reliability Gas Turbine Combined Cycle (HRGT-CC) Powerplant Program consists of seven phases. The first phase, which is the subject of this report, involved examination of aerodynamic, cycle performance, maintenance, and structural considerations and led to a selected design concept for the gas turbine. At the same time, conceptual layout design of the entire combined cycle powerplant was undertaken. This included the selection of steam bottoming heat recovery equipment and auxiliary and accessory equipment as well.

The overall objective of the program is to achieve a highly reliable gas turbine combined cycle plant which would be an economically attractive base load generation alternative available to the electric utility industry by the mid-1980's.

OVERALL REQUIREMENTS

Besides reflecting a pre-eminent emphasis on high reliability, the HRGT-CC design should, in view of the future fuel availability uncertainty, also be capable of handling a wide variety of alternate fuels. This fuel flexibility must not come at the expense of low exhaust emissions. At the same time, to provide competitive power generation economics, the design must offer low fuel consumption, low capital cost, and low maintenance expense. Finally, the gas turbine must be a marketable product from the standpoint of the engine manufacturer.

In summary, then, we were guided by four interrelated measures to judge the reasonableness of candidate designs:

- Highest System Availability
- Lowest Practical Cost of Electricity
- Availability for Procurement in the mid-1980's
- Applicability to a Broad Market

High Availability

The most important characteristic sought was high overall plant availability. Means to achieve it were to design or select individual parts and components to be:

- Reliable - have low failure rates
- Durable - have low wear-out rates
- Maintainable - in the event of wear or failure be able to effect rapid replacement

Redundancy of ancillary subsystems was employed where practical to minimize downtime.

Condition monitoring can be used to identify incipient failures, allowing corrective maintenance outages to be set by conditions, not by schedule.

Lowest Cost of Electricity

Achieving the lowest practical cost of electricity was sought by:

- Choosing the thermodynamic cycle that would accommodate existing technology and also permit growth when satisfactory performance and reliability of a new technology is demonstrated.

- Choosing components that would provide highest efficiency consistent with reliability and durability goals.

- Choosing component and system designs which have reasonable first cost and can be manufactured using readily available materials.

Availability for Mid-1980's Procurement

It is important that near term capability be achieved for two reasons:

- This new breed of engine needs to begin providing economic benefits to the utility industry as early as possible and, simultaneously, to begin building the operating confidence needed to justify large investments in base load coal gasification combined cycle plants.

- Engine manufacturers need the revenues from early sales to help generate funds necessary to complete the development and to justify the total program.

The time constraint has meant that component design selection was based on using either existing technology or that technology which has demonstrated the necessary maturity. No technological innovations were used other than those necessary to meet reliability and durability objectives with one exception: fuels and emissions requirements necessitated that new combustor technology, now being developed under parallel programs, be used.

Market Applicability

The High Reliability Gas Turbine Engine needs to have broad market applicability and user acceptance in order to:

- Find broadest U. S. utility market acceptance - peaking, intermediate, and base load.

- Find broad non-utility service acceptance in world-wide markets.

Such capability is obviously needed to justify the large scale investment to be made in this program by the utilities, through EPRI, and by the gas turbine engine manufacturer.

PROGRAM GOALS AND SPECIFIC REQUIREMENTS

At the program outset, EPRI specified the goals shown in Table 2-1

SUMMARY

This report covers all contractual work conducted during Phase I. Some of the work directly involved conceptual design. Other tasks contributed but were subsidiary to the actual design.

The material presented in this report is organized under nine sections plus appendices. The sections of this report consist of:

TABLE 2-1

HIGH RELIABILITY GAS TURBINE COMBINED CYCLE PROGRAM GOALS

Reliability:	
Gas Turbine	
Availability (FOH + SOH)	95% or better
Maintenance Cost (Mills/kw-hr)	1
Total System	
Availability	90% or better
Maintenance Cost (Mills/kw-hr)	2
Durability:	
Operating hours between shutdowns	9,000
Hours between major maintenance	13,000
Hot section parts life, hours	50,000
Cold section parts life, hours	100,000
Other:	
Starting reliability - 99%	
Fuel flexibility capability for light and heavy oil	
Turbine inlet temperature - 2300°F maximum	
Heat rate for the combined cycle - 7500 Btu/kWh	
Heat rate for single cycle - 9500 Btu/kWh	
Dry NO_x capability to meet the new source standards (expected to be 75 ppm)	
Overspeed requirements - 120%	
Load following response - ±25%	
Low cycle fatigue - 12,000 cycles	
High cycle fatigue - 10% cycles	
Power shaft rpm - 3600	
Nominal gas turbine size - 60-100 Mw	

Note: These EPRI specified goals and requirements acted to provide focus for the new centerline conceptual design effort undertaken during Phase I.

SECTION 1 - SUMMARY

SECTION 2 - INTRODUCTION

SECTION 3 - NEW CENTERLINE CONCEPTUAL DESIGN DESCRIPTION

This section describes the final High Reliability Gas Turbine Combined Cycle (HRGT-CC) design concept which represented the end product of the total Phase I studies. Sub-sections describe, in turn, configuration and salient features of: the gas turbine engine; the gas turbine ancillaries and start system; the balance-of-plant equipment, including

the steam bottoming system, building and site, auxiliaries, and plant control system.

SECTION 4 - CURRENT PRODUCT RELIABILITY ASSESSMENT

This section presents the results of an evaluation of recent FT4 field experience and the establishment of a reference powerplant. The FT4 experience was used as an initial point of departure to assess endemic past problems to be avoided.

To bridge the gap between the present day peaking application and future combined cycle base load duty, a broader reliability assessment was made using an established reliability analysis technique in conjunction with United Technologies' broad base of gas turbine operating experience.

To serve as a reference, a complete base load plant was developed around an existing modern FT4 combined cycle installation (Ulsan, Korea).

SECTION 5 - RESULTS OF STUDIES LEADING TO THE DEVELOPMENT OF GAS TURBINE DESIGN CONCEPT

In this section, results of all the background conceptual design studies are summarized. This includes: broad trade-off studies; aerodynamic parametrics; and evaluation of candidate configurations. This material explains the rationale which led us to the final selected design.

SECTION 6 - COMBINED CYCLE POWERPLANT RELIABILITY ANALYSIS

This section is devoted to covering the reliability analysis used to develop the powerplant design concept as measured against the original program goals for overall availability. Contributions for the basic plant elements, including the gas turbine and its ancillaries, are evaluated.

SECTION 7 - GUIDANCE FOR PHASE II PRELIMINARY DESIGN

The material in this section includes a technology assessment for the gas turbine, a summary of combustor fuel flexibility considerations (to include coal-derived fuels), and a specification which details the gas turbine control design concept.

SECTION 8 - PRIOR SSBL DESIGN BACKGROUND

This section summarizes prior design study results which were pertinent to the Phase I HRGT design and were provided by United Technologies as a program cost share. The studies culminated in the Single-Shaft-Base-Load design concept. Historical background design studies are presented in some depth for reporting completeness.

SECTION 9 - SUPPORTING DOCUMENTS

The three supporting documents, the Gas Turbine Design Criteria, Gas Turbine Ancillary Equipment Design Criteria, and the Reliability Plan, are contained in this section. These were generated as part of Phase I to serve as background for the Phase II preliminary design.

APPENDICES

This section contains the following appendices:

A Cost of Electricity Calculation Methods

B Model to Evaluate Availability

C. Monte Carlo Analysis Procedure, Results, and Simulation Computer Program

D. HRGT Combined Cycle Plant Reliability Estimate

SECTION 3.0

NEW CENTERLINE CONCEPTUAL DESIGN DESCRIPTION

GAS TURBINE ENGINE

Overall Engine Description

The Highly Reliable Gas Turbine (HRGT) engine which evolved from the studies described in the foregoing paragraphs is specifically aimed at being used for the base load, combined cycle application, beginning service in the mid-1980's. The basic features embodied in the design can be highlighted as follows:

- Single-shaft, front-end drive rotor.
- Compact compressor and turbine for mechanical simplicity with a minimum of parts.
- Combustor with complete fuel flexibility, to include coal-derived liquids and gases.
- Front and rear journal bearing rotor support with stiff frame and split cases for ease of access and removal.
- Modularized maintenance construction for minimizing outage downtime.
- Low metal temperature, turbine airfoils with corrosive resistant coatings and all-convective internal cooling.

Figure 3-1 is an overall cross-sectional layout of the HRGT conceptual design. Specific design features for each major component are discussed in the following paragraphs.

The compressor produces a pressure ratio of 14:1 in nine stages. It has a constant outer diameter and a first-stage tip speed of 1,150 ft/sec.

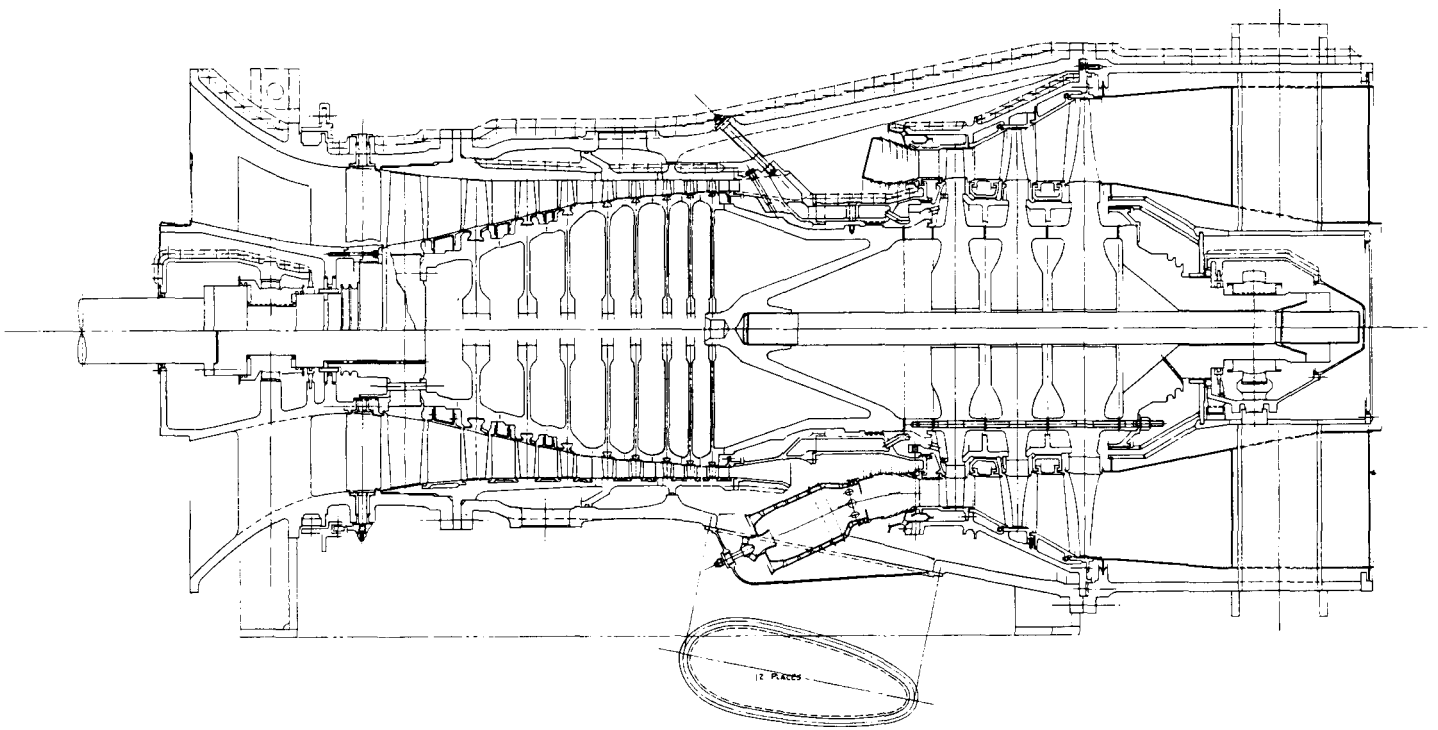


Figure 3-1 Layout of High Reliability Gas Turbine

The combustor system is specifically designed to provide low emissions operation for a wide variety of fuels. The configuration is comprised of twelve individual rich burn cans which feed an annular lean burn and mixing section. The connection point between cans and the annular portion serves as the quick quench zone where large quantities of air are introduced through a series of holes. Primary air is used entirely for rich zone combustion and enters the can through a swirler around the fuel nozzle that is part of the cast can centerbody. After the quick quench, lean burning takes place in the forward section of the transition duct. Beyond the lean burn zone, this assembly blends from the can configuration into a full annulus so as to uniformly feed combustion products to the turbine.

From aerodynamic parametric studies, a three-stage turbine design was evolved which could provide design simplicity and compactness for high reliability as well as high efficiency for good performance. The chosen turbine design emphasizes durability and reliability for the critical hot section airfoils and a high maintainability structural arrangement.

Structural and Mechanical Features

The structural and mechanical features incorporated in the final engine have been selected to complement the selected aerodynamic features and result in initial cost attendant with a maintainable product.

The engine static case philosophy is set by the desire to achieve as much "in situ" modularity as is possible while at the same time to allow fully assembled rotor balancing. The modularity concept markedly reduces the time necessary to remove a major engine section in a horizontal position on site and again return to engine to service. It does, however, imply an adequate supply of spare modules to minimize turnaround time.

External Cases. A double wall external case construction has been selected because of its contribution to reliability. It consists of an inner wall, which supports the vanes and contains the hot turbine gases, and an outer wall, which contains the gas pressure while being bathed with cooler compressor discharge air.

A typical case construction is shown in Figure 3-2 The cooler outer case makes the structure much less prone to thermal stress and distortion. In addition, the

construction allows good blade tip clearance control, which allows tight clearances without risk of blade rubbing. This capability for tight tip clearance also reduces the requirement for blade tip shrouds, which in turn would cause high blade root stresses.

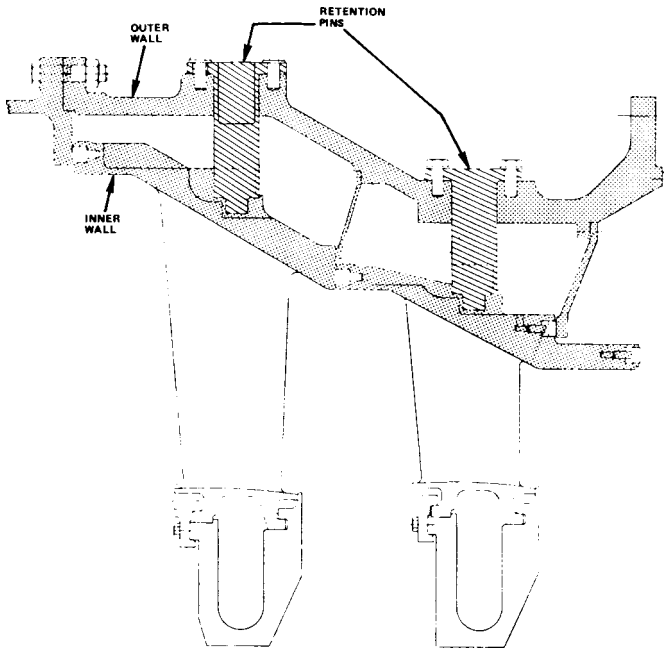


Figure 3-2 Double-Wall Case Design - The double wall structure eliminates shrouded rotor blades for increased reliability.

Another feature is the use of cylindrical end walls over the blade tips, as shown in Figure 3-3. In this design, axial motions of the rotor will not cause blade tip clearances to change as they would with a conical end wall. This feature contributes to both reliability and performance.

All engine cases are of cast material with axial split bolt planes at the horizontal engine plane. In addition, circumferential bolt planes are located near the front of the compressor and at the rear of the turbine. The variable inlet guide vanes are purposely included in the front case assembly to both facilitate maintenance and to provide flexibility in the event that future growth versions require the addition of a front "supercharging" stage.

A mid-compressor bleed port discharges from a collector plenum made up by an area between the inner and outer cases. The bleed control mechanism is mounted external to the engine where it is easily maintained.

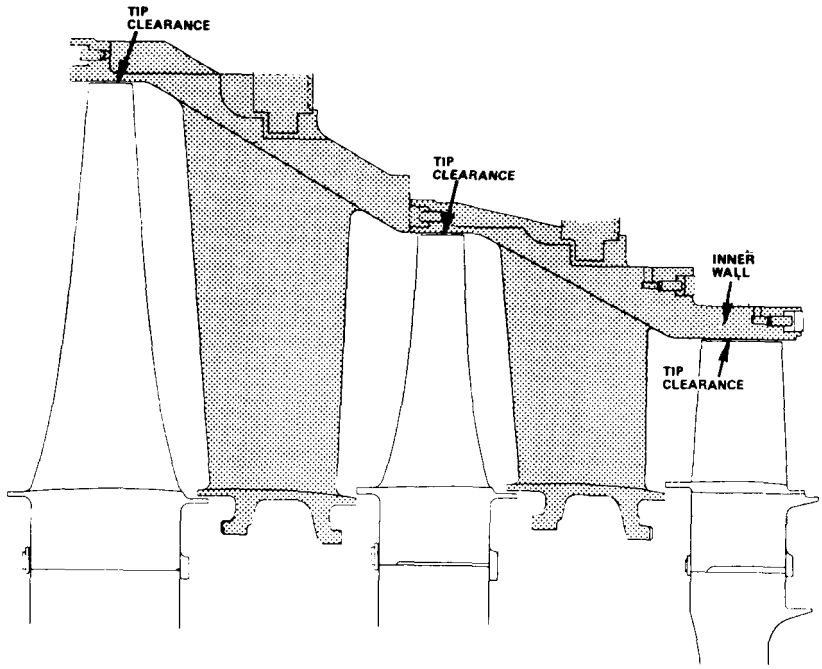


Figure 3-3 End Wall Design in Turbine - The use of cylindrical end walls preserves blade clearances during axial blade tip motion.

Rotor Support System. The welded drum compressor and tie bolted turbine rotor construction along with the short, large diameter compressor and turbine design permits a relatively light, stiff rotor design to be used, supported on only two bearings. This design allows modular component repair to the turbine while leaving the compressor portion of the rotor undisturbed.

Journal bearings are of the tilted pad type to provide damping and dynamic stability. The bearing compartment locations at the front and rear of the engine are readily accessible for maintenance. Seals are of the labyrinth type to avoid direct contact damage or wear, thus high reliability.

The front, or compressor, support and bearing takes both radial and thrust leads, the rear support and bearing carries radial loads only.

Rotor balancing at the factory would normally include individually balancing the compressor and turbine assemblies. This is followed by multi-plane balancing in a vacuum balancing facility of the complete built-up rotor at 100 percent speed (with a proof test to 120 percent speed). In the field, the rotor final trim

balance can take place in-mount with several multiple-plane balance locations incorporated into the engine design. To accomplish these options, the final Phase I HRGT layout uses both vertically and horizontally split cases.

Even with these balancing provisions and a stiff rotor, rotors vibrate briefly from slightly imperfect balancing at certain points in the operating range. Generally, the resultant vibratory modes can be limited by proper design to low speeds where the strain energy will cause little trouble. This is achieved by providing adequate bearing damping and by keeping the shaft well balanced. Any high strain modes (Mode III in Figure 3-4) can be kept beyond the operating range by careful balancing of the damping and spring rate characteristics of the bearing support system.

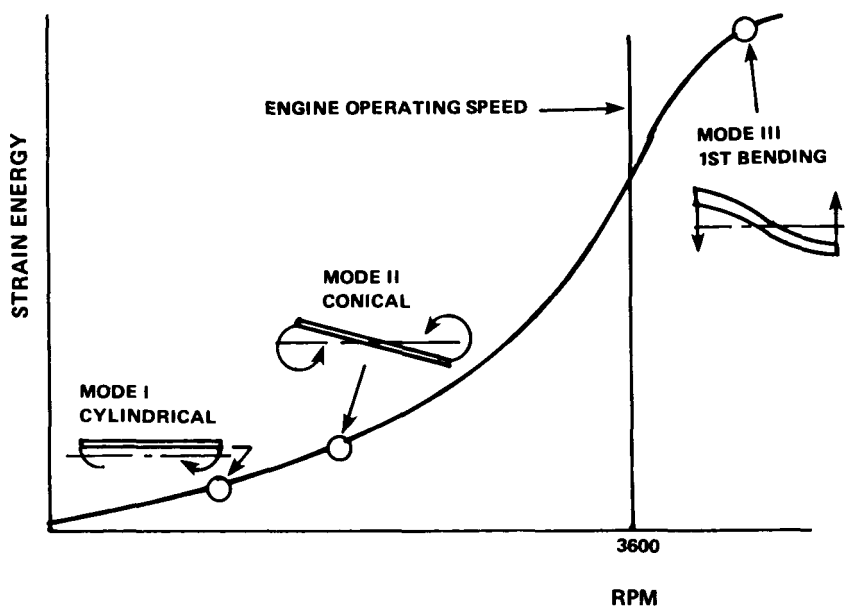


Figure 3-4 Shaft Critical Speed Vibration Modes - Mode III strain modes can be kept beyond the operating range by careful balancing of the damping and spring rate characteristics.

The use of only two radial bearings minimizes the power loss and heat rejection requirements for the lubrication system. This results in a further simplification for enhanced reliability. The selection of tilting-pad bearings provides damping of shaft vibrations in the operating range. In addition, the tilting-pad feature prevents oil-film instability, a journal bearing phenomenon

which can result in severe metal-to-metal contact. The tilting-pad bearings also have good self-alignment characteristics. This feature coupled with the overall shortness of the shaft minimizes shaft misalignment problems.

Each bearing compartment is protected from contamination by buffered labyrinth seals. A controlled pressure drop is maintained in the direction of the compartment, from the gas path, at all times so that no oil can escape into the gas path. Both compartments are thermally protected from over heating the oil in them by a combination of circulating cooling air, and insulation.

Easily accessible front and rear bearing compartments, with the drive shaft accessible from the front offers the best arrangement for easy maintenance. The rear compartment can be inspected and repaired without disturbing any other part of the engine. Both front and rear compartments can be removed and replaced as separate modules to minimize down time.

Compressor

The compressor selected is a constant outer diameter, nine-stage unit with a pressure ratio of 14:1. This choice was carefully worked out in compressor aerodynamic parametric studies. It makes use of new technology with potential to significantly improve engine reliability and durability, and to reduce cost.

The Reliable Engine compressor design features:

- Low aspect ratio airfoils to provide thick, damage resistant blades and a low total number of airfoils.

- Constant, larger outer diameter flowpath which results in a stiff (high L^2/d) rotor and few stages for good rotor dynamics and low cost.

- Aerodynamic loading sufficient to provide stable, surge-free operation without the need for a large number of variable geometry stators and subsonic aerodynamics throughout for high efficiency.

- Cantilevered (unshrouded) stator vanes and welded drum construction for reliability.

Mechanical Design. The mechanical design features of the compressor include cantilevered (unshrouded) stators and welded rotor construction. This design has been chosen to maximize reliability by reducing the changes of individual blade failure and by reducing the consequences if a failure occurs.

Airfoil failures can occur due to excitation of a vibratory mode due to aerodynamic design or a mechanical problem (such as a case rub), foreign object ingestion, or undetected blade flaws. The low aspect ratio blading is inherently more resistant to aerodynamic vibration (flutter), foreign object damage, and secondary damage due to blade failures.

Rubs of airfoils on the case are avoided by careful design (which is more easily accomplished with the welded drum rotor) and control of the engine to keep clearances within tolerances. To ensure the engine is operated as originally designed, surge pulsations must be avoided. To further maximize reliability, regions of potential contact with airfoil tips will be safeguarded by installing abradable material on static surfaces and incorporating thin wearaway sections on airfoil tips. The added expense of the abradable material will result in improved reliability.

Compressor Stability. The selected compressor design uses an aerodynamic loading which should be sufficient to maintain adequate surge margin for extremes of hot ambient temperatures and part-power operation. This is achieved with only a single variable inlet guide vane. Higher aerodynamic loading would necessitate additional stages of variable geometry, increasing complexity by adding actuation equipment and controls.

A safe margin between the operating and surge condition is provided by choosing the proper aerodynamic loading and design aspect ratio. As a rule, low aspect ratio designs are more tolerant to off-design stall conditions allowing a relatively high level of aerodynamic loading to be used while providing adequate surge margin.

A preliminary surge investigation conducted during Phase I indicated that fixed geometry operation could provide adequate stall margin up to 120°F ambient temperature at 3600 RPM (Figure 3-5). Further investigation during Phase II will verify that a combination of inter-stage bleeds and variable inlet guide vanes can handle start-up. Start motor power requirements depend to some extent on

compressor stability under starting conditions and therefore will be coordinated with the surge feasibility study.

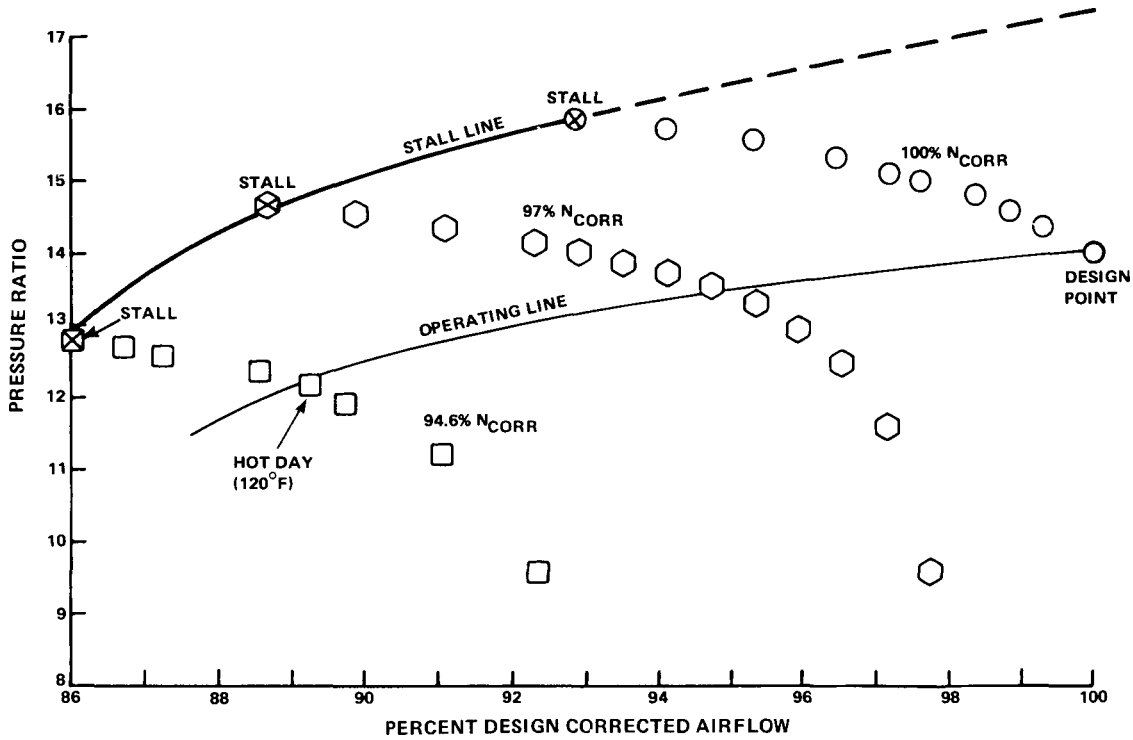


Figure 3-5 Compressor Off-Design Performance Predictions - Performance predictions indicate adequate stall margin up to 120°F.

The selected low aspect ratio compressor design is an advance in compressor state of the art, and hence represents a technical risk. This means that additional development testing is required to verify that the proven design techniques used to define this compressor remain valid for this low aspect ratio, low number of stages design. Such development is now underway.

Low aspect ratio airfoil design has been pioneered by the Government Products Division of the Pratt & Whitney Aircraft Group. This work started before 1972 as part of in-house R&D programs, but more recently has been carried forward as part of a DOE research program.

Several design test rigs at Pratt & Whitney Aircraft have successfully demonstrated the advantages of low aspect ratio aerodynamics. In 1977, a 7-stage rig demonstrated 7.5 pressure ratio with good efficiency and surge margin. In the DOE program, a small size compressor with six low aspect ratio axial stages followed by a high pressure ratio radial-stage has been assembled in preparation for tests. Results of this test will have an important bearing on future R&D requirements for low aspect ratio compressors.

Combustor

General Description. A major objective for the HRGT program was to define a reliable and durable combustor capable of meeting emission requirements when using fuels ranging from light distillate to heavier residuals, and coal derived liquids and gases. There is today insufficient technology to design such a combustion system. There is, however, a growing body of knowledge that shows the directions which must be taken to achieve design maturity. United Technologies was the Contractor for the Environmental Protection Agency's Contract 68-02-2136, "Advanced Combustion Systems for Stationary Gas Turbine Engines". The contract had as its goal the identification of combustor concepts which would permit low emissions operation using clean fuels and fuels containing bound nitrogen. From this contract emerged the Rich Burn/Quick Quench combustor.

The Rich Burn/Quick Quench combustor is shown in Figure 3-6 and is the only demonstrated concept capable of environmentally acceptable combustion of coal derived liquid fuels which can contain high levels of bound nitrogen. Several DOE/NASA contracts have recently been implemented with the goal of extending this combustor concept, and potentially discovering others.

Within the constraints of the available state-of-the-art, the Rich Burn/Quick Quench (RBQQ) combustor has been configured into a sound conceptual design for the Reliable Engine. A high level of specificity has been achieved in the design because of two factors: 1) United Technologies Corporation experience from having developed the concept, 2) United Technologies Corporation-funded studies of alternate technical approaches for developing the concept to an engine-readiness state.

A description of this design is presented below, along with a brief discussion of RBQQ NOx control phenomena. Important development work which is required to

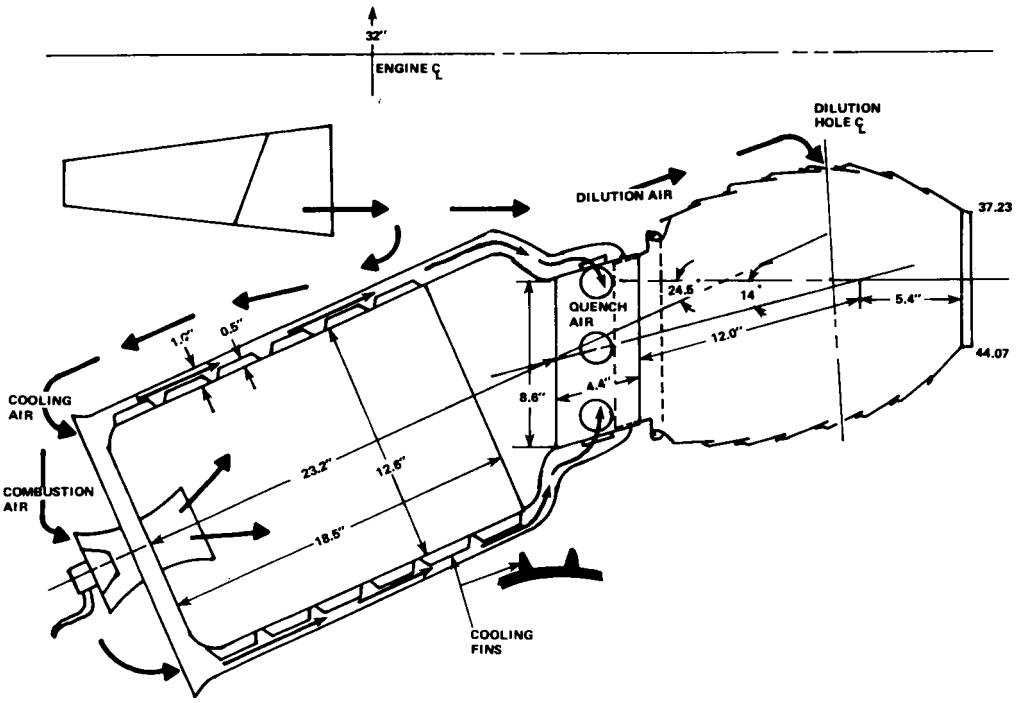


Figure 3-6 Rich-Burn/Quick Quench Combustor Design - This concept is the only concept currently known that is capable of environmentally acceptable combustion of coal-derived liquid fuels.

realize fuel flexibility with low emissions for utility engines is summarized. The procedure used to select this combustor for the HRGT is also described.

Combustion Process. Two combustion processes were investigated and evaluated. The first, lean burn, is the process developed for the DRY NOx™ Burner in the FT4 engine. The second process, Rich Burn/Quick Quench, was developed in an experimental program to meet the EPA's emission standards while burning coal derived fuels.

The lean burn process minimizes thermal oxides of nitrogen (NOx) by keeping the flame temperature low. At present, this process meets the EPA emission standards when burning distillate fuels.

The selected Rich Burn/Quick Quench burner has three major sections to minimize the production of oxides of nitrogen (NOx), carbon monoxide (CO), and unburned hydrocarbons (UHC). The three sections are the rich burn, quick quench, and the lean burn sections.

The rich burn process is employed in the initial stages of combustion. The flame in this section is starved for oxygen, which causes the NOx to dissociate to diatomic nitrogen (N_2) and oxygen, which aids in the combustion process.

The quick quench section introduces a large quantity of air rapidly, which lowers the gas temperature to minimize the formation of thermal NOx, and stops the formation of CO. The CO is formed by the dissociation of CO_2 at temperatures greater than 2800°F in the rich burn section.

The final-stage is the lean burn section. The temperature is held between 2600 and 2800°F to facilitate the oxidation of CO to CO_2 and the completion of the combustion process.

It has been demonstrated by rig tests that the rich burn/quick quench process has a high potential to meet present and future EPA emission standards when used in an engine that uses dirty fuels as well as distillates. The level of fuel bound N_2 does not seem to affect the processes of emission when below 1% concentration.

Rich Burn Quick Quench Combustor Background. The rich burn/quick quench combustor concept (Figure 3-7) has been chosen for the proposed new centerline design because it has demonstrated the capability of achieving low emissions with nitrogen containing fuels.

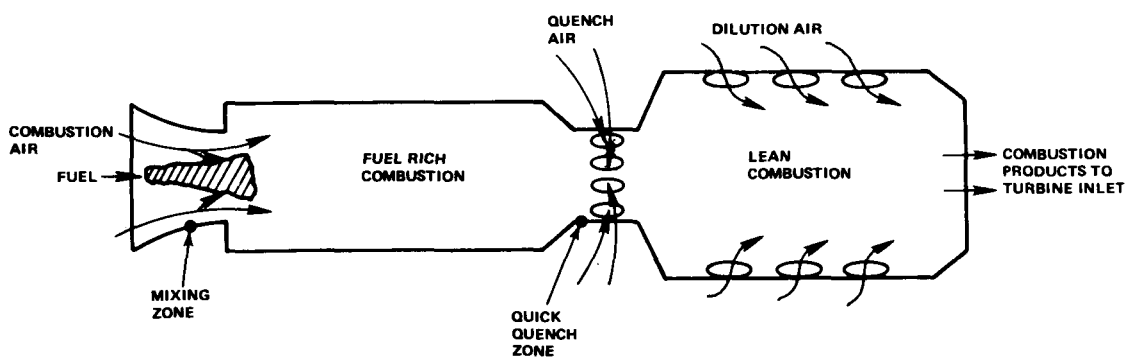


Figure 3-7 Rich Burn/Quick Quench Combustor Concept - The system provides both a rich and a lean combustion zone to minimize emissions.

Emissions test results for the combustor are shown in Figure 3-8. As shown, NOx level of 50 ppm (corrected to 15% oxygen) has been demonstrated in a subscale combustor with distillate fuel doped with 0.5 weight percent nitrogen. A NOx level of 78 ppm has been achieved in an FT4 scale RBQQ combustor with the same fuel. Further, the concept has been shown to be rather insensitive to fuel nitrogen content which may allow operation on residual, coal derived, and shale oil derived fuels with over 1 percent nitrogen within EPA limits.

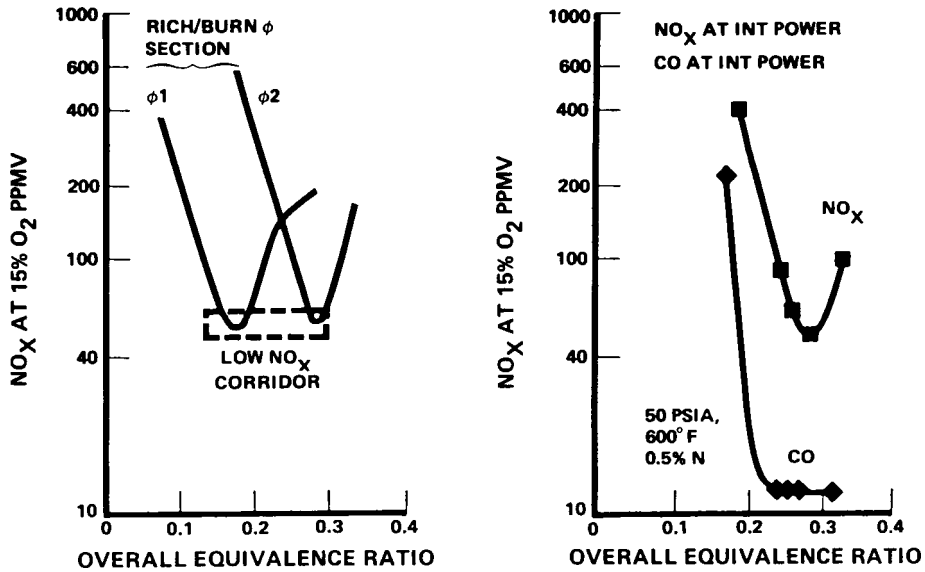


Figure 3-8 Emissions Results Obtained for Rich Burn/Quick Quench Subscale Combustor with Distillate Fuel Doped with 0.5 Weight Percent Nitrogen - NOx level of 50 ppm has been demonstrated

The DOE-NASA Low NOx Heavy Fuel Combustor Concept Program and other United Technologies Corporation advanced combustor programs address further optimization of the rich burn quick quench combustion for even more difficult heavy fuels.

Cooling Analysis. A preliminary cooling analysis was conducted for the rich burn/quick quench canted can annular burner selected for the HRGT. The analysis of the lean burn annulus section indicates that air will be adequate for cooling the panels. The metal temperature will be maintained around 1450°F, which will allow the burner to have the specified life. The analysis indicates that approximately 12.5% of the burner airflow will be required to cool this section.

The preliminary heat transfer analysis for the rich burn section indicates that air cooling will require more detailed analysis. Further analysis of other cooling techniques is also recommended. Other techniques would include materials (metals, alloys) presently being developed, ceramics, and steam.

Combustion System Mechanical Description. The Reliable Engine uses a can-annular combustor configuration, a proven concept in gas turbines for both aviation and power generation. The combustion system as shown in Figure 3-9 consists of 12 equally spaced rich burner cans, each with its own fuel nozzle and quick quench section, and a two piece lean burner/transition duct operating within an annular combustion compartment.

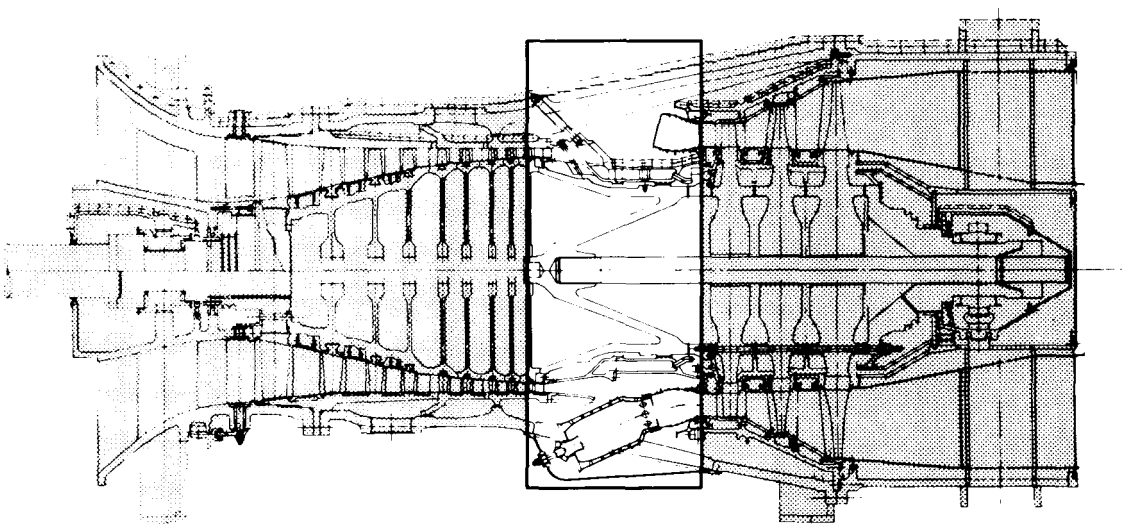


Figure 3-9 Engine Layout Showing Combustor Section - the combustor section consists of a 12-can burner system, with individual fuel nozzles and quick quench system.

The annular combustor compartment is bounded by the core cooling duct on the inside and the main engine cases on the outside. These cases are substantial structural members, and are used for combustion system support. The fuel nozzles for each are designed to be mounted on the combustor access covers of the main case to permit easy inspection and replacement.

The combustion chamber of each can is a casting featuring external cooling fins (axial) and large air quenching holes in the aft end. Over this piece fits a sheet metal liner that serves to duct secondary air past the finned region for

cooling and feeds the quick quench holes. Attached to this liner is a support strap system that bolts to ribs integral with the main engine case, supporting and positioning the can. Primary air is used entirely for rich zone combustion and enters the can through a swirler around the fuel nozzle that is part of the cast can centerbody.

After the quick quench, lean burning takes place in the forward section of the transition duct. This unit is composed of film cooled louvres made of perforated ring segments. Beyond the lean burn zone, this assembly blends from the can configuration into a full annulus so as to feed combustion products uniformly to the turbine. The transition duct is axially split into two 180° segments to facilitate maintenance and disassembly. These ducts are supported from the turbine module support base, thus providing good alignment with the turbine first-stage vanes.

The rich burn/quick quench burner scheme for the SSBL engine is designed to replace the swirl cans and thus reduce modification to a minimum by confining it to the burner cans, transition duct and fuel system. Burner liner cooling options exist because both steam and water as well as engine air would be available for maintaining temperatures at levels necessary for durability. The use of water or steam for cooling could permit a greater latitude in combustion control particularly in the rich burn section since air/fuel interaction is critical to proper combustion reactions.

Fuel Capability. The EPA program discussed above had made great strides in demonstrating the potential of the rich burn/quick quench system in handling nitrogen containing distillate fuels. The DOE-NASA heavy fuels combustor program addresses even heavier residual and coal derived fuels which can have direct benefit to the Reliable Engine Program. Advanced combustor programs which have been proposed by United Technologies Corporation to EPRI could extend and optimize further the fuel and operational capability of the Rich Burn Quick Quench combustor for the Reliable Engine.

The low and medium Btu coal gas combustion United Technologies Corporation program sponsored by EPRI has provided the basis for future coal gas combustor designs for the Reliable Engine. The inherent ease to maintain and service the proposed combustor for the new centerline design provides the flexibility required for dual fuel (gas or liquid fuel) operation with minimum switch-over time for nozzle or first-stage can combustor replacement.

A study which involved trade-off of fuel flexibility vs emissions was conducted early in the program. This showed that the reference gas turbine and its auxiliaries could be modified to operate on a heavier fuel, such as No. 6, provide a plant equivalent availability of 85.6% and a cost of electricity up to 5 mills/KWHr lower than on No. 2 fuel. This is due in part to the modular design of the engine which permits rapid repairs and minimizes the effect of more repairs during hot section inspections associated with the heavier fuel. The study also showed that the plant can operate on a blend of No. 6 and No. 2 fuel with an availability of 86.4% at a cost of electricity up to 2.6 mils/kWHr lower than on No. 2 fuel.

Combustor Maintainability and Accessibility. The combustion system has traditionally been an active maintenance area due to high temperatures and thermal gradients. With this in mind, the Reliable Engine has been designed to provide ready access for inspection of critical items, as well as quick changing of parts with minimum disassembly of the engine as shown in Figure 3-10.

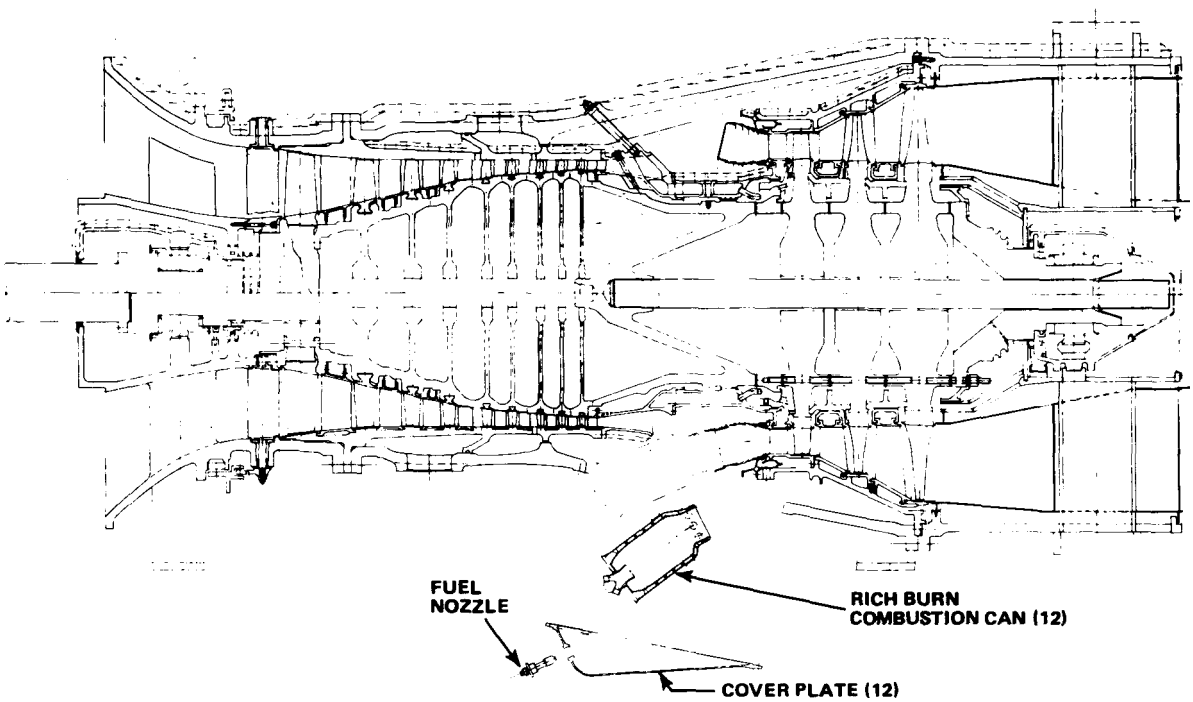


Figure 3-10 Combustor Disassembly - The combustor has been designed to permit rapid inspection of critical parts and quick parts changes.

Individual fuel nozzles can be removed, inspected and replaced with no engine disassembly. This is a simple screw-in/screw-out type operation. Any one of the twelve rich burn/quick quench cans is easily removed with a minimum of engine disassembly. An eliptical cover over the subject can is removed. The can support ring is then unbolted from the rib on the main case inner wall. The burner is slid forward to disengage the piston ring type seal to the transition duct, and then lifted radially away from the engine. Reversing this procedure replaces the can.

The translating burner case permits in-mount hot section maintenance by providing access to the transition duct, nozzle guide vanes and first-stage turbine blades. The access to the inside fasteners on the split annular combustor and transition duct is poor. If this combustor scheme is selected for development, improved access to these fasteners should be a prime design consideration.

To remove and replace the transition duct, the twelve rich burn/quick quench cans are removed and the upper-half of the engine case is lifted from the engine. The 180° upper duct segment can now be unbolted from the turbine module support case, and slid forward to disengage the seals to the turbine assembly. The upper segment is then lifted away. The lower segment is removed in the same way, after being rotated about the engine centerline until it clears the main case.

Ignition Techniques. Two proven techniques were investigated. The first involves the use of cross-over tubes which rely on flow between two cans for ignition. This technique has had durability problems in the past.

The second technique employs igniters in each can and relies on redundancy for reliability. Two igniters are used for each can, requiring a total of twelve ignition systems. Each system is connected to two cans, and no two cans are conected to the same system. A system for verifying ignition is included. The use of redundant igniters and ignition verification results in a more reliable engine.

Turbine

The final flowpath design for the turbine is shown in Figure 3-11. It is designed to pass 840 pounds per second of net airflow at an inlet pressure of 206 psia and a vane inlet temperature of 2150°F. It is a three-stage turbine with essentially constant inner diameter. The work split shared between the three stages is 37%/37%/26% with-stage efficiencies of 89.0/90.3/92.3 percent, respectively. The mean velocity ratio is 0.62 and the overall expansion ratio is 12.07.

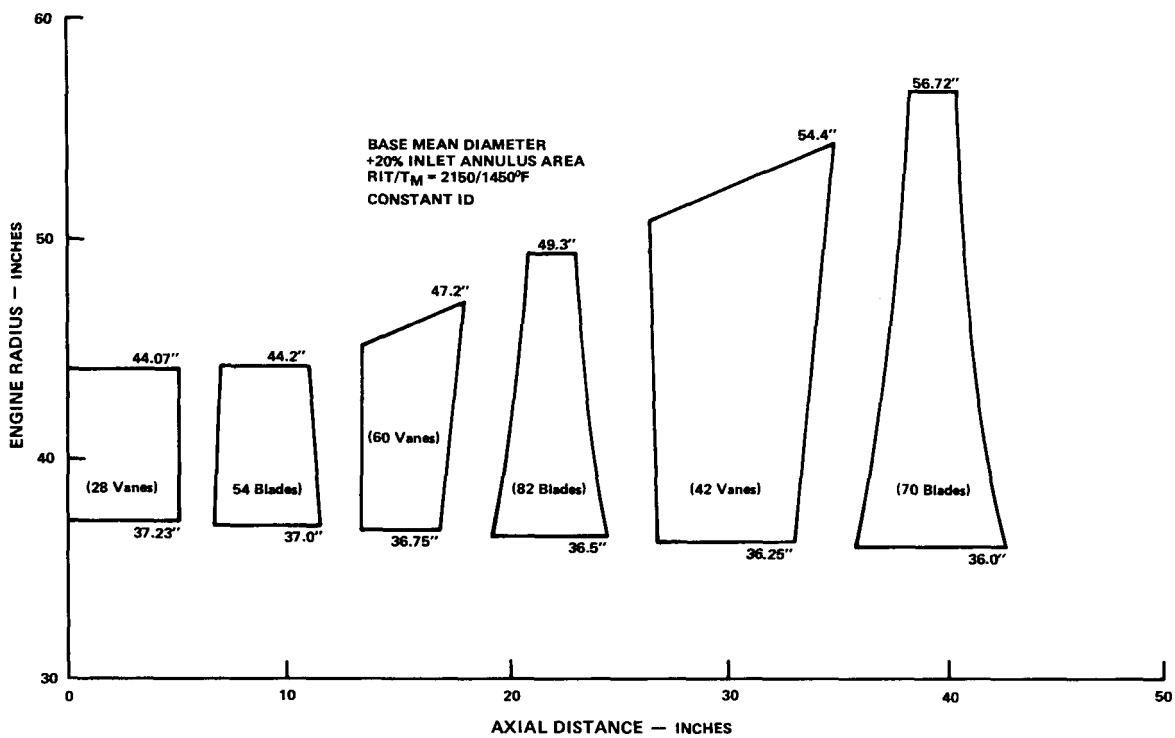


Figure 3-11 Turbine Design - The turbine provides an overall efficiency of 92.3 percent with 336 airfoils.

The total number of airfoils in the turbine is 336. This number was the minimum consistent with the airfoil chords and the maximum airfoil loading (Zweifel coefficient) judged reasonable based on United Technologies' accumulated experience. The same criteria for load coefficient was used in the parametric study as was used in the baseline.

The maximum blade and vane surface hot spot temperature was set at 1450°F. Average metal temperatures for all blade rows are shown in Figure 3-12. Significantly, the corrosion life related maximum metal temperatures are several hundred degrees F below the levels experienced by current combustion turbines in peaking service.

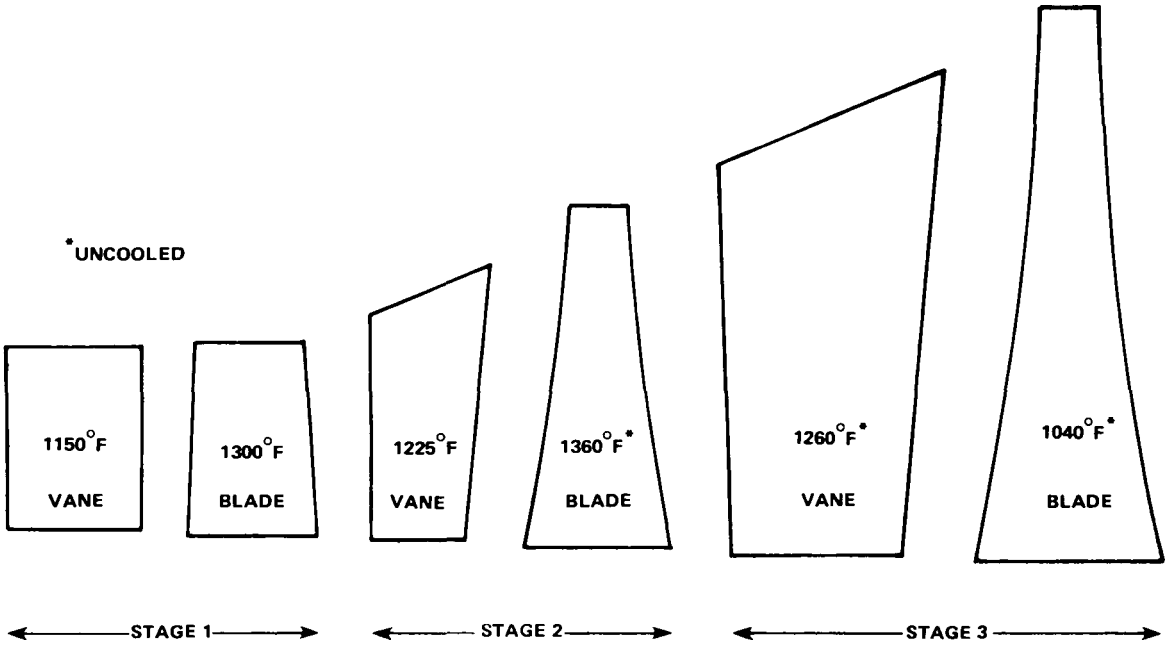


Figure 3-12 Average Turbine Airfoil Metal Temperatures - The metal temperatures are significantly below those of current combustion turbines in peaking utility service.

For reliable operation, even when the engine is required to use heavy, high ash-content liquid fuels, the cooling design uses no film cooling. Film cooling holes can be plugged from the ash residue contained in fuels such as oil or coal-derived liquids.

The turbine airfoil design uses only all-convective cooling techniques. These techniques take advantage of the latest proven technology available from on-going heat transfer, materials, and airfoil fabrication process programs. It has been found that high internal heat transfer effectiveness levels can be realized by forming blade shapes in twopiece castings which are then bonded. The

cooling designs preliminarily selected are illustrated in Figure 3-13. The first two vanes have two-piece cast internal convective casting with high surface area internal cooling passages. The last two blade rows and third-stage vanes are uncooled.

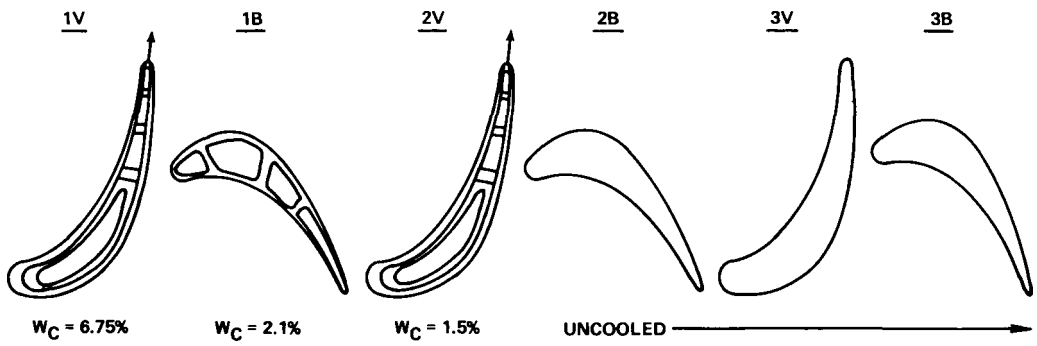


Figure 3-13 Turbine Airfoil Cooling Geometry - Internal cooling is used to avoid external cooling holes subject to clogging.

A critical element in the cooling design is the temperature of the coolant airflow. This dependance is shown in Figure 3-14, where it is seen that reducing the temperature of the coolant flow from the compressor from 700°F to 400°F provides a major benefit in reducing cooling airflow quantities for a given turbine metal temperature with a beneficial result on turbine performance.

In conjunction with combined cycle operation, providing 400°F turbine cooling air is achieved by a simple system. Compressor discharge air is brought outside the engine into an air-water exchanger, which transfers heat to the steam bottoming system feedwater, as shown in Figure 3-15. The system is simple and reliable, and the only added mechanical component is a heat exchanger.

The engine design allows the entire turbine section to be removed from the engine while the rest of the engine remains in place. Delays associated with removal, repair, and replacement can be minimized. Provisions for support rods are designed into the compressor case to hold its rotor in place with the turbine removed. The turbine shaft is fastened to the compressor shaft by a central tie-bolt which can be untightened from the rear. Torque is transmitted through the shaft by a splined connection joint. Figure 3-16 illustrates how the turbine module as a complete unit can be removed from the engine, simply by removing the central rotor tiebolt and turbine outer flange bolts.

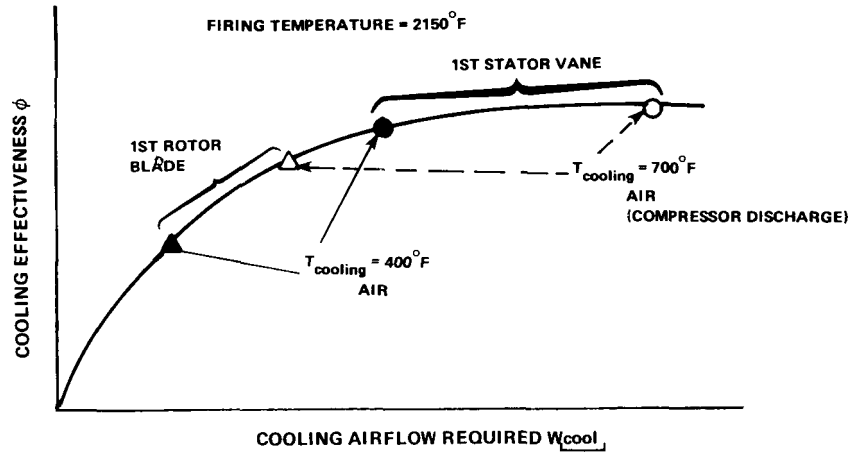


Figure 3-14 Effect of Coolant Temperature on Cooling Effectiveness - Reduced coolant temperature significantly reduces the required coolant flow rate.

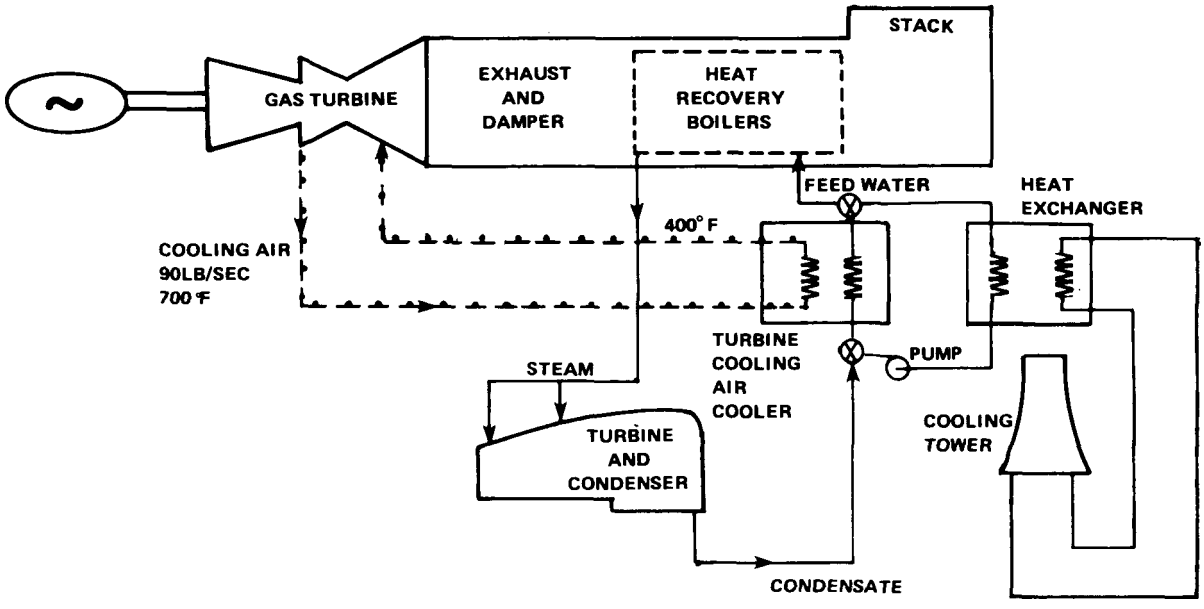


Figure 3-15 Turbine Airfoil Cooling Air Supply System - An air-water exchanger adds minimum complexity with large benefits to engine performance.

The first-stage vanes and blades are designed so that they can be removed from the engine by direct access through the combustor. Being able to inspect and individually replace these critical airfoils as part of a minor hot section maintenance procedure will ensure high engine availability. Turbine vane removal

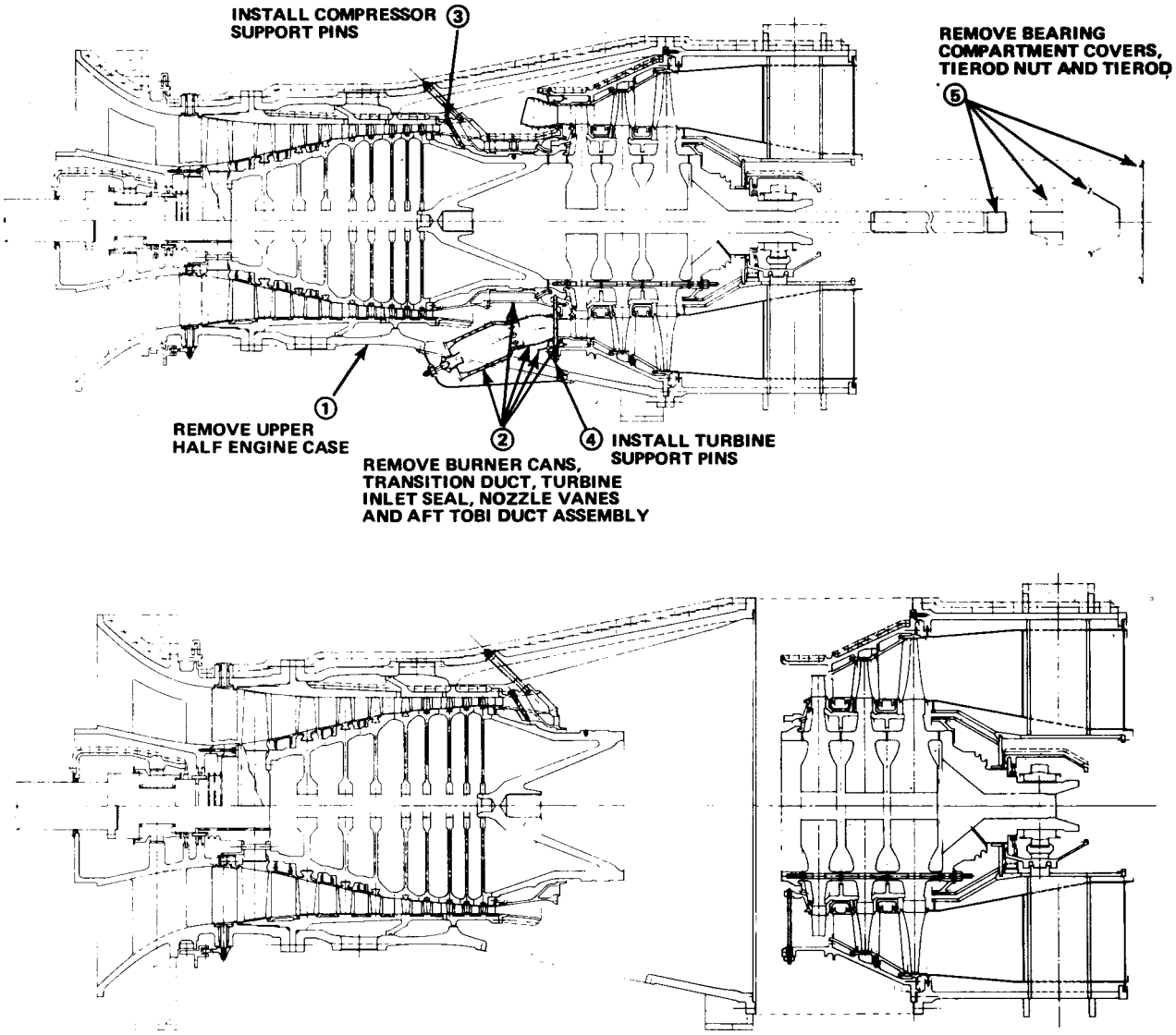


Figure 3-16 Turbine Module Removal - The turbine module can be removed as a unit by removing the central rotor tiebolt and the outer flange bolts.

is illustrated in Figure 3-17. Details of vane and blade removal will be further developed during the proposed Phase II design.

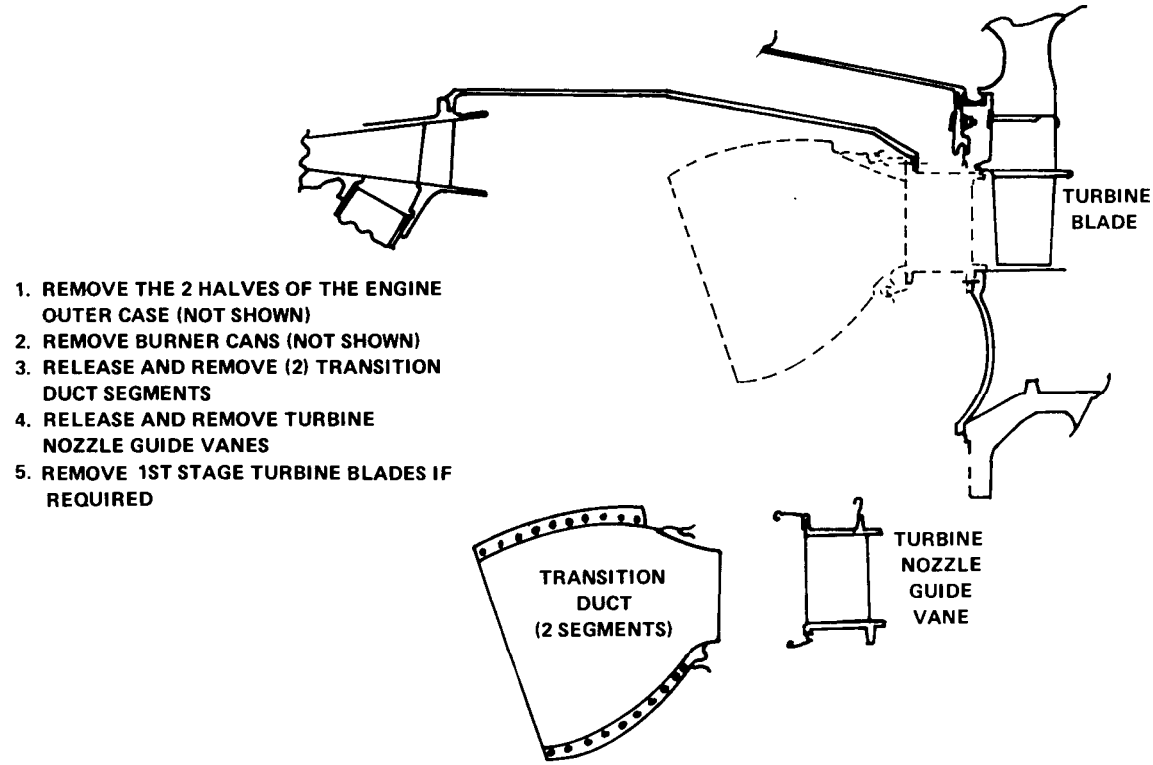


Figure 3-17 Turbine Vane Removal - Removal of the turbine vanes requires removal of the outer case and the burner cans.

GAS TURBINE ANCILLARIES AND STARTING SYSTEM

General Description

In the gas turbine auxiliaries design for the HRGT-CC, emphasis on improved reliability focused on system design and controls. In the auxiliary systems design improved availability is projected through selection of components with inherently better reliability, redundancy at the component level and a more reliability focused assumption for on-site spare components and parts. In the controls area reliability improvement is projected through the use of redundant sensors and an integrated gas turbine controller with manual override provisions.

In this section of the report, studies covering the electric generator, the start drive system for the gas turbine and reliability improvements in auxiliary fluid systems are summarized. Discussion of the integrated gas turbine controller is covered together with overall plant controls.

Electric Generator System

Generators

The electric generator for the gas turbine is a brushless excitor unit operating at 3600 rpm, to produce 60 Hz, 125 MVA power. The unit is mounted on the compressor end of the gas turbine, and has a double-ended shaft for connection to the electric start motor. Features promoting high reliability include:

- Lubrication system common with the gas turbine.
- Hydrogen cooling to provide longer-life.
- Condition monitoring to sense stator temperature, vibration, gas conditions and show trends needed for corrective action.

Several generator alternatives were considered related to generator cooling and excitation. These alternatives are summarized in Table 3-1.

Brushless excitation was selected on the basis of its being a well developed technology that eliminates periodic shutdowns for brush replacement.

A hydrogen cooled generator was selected because generators of this type are more fully developed in the 125 MVA size and offer the potential of better reliability in base load service than air-cooled machines. Somewhat better reliability is expected because of the improved efficiency and reduced internal heat generation and hot spot formation. The closed hydrogen cooling system also eliminates impurities and contaminants. There is also a performance benefit with hydrogen cooling, hydrogen cooled generators typically having 0.5 to 1 percent higher efficiency than air cooled generators. The option to consider air cooling should be retained to benefit from possible advantages.

TABLE 3-1. GENERATOR COST AND RELIABILITY COMPARISON

	H_2 Cooled		Air Cooled
Vendor	A	B	C
Relative Cost	1.0	1.015	.993
MTBF (Generator Only)	12,144		9,523(2)
Mean Corrective Time, Hrs.	16		104(2)
POH, Hrs.	(1)		(1)

(1) Overhaul, including rotor removal at 4 year intervals, plus minor maintenance, which can be accomplished during turbine POH.

(2) Experience from peaking units with many cycles. Includes a number of rotor failures related to peaking duty.

Auxiliary cooling water is used for cooling the hydrogen to avoid environmental problems associated with direct air cooling such as an intermediate water-glycol loop, louver controls, or fans.

An on-line condition monitoring and diagnostic system is planned for the generator to monitor winding temperature, vibration, gas pressure, and purity. In this way minor problems can be diagnosed and corrected before they escalate to a major repair, and scheduled outages for inspection typical of large hydrogen cooled generators may be reduced or eliminated. This is important because normal routine rotor inspection conducted on large hydrogen generators at 4 year intervals can take up to 40 days.

Lubrication Oil System

The lubrication oil system shown in Figure 3-18 includes redundant lubrication oil pumps and a dc backup pump to ensure lubrication of the gas turbine and generator even in the event of an ac power outage. The lubrication oil system includes duplex lubrication oil filters with automatic changeover to eliminate gas turbine shutdowns for filter changes.

The oil tank has baffles and special provisions to avoid flammable concentrations of air from the buffered bearings in the gas turbine and hydrogen

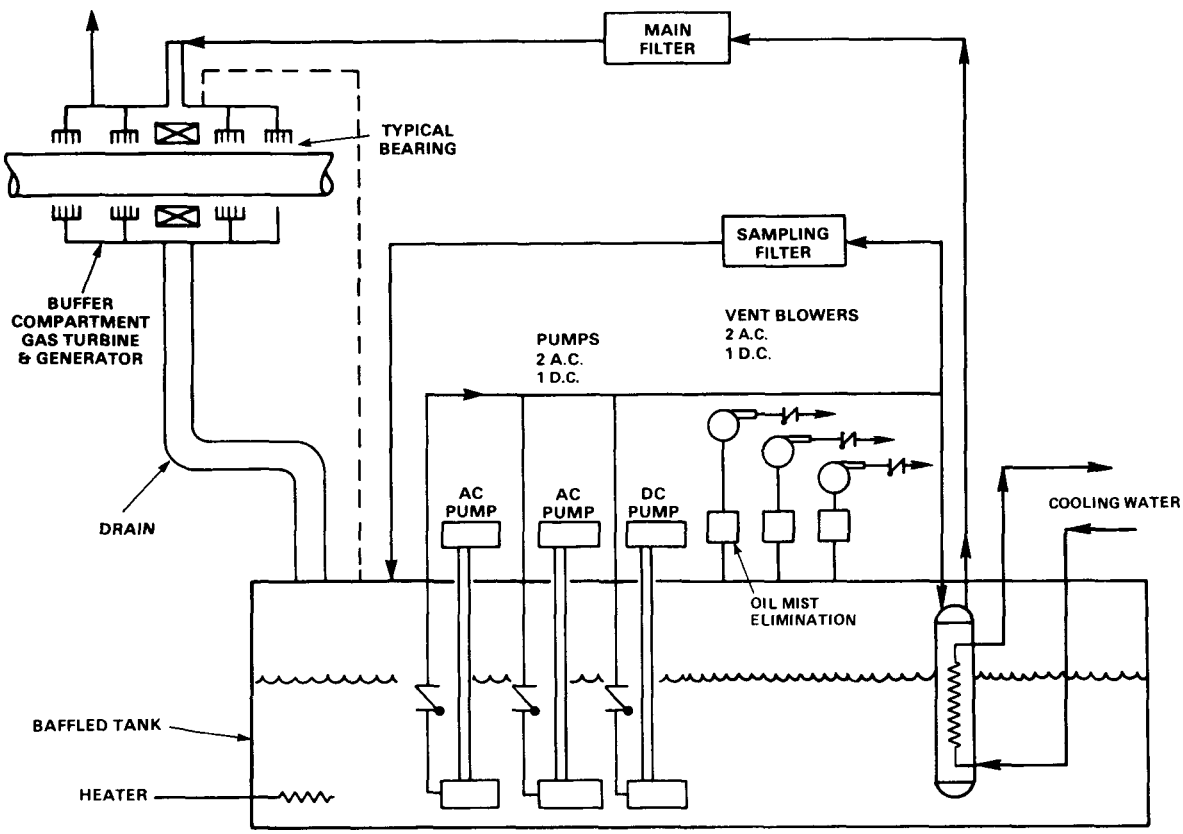


Figure 3-18 Simplified Lubrication Oil System Schematic

from the oil returning from the hydrogen cooled generator. The oil system also includes redundant vent blowers.

Fuel System

In the fuel system shown in Figure 3-19, a daytank is included to provide uninterrupted service in the event that a fuel forwarding system pump fails. The system has duplex fuel filters and a flow divider to ensure equal flow to the combustors at low flow conditions by offsetting the gravity head differences at the combustors located around the HRGT. The fuel system also includes a hydraulic servo system to hold open the fast acting fire valves. The hydraulic servo system may also service the controls for variable vanes and bleeds on the gas turbine. When the HRGT-CC operates on heavier fuels, additional elements in the fuel systems include fuel treatment and atomizing air.

Water Wash System

An automatic water wash system is incorporated for both the compressor and turbine to extend turbine life and retain high-pressure compressor performance. Water wash was selected for engine cleaning as a reference approach, but on-line methods will be considered further in Phase II to improve availability.

Fire System, Air Filter, and Salt Monitor

The preferred fire system uses Halon as the fire extinguishing agent. The fire system protects not only the gas turbine system, but each of the critical auxiliary subsystems in their packaged configuration. A special air filter and salt monitor are preferred to remove the maximum amount of salt from the engine intake air to reduce the contribution to salt sulfidation corrosion and to extend compressor wash intervals in the interest of high availability.

Each of the gas turbine auxiliaries is packaged as a skid-mounted subassembly designed with maintainability as the primary consideration in order to reduce the time for repair and maintenance and to increase availability. Auxiliary subsystems are interconnected using guarded pipe at selected critical locations to avoid mechanical damage by maintenance personnel or equipment and improve the general reliability of the auxiliary pipe system. Special attention is also planned for all interconnecting control and instrument wiring to eliminate vibration, wear, or heat induced failures.

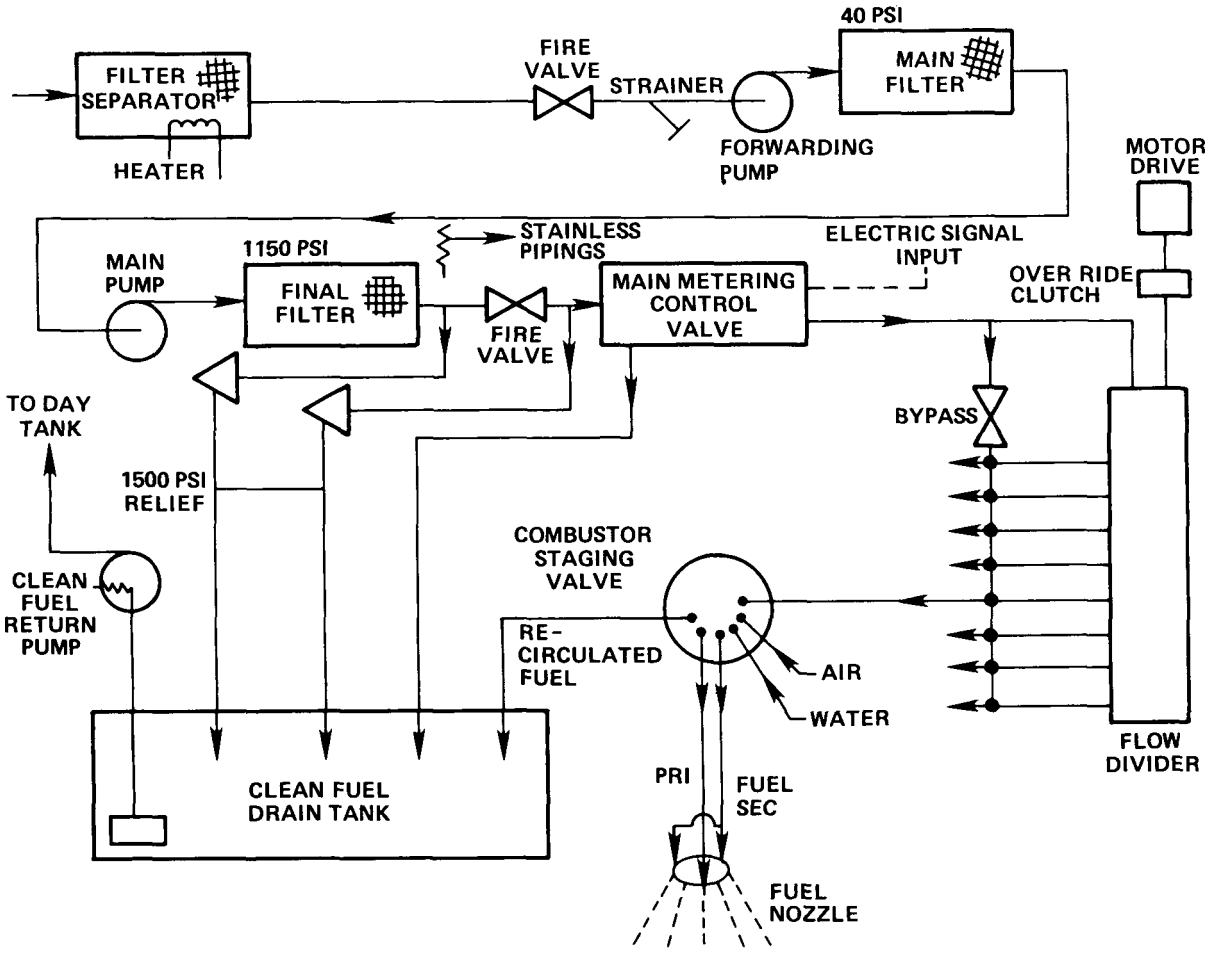


Figure 3-19 Simplified Fuel System Schematic

A breakdown of the failure rates and MCT for the gas turbine auxiliaries is presented in Appendix D.

Gas Turbine Controls

Experience by electric utility operators in general has shown that the gas turbine control system can be a major source of trouble. This was specifically borne out by the results of our site survey (Section 4). In consequence, the Phase I study has afforded an excellent opportunity to investigate root causes of past problems with an eye towards correcting them by an all-new design concept. Just as with the gas turbine itself, the "clean-sheet" (or "new centerline") design philosophy was adapted for the gas turbine control as part

of the Phase I effort. In fact, the control design task was expanded by EPRI during the course of Phase I and, as such, the results of that augmented study are presented separately in Section 7 of this report.

The gas turbine control system governs a multiplicity of functions. Recognizing that, we call the controller an integrated system (Integrated Engine Control - IEC). The integrated engine control governs start-up and shutdown, both normal and emergency, in automatic sequence. It controls acceleration and deceleration and partial power operation. It also monitors and sets protective functions such as automatic compressor bleeds or variable geometry for surge avoidance, overtemperature, oil supply, etc. Thus, by its very nature, the control tends to be complex. This very complexity can contribute to unreliability.

To achieve improved control reliability, our design developed during Phase I is based on the following guiding principles:

- Capitalize on the highly reliable digital microprocessor technology perfected by the electronics industry in recent years and now available.
- Benefit from field experience to circumvent past problems.
- Use quickly replaceable modular components.
- Simplify fault diagnostics procedures for prompt identification of malfunctions.
- Incorporate system redundancy - made practical with low cost electronic components.

How these design principles are applied to the new control design concept is presented in more depth in Section 7 of this report. Specific design requirements for the integrated engine control system's component logic and its hardware and software needs are discussed.

Starting System. The starting system was selected after a study of six alternatives. From these studies (discussed in Section 5), the following conclusions were drawn:

The ac electric motor and hydraulic torque converter should be used as the reference start system.

The variable frequency drive conversion of the generator to a start motor should be considered further.

Improvement in the gas turbine system starting reliability to the Phase I goal of 95 percent will require additional redundancy in controls and instrumentation, and in the design of the sequencer.

The specific concept selected was a 1000 HP electric motor drive system as shown in Figure 3-20. Included with the electric motor is a hydraulic torque converter, as shown in Figure 3-21, and an input and output gearbox for matching electric motor and torque converter characteristics. Information on the electric motor was supplied by Brush Electrical Machines Ltd. and information on the torque converter, gearboxes, and clutch was provided by Twin Disc, Inc. Brush Electrical Machines Ltd. presently supplies 1000-1500 hp start motors for 14 gas turbine installations.

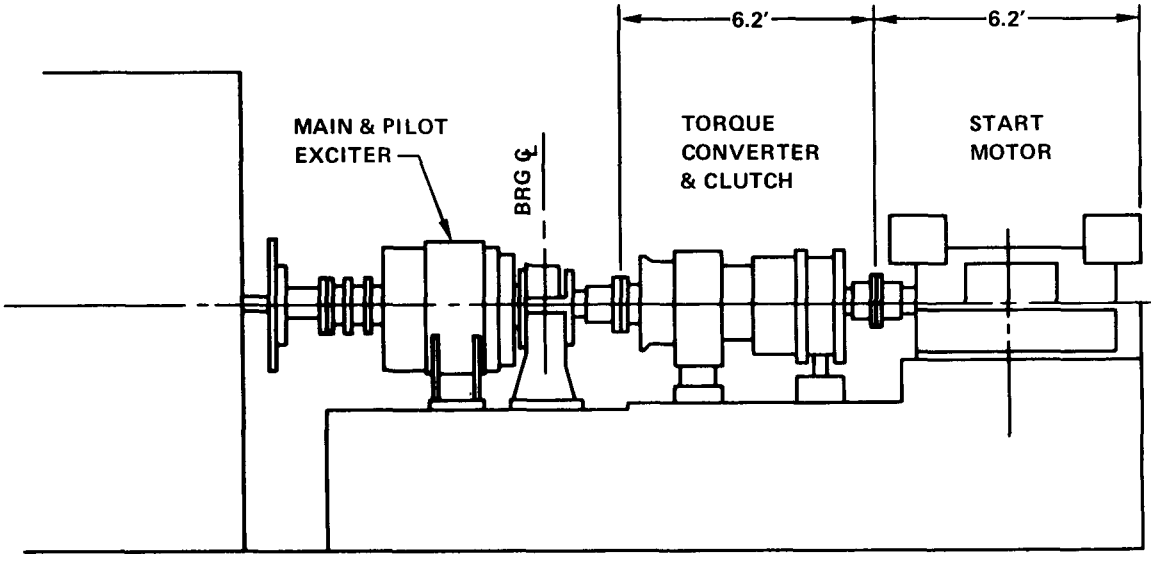


Figure 3-20 Selected Gas Turbine Starting System - The system selected features a 1000 HP electric motor drive system offering relatively low cost and high reliability.

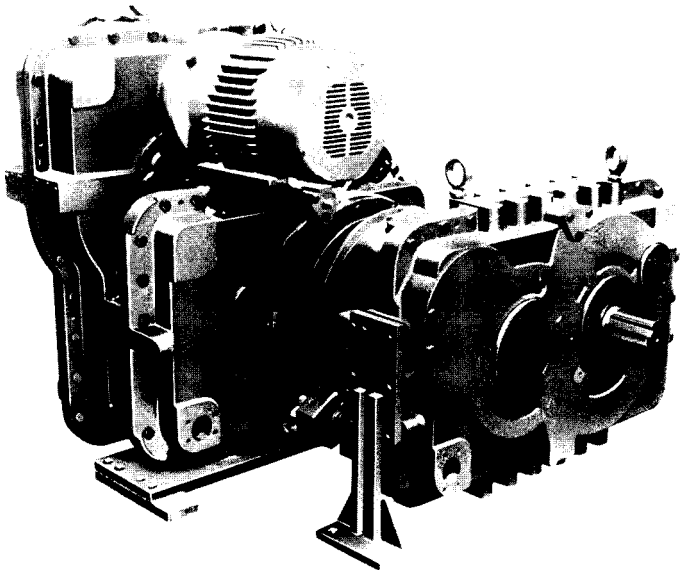


Figure 3-21 Starting System Auxiliary Components - The system includes a hydraulic torque converter and an input and output gearbox.

BALANCE OF POWERPLANT

Overall Description

In the design of the balance of the plant for the High Reliability Gas Turbine (HRGT-CC), emphasis on improved reliability was focused on system design using existing equipment. This was done because the design of the major balance of plant components already reflects many years of development for fossil plants. Attention was also given to the impact of repair parts and component replacement lead times on plant availability for large equipment. This in turn yielded more reliability focused assumptionsfor on-site components and parts.

The description of these systems includes the following:

- The steam generator
- The steam driven electric generator
- The steam system ancillaries

- The cooling tower

- The building and site

Steam Generation System

A standard design, three-pressure level boiler is used for exhaust heat recovery. The increased power and efficiency of this three-level design were judged to significantly outweigh the increase in complexity compared to single pressure level types, as further discussed in Section 6. Industrial experience with similar units indicates the basic steam tubes and drums have satisfactory reliability and durability. However, care must be taken in design of hot gas ducting and controls to achieve the program's reliability goals.

This steam generator system uses:

- High pressure throttle steam for the steam turbine.

- Intermediate pressure induction steam for the steam turbine.

- Low pressure steam for the feedwater deaerator.

Key features of the system are:

- The steam generator has natural convection water circulation for high reliability, since circulating pumps are not required.

- Bypass Dampers - Simple, field proven diverter valves are used to isolate a steam generator from its turbine to permit continued gas turbine operation if the steam system is out of service. The dampers also modulate turbine exhaust gas flow during system starting to eliminate thermal transients in the steam system.

- The stack temperature is maintained at or above 300°F to eliminate the possibility of corrosive condensation.

The steam conditions for the combined cycle plant are shown in Figure 3-22 and have peak conditions of 870 psig, and 850 ^{o}F to produce 300,000 lb/hour per

boiler. This plot shows the temperature differential between the hot gas and the steam. If the two low temperature boilers were omitted, it can be seen that the exhaust stack temperature would be much higher.

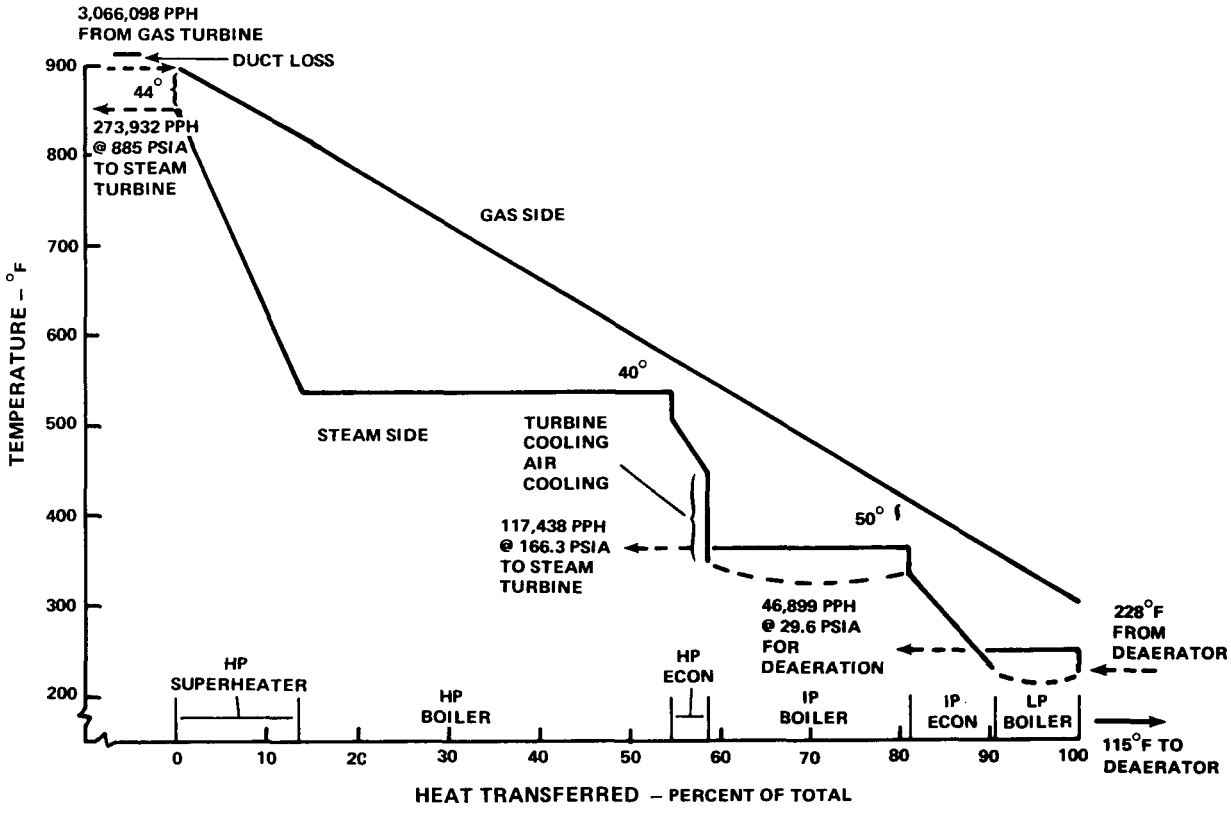


Figure 3-22 Steam System Conditions - Each boiler operates at peak conditions of 870 psig, and 850°F to produce 300,000 lb of steam per hour.

The high pressure steam is generated and superheated by the hottest gas at the gas turbine exit. The lower pressure steam is generated by the gas after it is too cool to generate high pressure steam. Economizers are used to heat the water to almost the boiling point before injection into the boilers. Placement of these components of the HRSG is shown in Figure 3-23. To maximize the heat recovery from all sources, heat from cooling the turbine cooling air is recovered in an air/water, shell/fin tube heat exchanger which serves as a first stage economizer for the high pressure boiler.

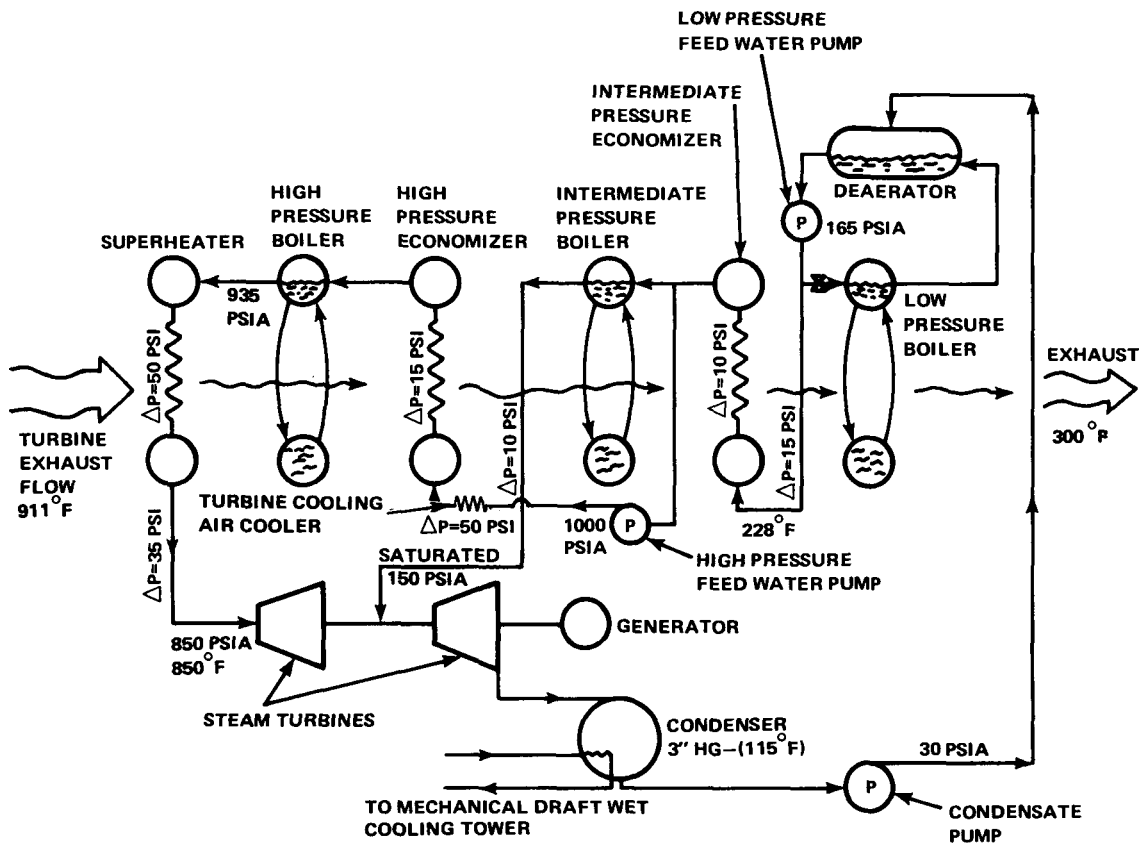


Figure 3-23 Schematic of Steam Bottoming System Plant - The boilers have been located to optimize efficiency.

The low pressure boiler provides 15 psig saturated steam to the deaerator. The intermediate pressure boiler and economizer provides 150 psi saturated induction steam to the steam turbine. The high pressure boiler, economizer, and superheater deliver 850 psi superheated steam to the turbine throttle. Steam from the turbine is condensed at 3 in. Hg. (115 °F) and returned to the boiler by a condensate pump and two stages of feedwater pumping: a low pressure feedwater pump and a high pressure feedwater pump. Gas turbine exhaust gases enter the waste heat steam generator at 911 °F and exhaust to the atmosphere at 300 °F. Feedwater from the deaerator enters the boiler at 228 °F and is delivered to the steam turbine generator at both 150 psi saturated and 850 °F superheated conditions.

The waste heat steam generator design provided by the Henry Vogt Machine Company is shown in Figure 3-24. The overall unit with its bypass damper is 122 feet long, 50 feet wide, by 50 feet high. For improved reliability, the modular heat recovery steam generator uses natural water circulation which eliminates the need for pumps. The Henry Vogt Machine Company has delivered steam generators of this type to approximately 50 installations, many of which have multiple units. The unit shown in Figure 3-24 is larger than most of the units delivered but is within the present design range of modular equipment supplied by the Henry Vogt Machine Company.

Experience of the Henry Vogt Machine Company with waste heat steam generators has been extremely good. In all of the units delivered to date, there have been only three instances of failure of boiler pressure parts. One because of internal corrosion, the others caused by damage in shipment. The system cycle was simulated on a computer program in conjunction with the gas turbine engine and included recommendations from the Henry Vogt Machine Company on factors such as bypass damper leakage and duct heat loss. Output of the steam turbine generator is based on information provided by Toshiba International.

A bypass damper was included for each waste heat steam generator to permit operation of the gas turbine during those periods when the waste heat steam generator or steam plant is inoperative. This device improves equivalent availability, and facilitates steam turbine start by modulating gas turbine hot gas flow through the boiler during boiler heat up and initial steam flow to the steam turbine.

Ducting between the gas turbine damper and boiler has a gradual transition to give better boiler performance and reduce gas turbine back pressure. The ducting materials have been selected for the highest abnormal temperature to avoid warpage. The ducting has been designed to avoid failures due to differential thermal expansion and exhaust induced vibration. Compatibility with temperatures exceeding those encountered under normal turbine operation is important so that if the turbine exceeds limits for a short period, excessive temperatures in the ducting and bypass damper will not cause warpage with resulting long outage periods for maintenance.

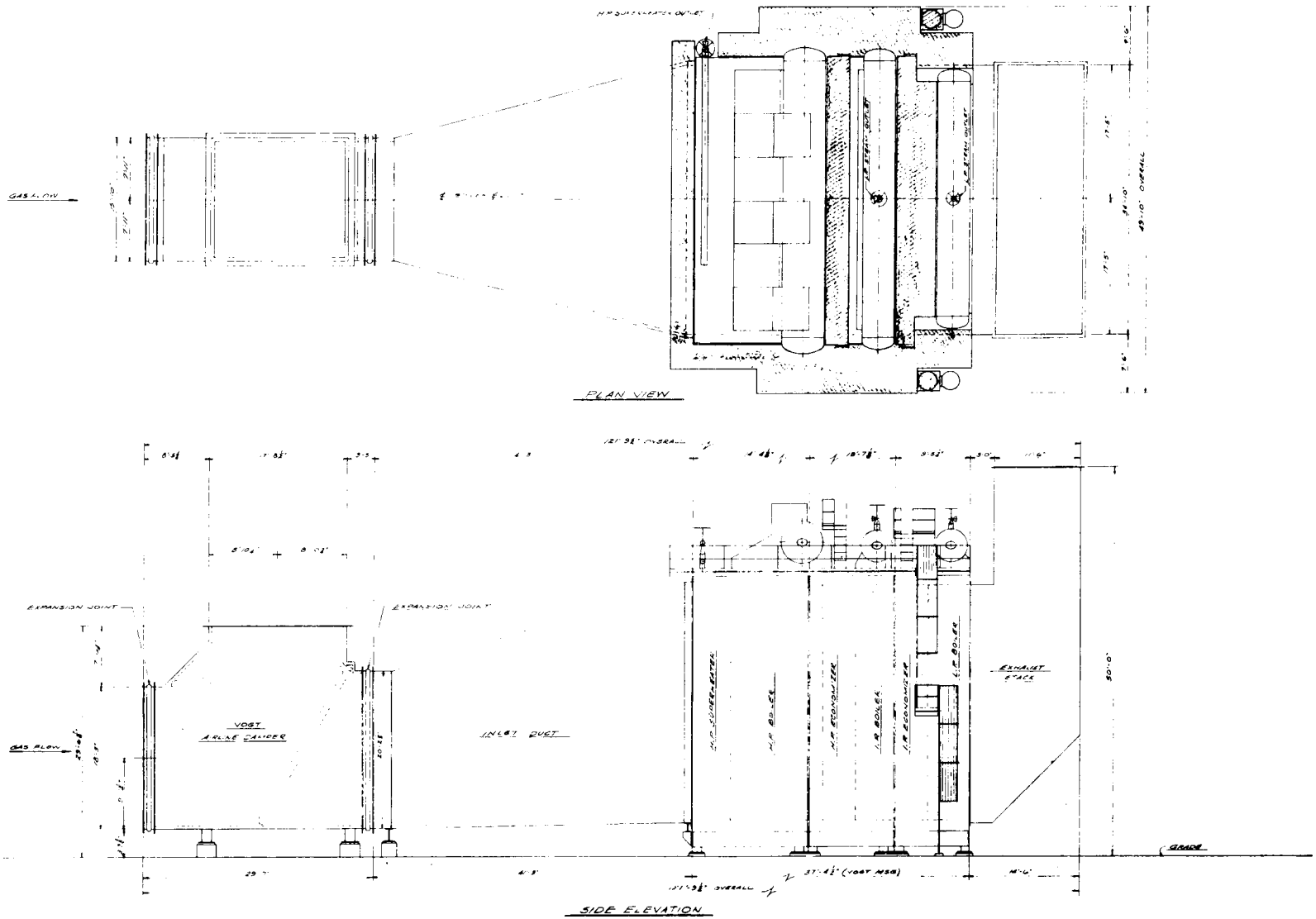


Figure 3-24 Steam Generator Conceptual Configuration - Ths overall unit is 122 feet long, 50 feet wide, and 50 feet high.

Steam Turbine/Generator

The steam turbine generator model selected is a Toshiba International Model TCDF-23. The steam turbine is a tandem compound double unit with a 23-inch last stage blade. Design of the steam turbine/generator as well as cost and reliability information were provided by Toshiba International.

The unit is shown in Figure 3-25 and 3-26. It has an independent lubrication system and its own electro hydraulic control system. Steam at 850 psia from the high pressure boiler expands through the high pressure turbine after which it is mixed with 150 psia steam from the low pressure boiler before expansion through the low pressure turbine.

The generator connected to the steam turbine is a hydrogen cooled, brushless excitor unit like the generator for the gas turbine. A condition monitoring and diagnostic system is used to track the health of the units to allow maintenance to be performed on an as-required rather than a scheduled basis.

In addition to the steam turbine generator itself, controls on the steam turbine include an EHC control unit (electrohydraulic) and an automatic sequencer.

Reliability information on these units was also provided by Toshiba International. The calculated MTBF and MCT of the steam turbine generator and controls from Toshiba data are 8040 hours and 22 hours, respectively. EEI data from fossil units in the 90 to 120 MW range taken from the 1965-74 report shows MTBF and MCT values of 8423 hours and 90 hours, respectively.

Experience on 7 units operating from 2 to 20 years with 665,000 service hours has provided an MTBF of 166,000 hours and an MCT of 357 hours, excluding controls. Experience on scheduled outages has been 967 hours per year average for 5 of the units over 53 unit years. However, Toshiba feels this is due to longer scheduled outage periods in early installations and their recommended 300 to 600 hours per year should be adequate for purposes of inherent reliability estimates. Conservatively, 600 hours was used as the resource POH for this equipment.

A breakdown of the failure rates and MCT for the steam turbine/ generator and its associated controls and the main condenser is presented in Table 16 of Appendix D.

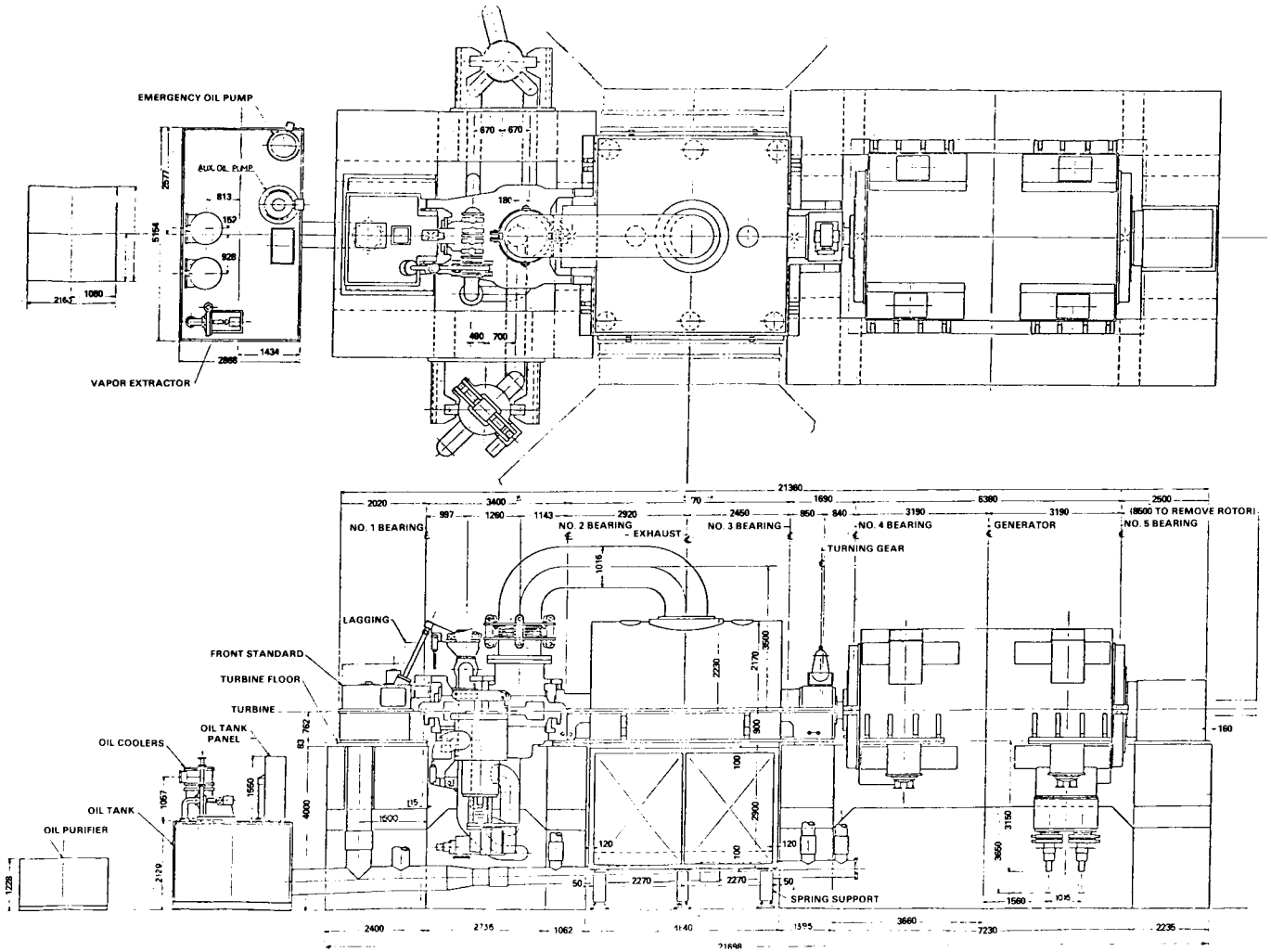
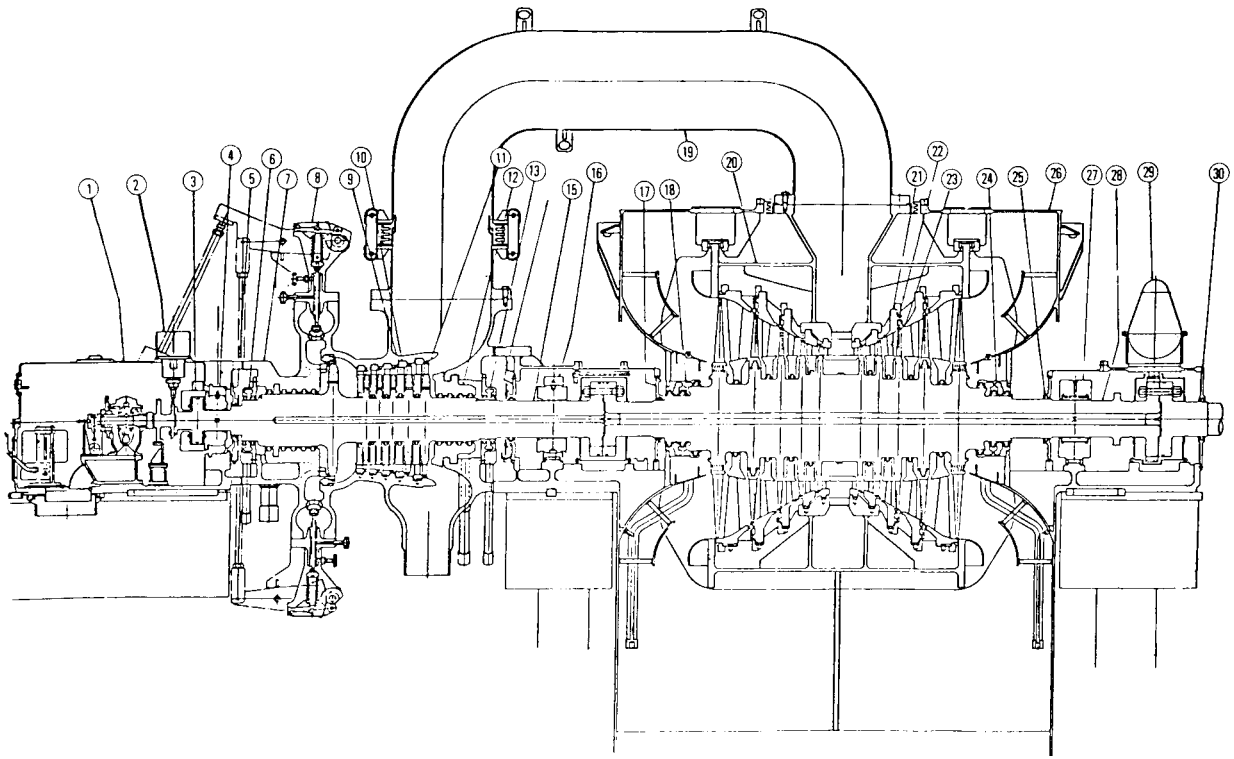


Figure 3-25 Steam Turbine Generator - The selected unit is manufactured by Toshiba International



TURBINE ASSEMBLY CROSS SECTION TC – DF 26"

NOMENCLATURES

1 FRONT STANDARD
2 THRUST BEARING WEAR DETECTOR
3 THRUST BEARING
4 NO. 1 BEARING
5 NO. 1 OIL DEFLECTOR
6 NO. 1 PACKING CASING
7 HP CASING
8 CONTROL VALVE
9 NOZZLE
10 NOZZLE DIAPHRAGM
11 BLADE
12 NO. 2 PACKING HEAD
13 NO. 2 PACKING CASING
14 NO. 2 OIL DEFLECTOR
15 HP ROTOR
16 NO. 2 BEARING
17 NO. 3 OIL DEFLECTOR
18 NO. 3 PACKING CASING
19 CROSS OVER PIPE
20 LP INNER CASING
21 NOZZLE DIAPHRAGM
22 NOZZLE
23 BLADE
24 NO. 4 PACKING CASING
25 NO. 4 OIL DEFLECTOR
26 LP OUTER CASING
27 NO. 3 BEARING
28 LP ROTOR
29 TURNING GEAR
30 NO. 5 OIL DEFLECTOR

Figure 3-26 Cross Section of Steam Turbine Generator - The generator is a tandem unit with a 23-inch last stage blade.

Steam System Ancillaries

Systems in the balance of plant supporting the steam turbine include the boiler feed water, condensate, main cooling water, auxiliary cooling water, and a number of supporting subsystems such as water treatment, waste water management, service air, and instrument air.

Some specific features identified in the design for reliability improvement include spare boiler feed pumps, spare condensate pump , the use of 3-50 percent pumps in the main cooling water system, a spare heat exchanger and pump in the auxiliary cooling water system, steam jet ejectors on the main condenser rather than vacuum pumps, manual overrides on all controls, manual bypass on selected control valves, and double shut-off valves on critical isolation controls.

Cooling Tower

The main steam condenser is cooled by a circulating fresh water loop from a mechanical draft wet tower. The cooling tower shown in Figure 3-27 was specified by Chas. T. Main and information on size, cost, and reliability was provided by the Marley Company. C.T. Main established that the plant can operate at rated output with 5 of 6 cooling tower cells at ambient conditions up to a wet bulb temperature of 72.5^{o}F. Therefore, one of the cooling tower cells in the design is considered as an operating spare in the reliability estimate.

Building and Site

A representative site plan for the HRGT-CC is presented in Figures 3-28 and 3-29. This site plan, prepared by Chas. T. Main Engineers, Boston, Massachusetts, shows the building which encloses the two gas turbines with their generators and auxiliaries and the adjacent building which encloses the steam turbine generator and balance of plant systems. Also shown are the two heat recovery steam generators (HRSG), the six cell mechanical draft wet cooling tower and cooling water pump building, fuel tanks for 30 day supply, fuel treatment system for heavy liquid fuels, tanks for raw water, demineralized water, and condensate storage, auxiliary boiler, fire pump building, the step-up transformers, maintenance building and administration building. The overall site is 840 feet by 544 feet and occupies about 9 acres.

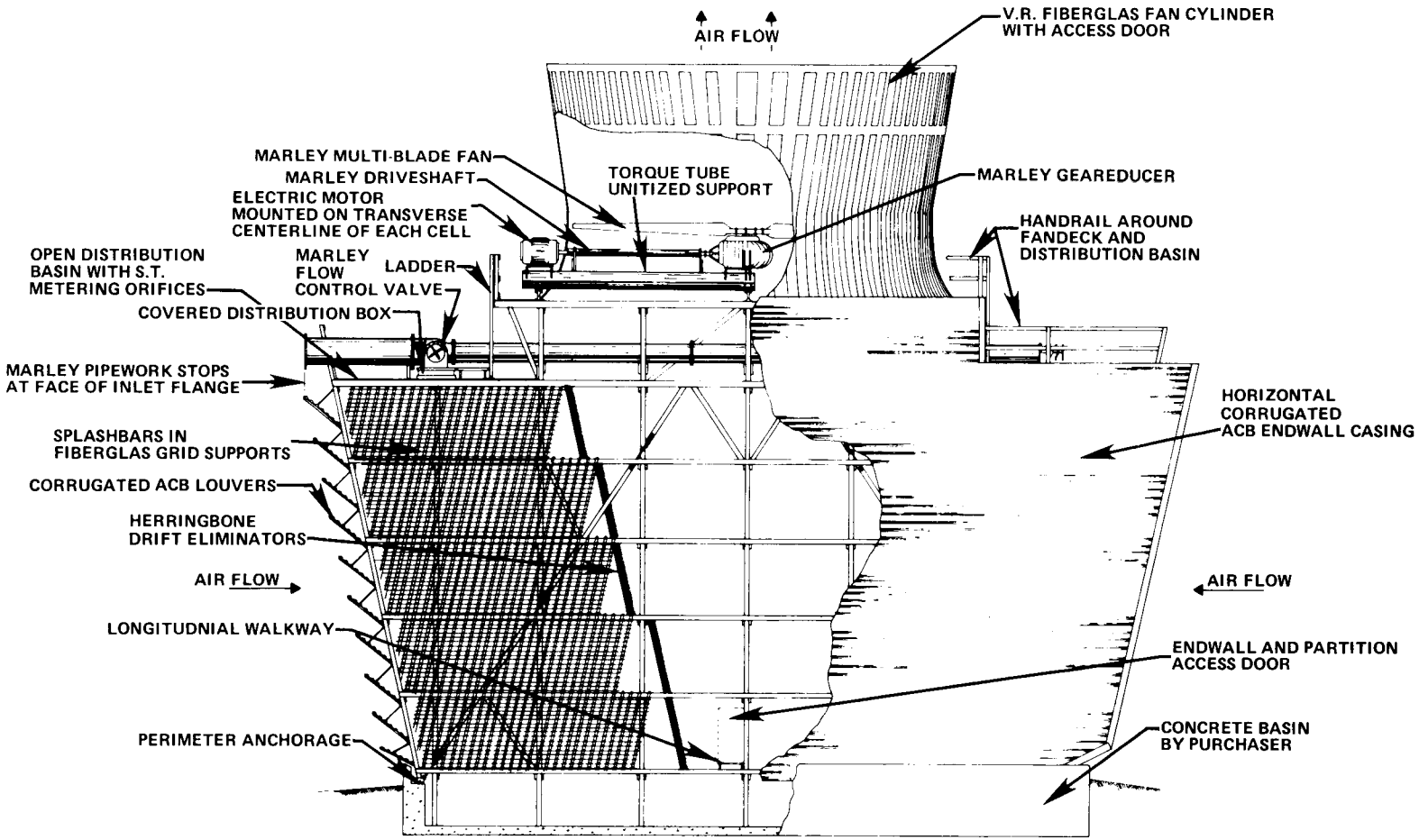
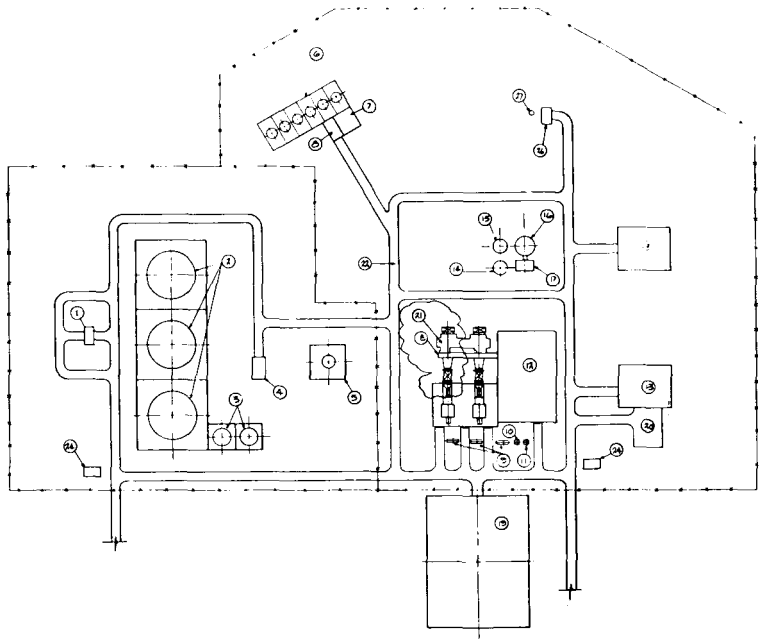


Figure 3-27 Cooling Tower Design - The design provides an operating spare cell for high reliability.



LEGEND

1. FUEL OIL UNLOADING BLDG.
2. F.O. STORAGE TANKS
3. F.O. PROOF TANKS
4. F.O. FORWARDING BLDG.
5. F.O. DAY TANK
6. COOLING TOWER
7. CIRCULATING WATER PUMPS BLDG.
8. CHEMICAL TREATMENT BLDG.
9. STEP-UP TRANSFORMER
10. AUXILIARY TRANSFORMER
11. START-UP TRANSFORMER
12. TURBINE BLDG.
13. SERVICE & ADMIN. BLDG.
14. CONDENSATE STORAGE TANK
15. DEMIN. WATER STORAGE TANK
16. RAW WATER STORAGE TANK
17. FIRE PUMP HOSE
18. PIPE RACK
19. SWITCHYARD
20. PARKING LOT
21. H.R.S.G.
22. ACCESS ROAD
23. WAREHOUSE BLDG'.
24. GUARD HOUSE
25. SECURITY FENCE
26. WELLWATER PUMPHOUSE
27. WELL

Figure 3-28 Site Plan for High Reliability Gas Turbine Combined Cycle Plant - The plan is typical for the planned application and occupies about nine acres.

All turbomachinery is housed in a single building to centralize control of the plant and provide better utilization of maintenance space and equipment. The arrangement of equipment in the gas turbine building and the adjacent steam turbine building is shown in Figure 3-30.

A centrally located control room allows operating personnel to be close to the equipment for sensing of equipment status and to minimize time for manually correcting problems.

A high bay section with overhead crane and a low bay section which houses the generators and auxiliaries are provided. The high bay houses the two gas turbines, turbine air coolers, and atomizing air systems.

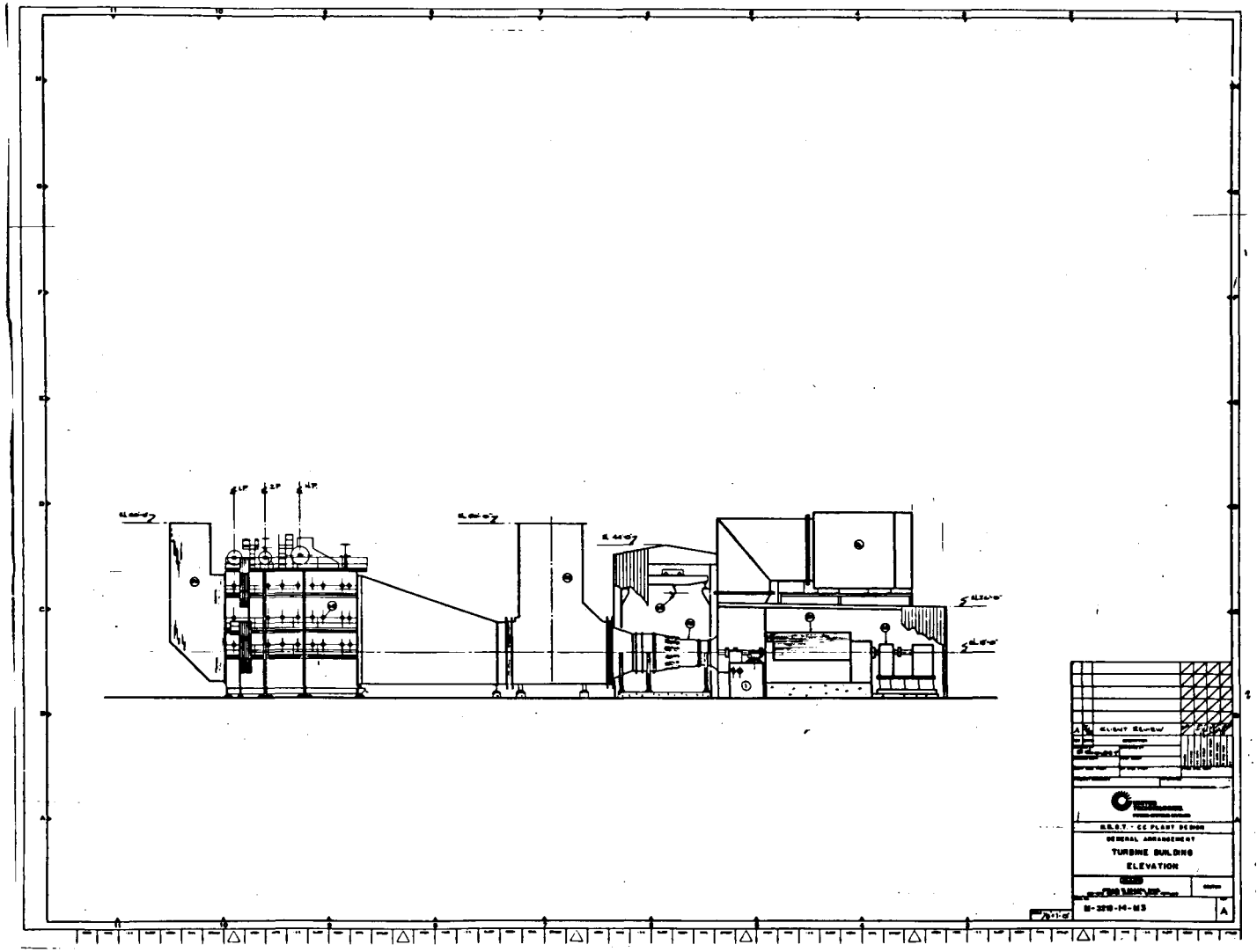


Figure 3-29 Elevation of High Reliability Gas Turbine Combined Cycle Plant

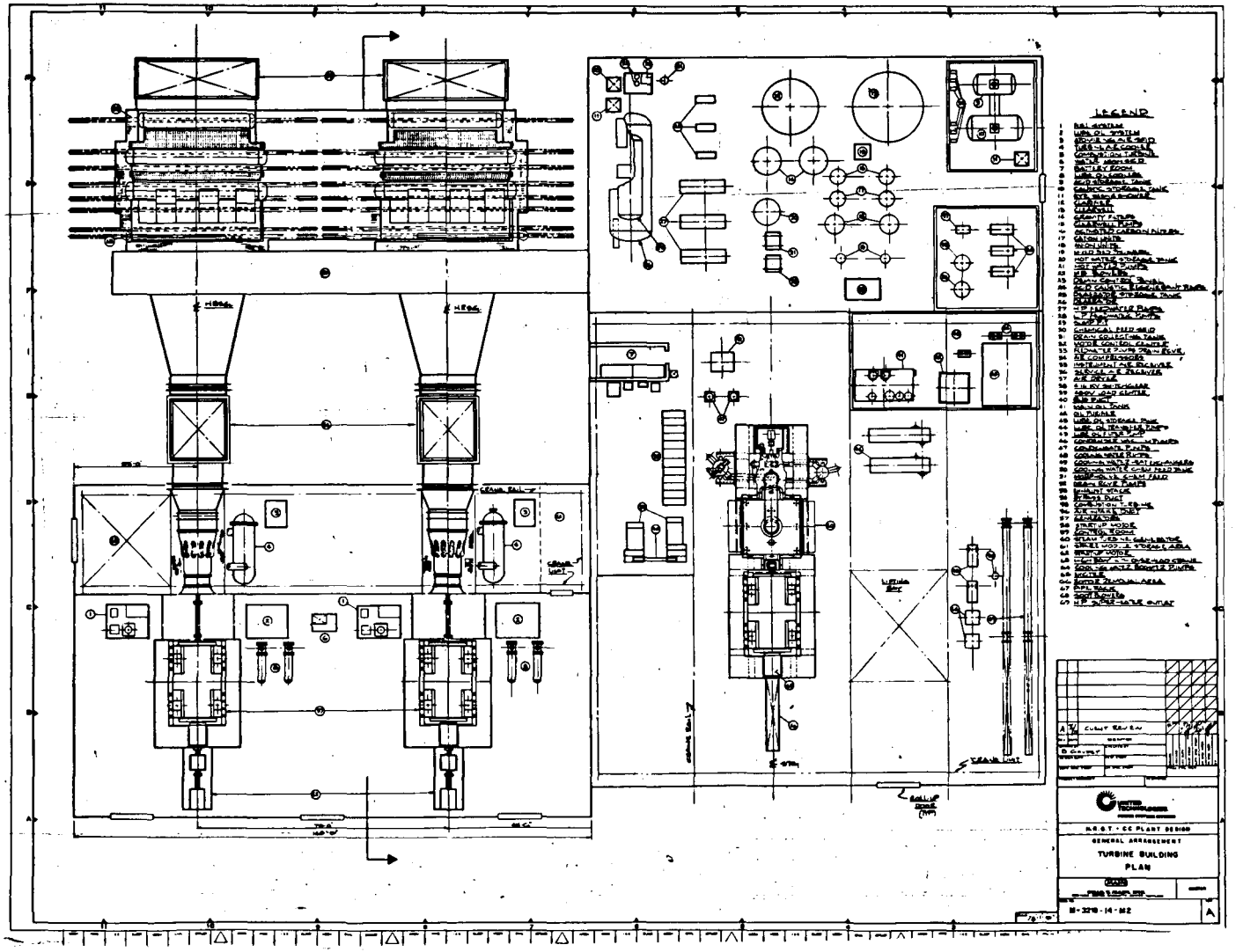


Figure 3-30 Gas Turbine Building - A prime design criteria was to provide facilities for easy maintenance of the system.

Exhaust from each of the gas turbines flows through a bypass damper to a heat recovery steam generator and then exhausts through a stack to the atmosphere. Partitions are used to isolate generators from the turbines and to provide gas turbine noise enclosures. There is sufficient aisle area beside each machine to provide a convenient sheltered "lay down" space when the unit is being maintained.

The several ancillary equipment skids which service the turbomachinery are located in this building to minimize plumbing and electrical line lengths and exposures.

Additions to the basic plant design to accommodate heavier fuels such as No. 6 and certain coal derived liquids, include a fuel treatment system, an atomizing air system to enhance fuel combustion, and soot blowers for the heat recovery steam generators.

The preferred fire system uses Halon as the fire extinguishing agent. The fire system protects not only the gas turbine system, but each of the critical auxiliary subsystems in their packaged configuration. A special air filter and salt monitor are preferred to remove the maximum amount of salt from engine intake air, to reduce contribution to salt sulfidation corrosion and to extend compressor wash intervals in the interest of high availability.

SYSTEM CONTROLS

Introduction

Initial studies of the HRSG controls were based on the combined cycle plant for the Korean Electric Company plant at Ulsan Bay. The Foxboro Company, which is supplying the control system for the KECO plant, contributed as a subcontractor in this study by estimating the reliability of the components in each of the 15 steam plant control loops.

System Description

Control of the combined cycle plant is accomplished from a central control room by the plant operators. The overall system is shown in Figure 3-31.

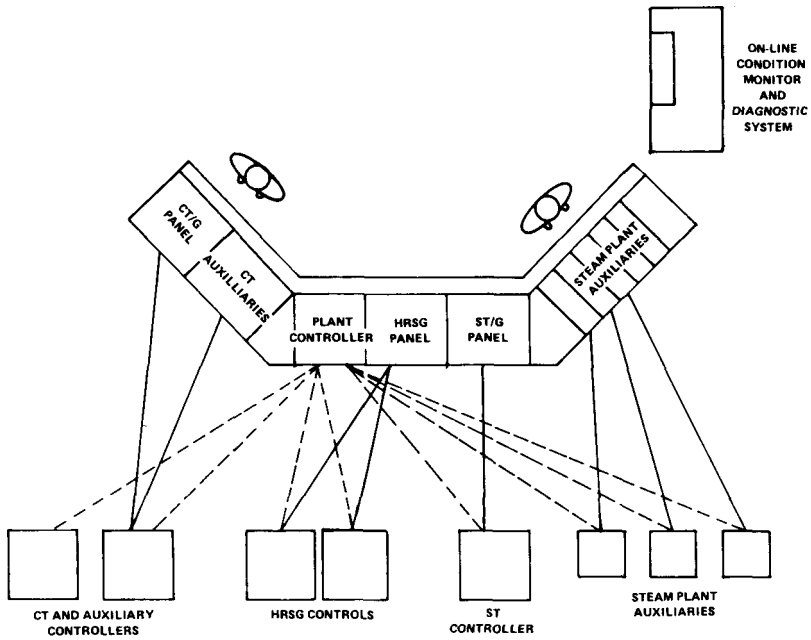


Figure 3-31 System Control Design - High reliability is provided through a combination of electronic automatic control with optional manual take-over.

Four important features have been included in the design to maximize availability.

- Integrated gas turbine, steam system and ancillary controls.
- Condition monitoring and diagnostic prediction system is integrated into the plant design.
- Digital microelectronics supervisory controls with redundancy, in-place replacement of circuit boards and override capability. Manual operation capability for all pieces of the system.

The controller has solid state logic and includes redundant processors for improved reliability. The control concept includes a totally independent protective system to ensure engine safety.

The system has provisions for either manual control by the operators or fully automatic control. This flexibility is accomplished by individual sectional controllers for each of the major parts of the system, i.e., the gas turbines, steam turbine, HRSG's, and steam plant auxiliaries. Each of these sectional controllers has provision for fully automatic operation of its subsystem or manual takeover, permitting operator control including override of all but the most critical shutdown parameters. Sequencing of the sectional controllers in the manual mode is accomplished by the plant operators.

Fully automatic operation is achieved with a solid state supervisory plant controller. This controller communicates the sequence status to the sectional controllers and receives "hand shake" signals to ensure an orderly sequence and plant safety.

All analog inputs are equipped with two-level alarms to allow time for operator intervention prior to automatic shutdown. The controller calculates standardized performance information with display in digital format for instant availability of the engine conditions. The on-line condition monitoring and diagnostic system supplements this display with additional visibility for the gas turbine generator and auxiliaries.

Instrumentation will include guarded wiring to reduce mechanical instrumentation problems and additional features such as redundant air conditioners for control room environment control to ensure life and reliability of the controllers. The control center will also include an event recorder so that control changes and manual procedures are properly documented for operator information in performing extended manual sequencing or operational work arounds.

Reliability

Two basic approaches have been taken to ensure system control reliability. The first is to use reliable system hardware with redundancy.

The reliability of critical sensors is enhanced by redundancy and auto-select features in the controller. Temperature sensors are in triplicate with a device that selects the median. Vibration sensors and monitors are redundant with control logic for alarms and shutdown.

Special features in the boiler controls to improve reliability include:

- Three temperature sensors plus a median select device improving sensor reliability by 76 percent.

- Rate of change lockout which permits replacing a failed logic card without disturbing the controller output.

- Redundant power supplies which improve controller reliability by 35 percent.

However, malfunctions either in the control system circuitry or the sensing or actuation equipment can cause a system trip, which influences the power plant reliability just as if the engine itself had actually failed. To reduce this likelihood, the HRGT control system concept allows for intervention by the plant operator. Human judgement is itself an effective control system, and represents the second important design concept for reliability.

This dual control approach of automatic operation with manual takeover modes is recommended to satisfy the contrasting requirement for rapid startup and consistent procedures versus automatic controller reliability. Automatic operation assures improved availability and reduces the probability of operator errors. Manual control modes and overrides delegate control decisions to the plant operators, and provide an important reliability backup for the plant automatic control sequencer.

In all cases, the operator intervenes to increase the overall operating reliability of the control system. In so doing, he overcomes the major difficulty of control systems: the complex interaction of sensors, computers and actuators. In a fully automatic control, the failure of any one of these elements can cause an outage. With the manual override, a human back-up is available to reduce the number of these outages.

A control which provides for operator intervention is accomplished by the following strategy:

Use primary sensors to activate alarms.

Start a timer from the alarm signal.

Have the operator check back-up sensors and permit shutdown only if indications verify that the abnormal conditions exist.

Activate an automatic trip after a specified time interval if the operator has made no positive response.

The function of the time delay is to ensure absolute safety. Every abnormal condition has associated with it a limited response time beyond which damage can be incurred. This response time varies widely depending on the condition. For example, if the gas turbine exhaust temperature measurement shows that turbine inlet temperature limits have been exceeded by, 50°F, the response of up to an hour might be appropriate in terms of absolute safety. If, however, a combustor can fails to ignite during start-up, raw fuel entering the turbine section could ignite directly on the turbine vanes and cause severe damage in a matter of minutes. In such instances, it is possible to have the operator check the accuracy of the condition prior to shutdown of the engine in response to the alarm signal.

The automatic sequencing of start up and shutdown for normal operation can also be made more reliable by making provision for operator intervention. Visual displays can show the action of the control during automatic operation. The operator would monitor these displays and, if the control malfunctions, the operator can take over manual operation and continue the operation.

Failure Mode Analysis

A failure mode and effect evaluation was conducted on each of the control loops to establish whether failure of a component in the control loop could cause a plant shutdown while the control was in either the automatic mode or the manual mode. In the system failure mode and effect evaluation, it was established which controls could be switched to manual mode by the plant operator and operation of the plant continued manually despite a control loop component failure. Difficulty in diagnosing a failed component in the control system and the time for the operator to react before an automatic plant tripout occurs were considered in the controls evaluation.

Table 3-2 shows typical steam plant control failure rates estimated by Foxboro for the automatic mode and adjusted by Power Systems Division for operation in the manual mode for those controls where the FMEA indicates that is possible.

TABLE 3-2. TYPICAL STEAM PLANT CONTROL FAILURE RATES (NUMBER OF FAILURES PER MILLION OPERATING HOURS)

	Automatic From Foxboro		Adjusted by PSD For Operation in Manual Mode	
Level Control	123	(110)	39	(26)
Pressure Control	35	(22)	25	(12)
Temperature Control	47	(34)*	44	(18.6)*
Flow Control	54	(41)	38	(25)

( ) - with redundant power supply

* with 3 sensors & median select

SECTION 4.0

CURRENT PRODUCT RELIABILITY ASSESSMENT

INTRODUCTION

Before initiating the actual conceptual design of the gas turbine engine and combined cycle plant, it was first important to assess the reliability of our current product. This assessment served to pinpoint problem areas to be avoided, thus taking full advantage of the opportunity to design a new concept from the new centerline. Also, it afforded the opportunity to identify the magnitude of improvements needed for the various gas turbine and plant components in order to meet pre-established reliability/availability goals. In addition, a complete base-of-reference, representing our current product, was established to serve as the basis for trade-off and parametric studies used to make key design selections.

Determining representative overall plant availability for the reference plant required piecing together reliability statistical data from a variety of sources. Specifically, for each plant element we needed:

1. Failure rate data (numbers of failures per operating hour) for determining unscheduled outages;

2. Repair times to determine unscheduled outage hours; and

3. Maintenance intervals to determine scheduled outage hours.

The combination of these factors determines overall plant availability. Our approach for developing representative values for each quantity was quite different for the gas turbine and for the steam bottoming system, engine ancillary equipment, and balance-of-plant auxiliary equipment. This was dictated by the character of the available data.

Establishing the baseline gas turbine reliability data presented a special difficulty. The FT4 engine has been used for and is primarily suited to peaking

duty. As such, its design concept is far removed from a future large (100 MW class) base load gas turbine used in a combined cycle plant. To circumscribe this anomalous situation, we related the available FT4 field experience to an appropriate, already pre-existing design concept, designated "Single-Shaft-Base-Load". To accomplish the connection between the FT4 and Single Shaft Base Load, we used United Technologies' already-developed Reliability Prediction Method (RPM), and we expanded the FT4 field data to include our entire aircraft engine field experience data base.

The gas turbine experience records have been analyzed and correlated by the Reliability Prediction Method system on a part-by-part or component-by-component basis to render the data largely independent of application, thus useful as a basis for predicting new design. Nevertheless, we felt that the link between the FT4 and Single Shaft Base Load was sufficiently tenuous that it was imperative to substantiate and corroborate the method used to establish that link. Validation of the Reliability Prediction Method was included as a special task in the earlier stages of the Phase I program.

The reference combined cycle plant design was based on current FT4 installations, principally our recently completed plant in Ulsan, Korea, operated by the Korean Electric Company (KECO). To develop the total plant, we simply tailored the KECO balance-of-plant equipment to the Single Shaft Base Load gas turbine.

Reliability for the balance-of-plant components was determined through a variety of sources, depending on the type of equipment. This is described in the paragraphs below.

FT4 ENGINE FIELD EXPERIENCE

Field Service Records Summary and Evaluation

The reliability of the GG4C, which is the gas generator portion of the FT4C utility engine, was estimated using field experience data stored in the United Technologies reliability data bank to verify the validity of the Reliability Prediction Model.

Only the gas generator portion of the FT4 was used in this analysis because it was considered more representative of the single-shaft configuration of the

reference SSBL engine and its anticipated high reliability derivatives. In addition, the analysis did not include external accessory or installation components except for fuel manifolds, nozzles, and fuel and oil piping mounted on the engine.

To obtain a statistical sample of reasonable size, data reported over a three-year period for the two latest gas generator models, the GG4C-1D and the GG4C-3F, were used. These data included gas turbine operation in several applications at various power levels with various fuels.

Data of this type is usually supplied by a number of users and repair shops. In the case of the GG4C data base, however, most of the information came from repair shops. Information of maintenance performed with the engine mounted concerning removal, replacement, or repair carried out in conjunction with a hot section inspection is not included in the data base since the user seldom reports this information. Such data would consist of such items as removal and replacement or repair of combustors, fuel manifolds, fuel nozzles, and turbine nozzle guide vanes. However, the absence of these data is offset to some extent by the inclusion of some of the problems which have caused unscheduled maintenance actions over the past three years and have been corrected by retrofit fixes.

In the Reliability Prediction Model analysis of the GG4C, the data described above was reduced and correlated to determine the number of Unscheduled Maintenance Actions per million engine operating hours (UMA) or the Mean Time Between Unscheduled Maintenance Actions (MTBUMA) for the gas generator and its components. An unscheduled maintenance action was defined as any corrective action in response to a failure of the gas generator which requires an unscheduled outage for repair. It also includes any module or complete engine removal and replacement or repair with the engine mounted beyond that normally expected during a scheduled inspection.

The inherent reliability of the GG4C based on the United Technologies reliability data bank information is shown in Figure 4-1. In this figure, the unscheduled maintenance actions are grouped into seven major categories for easy comparison with the predictions to be discussed later. Combining the rates for each of the categories in Figure 4-1 results in an MTBUMA of 5386 hours. This reliability represents the GG4C engine model as a relatively mature model. The

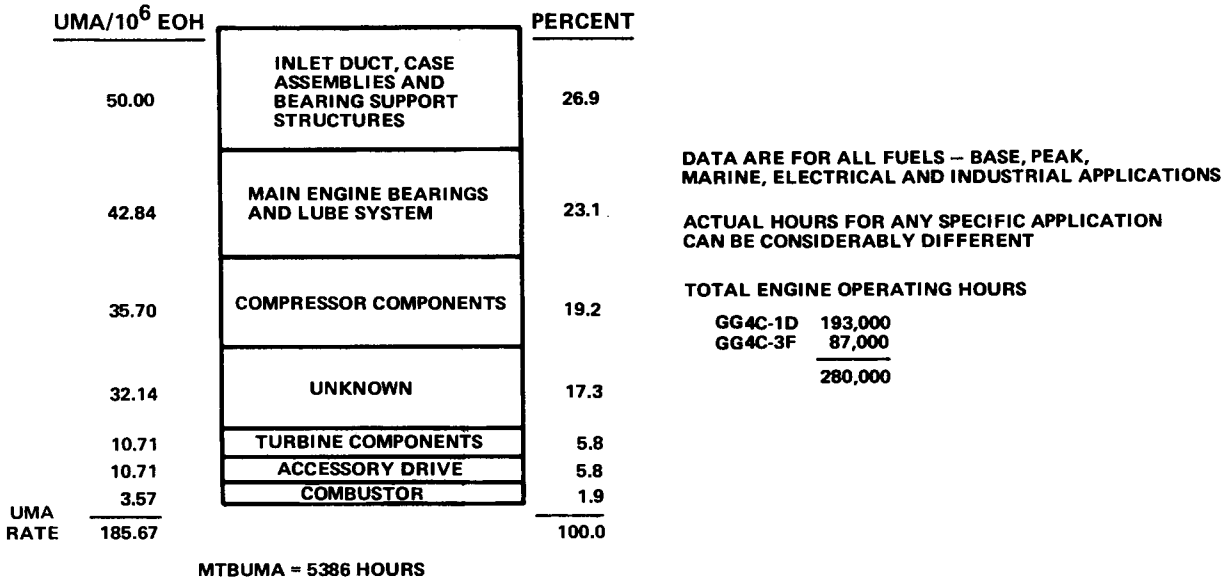


Figure 4-1 Reliability Prediction Model Results for the GG4C - The reliability was predicted using the Reliability Prediction Model based on the GG4 gas generator.

distribution of UMA's into the individual categories may be somewhat uncertain because of the way in which failures have been reported.

The top category in the diagram includes the bearing support structure which is responsible for most of the UMA's in this portion of the engine. These results primarily reflect Number 6 bearing and support rod failures. Those failures, attributed to rigid body sensitivity to small, low rotor imbalance have now been eliminated by incorporation of an oil damped Number 6 bearing.

The next section includes the bearings themselves and the lubrication system. Failures of the Number 2 bearing caused by overload at high power have now been eliminated by a modification to improve the thrust balance of the low compressor. Failures of the Number 3 roller bearing caused by a relatively light load and resultant edge loading of the bearing have been eliminated by the incorporation of a new, fully crowned roller. The only currently troublesome lubrication system problem is oil leakage from the Number 1 bearing housing. Correction of this problem by the use of a stronger spring in the Number 1 carbon seal is presently being evaluated.

In the compressor, the ninth-stage blade has been the main problem. The ninth-stage blade mid-span shrouds tend to wear beyond limits, resulting in a

loose rotor assembly. The worn shrouds no longer provide the required damping. In some instances excessive vibrations cause small pieces of the shrouds to break out. This problem has now been corrected by adopting a tighter mid-span fit at assembly.

The category designated "unknown" includes failures which have, as yet, unidentified primary causes.

Only three failures were found in the data base in the turbine category. One of these was a fracture of the low turbine third stage blade attributed to accelerated shroud notch wear. The wear is probably the result of engine operation at a particular low turbine rotor speed. A program is currently in process to eliminate the cause of the blade excitation which resulted in the observed accelerated notch wear.

In the accessory category, three failures have been reported in the gear box and towershaft. Gear box failures attributed to bearings have been eliminated by incorporation of an improved bearing of new material with better dimensional control. Towershaft bearing failures are attributed to fatigue failure of the bearing cage. A new bearing with a stronger case has been incorporated to eliminate the problem.

Only one combustor failure was found in the data base. The failure rates reported for hot section parts which include fuel manifold, fuel nozzles, burner cans, transition ducts, and first stage turbine vanes are probably unduly low. Failures of hot section parts are often corrected by in-mount maintenance conducted by the user and thus not reflected in the reliability data bank. However, any error in the reliability assessment of the FT4 which stems from this reporting deficiency is compensated for by the improvements which have been incorporated to correct normal problems in the past.

It is quite likely that the current reliability of the GG4C gas generator will prove to be considerably higher than indicated in Figure 4-1 as more field experience with the newly redesigned parts is accumulated. From the analysis it was apparent that several of the GG4C reliability problem areas were associated with the gearbox, the tower shaft, and the scavenge pump. These features were incorporated in the original J75 engine design for application in aircraft. The tower shaft and engine mounted gearbox are needed in an aircraft engine configuration to extract shaft power from the high pressure rotor and transmit

torque at the proper rotational speed to engine mounted accessories such as the fuel pump and alternator. On an industrial engine where weight and compactness are of much less concern, electrically driven floor mounted accessories can be used instead. Similarly, it is not necessary to provide scavenge pumps in the bearing compartments of an industrial engine. In the High Reliability Gas Turbine Engine, the opportunity exists to design these components for easy maintenance and low failure rates without the restrictions imposed by aircraft requirements.

Some GG4 problems were attributable to rotor induced vibration. In any engine design, an attempt is always made to move rotor induced vibration resonances outside the operating range. This is much easier in the case of a single shaft industrial engine used for electrical generation where the rotor speed is fixed, than it is in the case of a two-spool free turbine engine which operates over a wide speed range. Also, in an industrial engine where weight is a minor consideration, stiffer construction of rotor and structural components can reduce the vibrational stress levels.

Utility Operator Site Survey

The objective of the reliability survey was to gather and summarize the experience on equipment supplied by United Technologies Corporation and its suppliers. This information can then be factored into the design of the High Reliability Gas Turbine Combined Cycle powerplant.

Particular emphasis in the survey was directed to the reliability-experience with auxiliary components and controls to supplement the extensive reliability statistics kept by United Technologies Corporation Power Systems Division on the basic gas turbines themselves. Reliability statistics are not maintained on auxiliaries and controls because this equipment is generally serviced directly by the user and/or suppliers.

Three sites were selected for the survey, a peaking station operated by the Long Island Lighting Co. in New York State, a repowered combined cycle plant operated intermediate duty by the City of Glendale, CA, and a base load cogeneration plant operated by Dow Chemical Co. in Pittsburg, CA.

The Long Island Lighting Co., Holbrook Station is a 550 MW plant located at 180 Morris Ave., Holtsville, Long Island. The plant is located about midway on Long

Island between the Long Island Sound and Atlantic Ocean. The Plant includes ten Twin Pacs. Each Twin Pac consists of two FT4C-1 LF gas turbines coupled to a single generator. The base rating of the total station (20 gas turbines and 10 generators) is 550 MW using No. 2 fuel oil. This facility is used by LILCO as a peaking station. Over the last four full calendar years the twenty gas turbines have accumulated 42,000 hours of operations representing a good statistical sample for reliability information.

A total of fourteen people are assigned to this site on a full time basis. Their duties include performing the general maintenance activities such as emergency repairs, scheduled maintenance, general facility inspections and troubleshooting. Even though the gas turbines can be operated remotely from a dispatcher's office, it is normal practice to have the on-site personnel start and operate the units, record the performance data, and shutdown the units.

Maintenance is normally performed by LILCO personnel with contracted help to supplement the regular crew.

The site has four traveling cranes. Other features to assist maintenance are an oil collection system, intercom system, and platforms around filters to aid accessibility. The site now includes an engine maintenance facility that is being expanded to accommodate three engines.

The Reliability Survey at LILCO consisted of analyzing operating information provided by LILCO over the four year period 1975-1978. The data was then reviewed with LILCO management for first hand comments. The PSD survey team then spent several days at the Holbrook site reviewing the plant operating logs and interviewing plant operating personnel.

The Glendale powerplant is a 114 MW repowered combined cycle plant located at 800 Air Way at the City of Glendale, California Power Station. The combined cycle plant consists of two gas turbine generator units and two waste heat boilers providing steam to two steam turbine generators.

One gas turbine generator unit is a United Technologies Corporation Power Systems Twin Pac with two FT4C combustor turbines driving a single generator. The other gas turbine generator unit is a United Technologies Corporation Power Systems Power Pac consisting of a single FT4C gas turbine driving a single generator. The FT4C gas turbines are normally started on natural gas fuel and then switched to No. 2 fuel oil.

The fuel forwarding system includes redundant ac motor driven fuel pumps and a dc motor driven pump for black start. The generators on the gas turbines are air cooled with the air rejecting heat to cooling water from the site cooling towers.

Each gas turbine unit exhausts into a natural-circulation unfired modular steam generator supplied by the Henry Vogt Machine Co.; one boiler for the Power Pac and what amounts to a large double-size one for the Twin Pac. Each boiler contains a 630 psia 825°F superheated HP circuit to drive the steam turbines; a 340 psia saturated IP circuit for NOx injection and for induction into an intermediate stage of the steam turbines (for more power); and a 50 psia saturated low-pressure circuit to deaerate the steam condensate.

The two steam turbines installed in 1941 and 1947 are remnants of the old fossil plant repowered with the gas turbines and waste heat boilers.

This facility is used by the City of Glendale for intermediate duty with a 119 starts per year and a use factor of 35 percent. Since the repowering was completed in June 1977 the three gas turbines in the combined cycle plant have accumulated 16,800 hours.

The combined cycle plant is operated and serviced by the same personnel that operate and service the remaining large fossil units on the site.

The Reliability Survey at Glendale consisted of interviews with the plant superintendent, plant supervisor and shift supervisor and a tour of the powerplant. Further interviews were conducted over the next two days and a copy of the operating logs was obtained for detailed analysis. Reliability statistics were then extracted from the operating logs.

The Dow Chemical powerplant is a 65 MW Cogeneration plant providing electrical power and process steam to the Dow Chemical process plant in Pittsburg, California. Surplus electrical power is sold to Pacific Gas & Electric. The cogeneration plant consists of two FT4C Power Pacs and one FT4A Power Pac. Each Power Pac has an electrical generator and a Struthers-Wells waste heat boiler with auxiliary firing in the duct approaching the boiler.

The gas turbines are operated on natural gas with 25 percent hydrogen from the chemical processing plant.

This plant was initially installed in 1966 and has had a number of modifications through the years for increased power output and improved efficiency. During the last 2 years, which is the basis of the reliability survey, the powerplant has a use factor of 78.2%. During this 2 year period three gas turbines have accumulated 41,100 service hours with an average of 36 starts per year for each engine.

The reliability survey at Dow Chemical consisted of interviews with the plant superintendent, and supervisor, and a tour of the powerplant. Further interviews were conducted over the next two days and a copy of the operating logs for the last two years was obtained for detailed analysis. Reliability statistics were then extracted from the operating logs.

At each site non-inherent shutdowns or unavailable hours were discounted from the reliability statistics. Non-inherent outage time was based on the definition on non-inherent supplied by the EPRI. This is presented in Table 4-1. Several additional causes for outages which PSD felt were non-inherent are also indicated.

TABLE 4-1. EPRI DEFINITION OF INHERENT

INHERENT - When used as a prefix on any of the terms - implies a restriction of that term to failures/outages which are the "fault" of the hardware (e.g., inherent forced outage hours, inherent availability, inherent operating reliability, inherent force rate, inherent equivalent availability, etc.).

The following are exclusions (non-inherent) criteria:

- Acts of God (e.g., lightning strikes)
- Human errors (e.g., closing lube oil valve)
- One time events which will not happen in the future (e.g., warranty period inspections)
- Learning current events (e.g., debugging infant mortality)
- Failure modes which cannot exist in future equipment (e.g., failure/outages associated with components which has been deleted from the design)
- Failure which have been induced by other failures (e.g., when component A shorts out causing component B to fail, component A is charged with a failure, and component B is not)

The results of the reliability survey are presented in Table 4-2 for each of the three sites. The results of the survey confirm analytical reliability predictions for the conceptual design reference SSBL combined cycle plant

described in the following section. The survey showed that the majority of forced outages are due to controls and auxiliaries rather than the gas turbine itself, but that the gas turbine is responsible for the majority of unavailable hours due to scheduled maintenance such as water wash and hot section inspections.

ESTABLISHMENT OF REFERENCE POWERPLANT

Gas Turbine (SSBL)

Description. A conceptual United Technologies engine configuration, the Single Shaft Base Load Gas Turbine Engine, was used as a data base for tradeoff studies leading to the HRGT candidate conceptual designs. The SSBL mechanical and performance features derive from a series of United Technologies-sponsored studies and iterations dating back to 1970. These studies of single spool direct drive and free turbine drive gas turbine engines were focused on a base load engine using more advanced technology than the FT50 peaking/intermediate duty engine. The goal of these previous United Technologies studies was to simplify the mechanical design and to reduce the number of parts, while maintaining a given performance level. The simpler design with reduced parts would then yield greater reliability, reduced initial cost, weight, and maintenance cost.

Key features of the SSBL engine are listed in Table 4-3.

With the SSBL available to serve as a basic reference gas turbine, a complete base load combined cycle plant concept was developed at the beginning of the program.

Recognizing the key role played by the SSBL design background in conducting the Phase I studies, Secton 8 of this report is devoted in its entirety to an historical description of the SSBL. This section primarily describes how the SSBL was employed as a base of reference during the early stages of Phase I.

Prediction Method Description. The current production reliability assessment was based on the broad United Technologies operational and field experience with a variety of gas turbine engines and in particular with the FT4 enigne. Along with this data base, which included aircraft gas turbine reliability statistics broken into separate components, was a developed statistical reliability methodology which correlated the failure rates in such a way that they could be

TABLE 4-2. RESULTS OF SITE RELIABILITY SURVEY

Plant	Lilco		Gendale			DOW		
Location	Holbrook, NY		Glendale, CA			Pittsburg, CA		
Plant Application	Peaking		Intermediate			Base		
Type Plant	Simple Cycle		Reformer Combined Cycle			Cogeneration		
Rating, MW	550		114			65		
Fuel	No. 2		No. 2			Natural Gas & Hydrogen		
No. of Combustion Turbines	20		3			3		
Survey Period, Years	4		2			2		
Use Factor, %	6.6		35.1			78.2		
Hours/Start	6.0		23.6			188.5		
POH/Stand/Year	502.0		149.9			58.4		
	A	B	A	B	C	A	B	C
Availability/ %	93.4	95.3	88.7	96.2	88.9	91.3	96.2	93.0
Reliability Factor, %	99.65	99.82	94.2	99.8	94.3	94.98	98.9	95.6
Unscheduled Outage Rate, %	0.44	0.19	10.2	3.5	9.9	8.4	3.6	6.68
Starting Reliability, %	93/6	98.6	88.5	94.8		89.0	90.2	
Contribution to Number of Forced Unavailabilities, in Percent								
Combustion Turbine	16.4		24			24		
Generator & Electrical	8.2		6			13		
Ct Controls	16.4		7			31		
Other Auxiliaries	59		44			18		
Boilers & Controls	--		19			14		
Contribution to Number of Forced Unavailable Hours, in Percent								
Combustion Turbine	65.6		47.3			48.2		
Generator & Electrical	3.6		0.7			10.1		
Ct Controls	2.4		0.1			2.2		
Other Auxiliaries	28.4		49.3			19.7		
Boilers & Controls	--		2.6			19.8		

A - Survey Period Average
B - Best 25% of Survey Period
C - Survey Period Average Without Waste Heat Boiler

projected into new designs. It remained as a major initial task to verify this technique, known as the Reliability Prediction Model (RPM), as it related to the FT4 electric utility operating experience.

TABLE 4-3. DESIGN FEATURES OF SINGLE SHAFT BASELOAD GAS TURBINE ENGINE

PERFORMANCE	
Combustor Exit Temperature (°F)	2300
Overall Pressure Ratio	14:1
Airflow (1 bm/sec)	840
Power (MW)	100 MW
Heat Rate (Btu/kWh (ISO, HHV))	11,130
MECHANICAL FEATURES	
Front Drive	
Three-Bearing Rotor	
Ten-Stage Compressor	
Can-Annular Combustor	
Three-Stage Turbine	
Modular Construction Compressor Case	
Horizontally Split Burner	
Vertically, Split Turbine Cases	

By using the Reliability Prediction Method, the Single Shaft Base Load engine, installed in a complete combined cycle powerplant, could be assessed from a reliability, maintainability and overall cost-of-electricity standpoint. The results of the SSBL reliability assessment were used in comparison with the original program goals as the means for selection criteria in the trade-off studies which followed.

The data base and statistical methods used to predict reliability were established by the Gas Turbine Operations of Power Systems Division, responsible for the FT4 engine, in cooperation with the Government Products Division of the Pratt & Whitney Aircraft Group.

The FT4 has undergone extensive modification for the utility market. In its configuration for electric utility service, the FT4 has seen wide application as an electric utility prime mover for peak load duty, as well as service in combined cycle power plants.

The approach to defining gas turbine reliability involved combining this FT4 field experience with the operational experience of all United Technologies' gas turbines including extensive commercial and military aircraft engine experience. This total experience was then applied to a new conceptual engine design expressly intended for base load, combined cycle service.

Connecting this Single Shaft Base Load engine design concept with reliability service experience was accomplished by use of the Reliability Prediction Model - a statistical method developed at United Technologies as one of its basic reliability analysis techniques.

The gas turbine reliability analysis was conducted in four major steps:

- FT4 Gas Generator (designated GG4) Reliability Analysis
- Verification of the Gas Turbine Reliability Prediction Model
- Estimates of the SSBL Engine Reliability and Maintainability
- Gas Turbine Starting Reliability

The Reliability Prediction Model developed and used by United Technologies for the prediction of the reliability of new gas turbine designs is based on the use of field failure rate data accumulated on a number of operational gas turbine engines. This information is organized in a part-by-part breakdown and grouped into categories associated with functional sections of the engine. The levels of the breakdowns in these functional categories are detailed enough to enable assessment of engineering features and comprehensive enough to provide statistical accuracy in the documented base failure rate.

The Reliability Prediction Model makes use of the information in the data base by selecting parts which resemble as closely as possible those parts in the new design and then adjusting the data base failure rates for these selected parts. These adjustments account for differences in the design, operational conditions, and the maintenance approach which exist between the data base parts and the parts in the new engine design.

This is accomplished by multiplying the data base failure rate or frequency by a series of so-called K factors, which are estimated from the differences between the data base engine and the new design. The K factors used in the Reliability Prediction Model are the mission K factor, the design K factor, and the maintenance K factor.

The approach to determining the Duty Cycle K Factor is the mission K factor illustrated in Figure 4-2. The Duty Cycle K Factor is a function of three demand factors: (1) the high stress factor; (2) the cycle factor; and (3) the thermal fatigue factor. These separate factors are defined in Figure 4-2.

$$K_{\text{MISSION OR DUTY CYCLE}} = f\,(K_{HS}, K_C, K_{TF})$$

DEMAND FACTORS

$$K_{HS} = K_{\text{HIGH STRESS}} = \frac{(\%\ \text{TIME AT HIGH STRESS})_{\text{STUDY ENGINE}}}{(\%\ \text{TIME AT HIGH STRESS})_{\text{BASE ENGINE}}}$$

$$K_C = K_{\text{CYCLES}} = \frac{(\text{TYPE I} + 0.25\ \text{TYPE III/HR})_{\text{STUDY ENGINE}}}{(\text{TYPE I} + 0.25\ \text{TYPE III/HR})_{\text{BASE ENGINE}}}$$

$$K_{TF} = K_{\text{THERMAL FATIGUE}} = \frac{(\text{MISSIONS/HR})_{\text{STUDY ENGINE}}}{(\text{MISSIONS/HR})_{\text{BASE ENGINE}}}$$

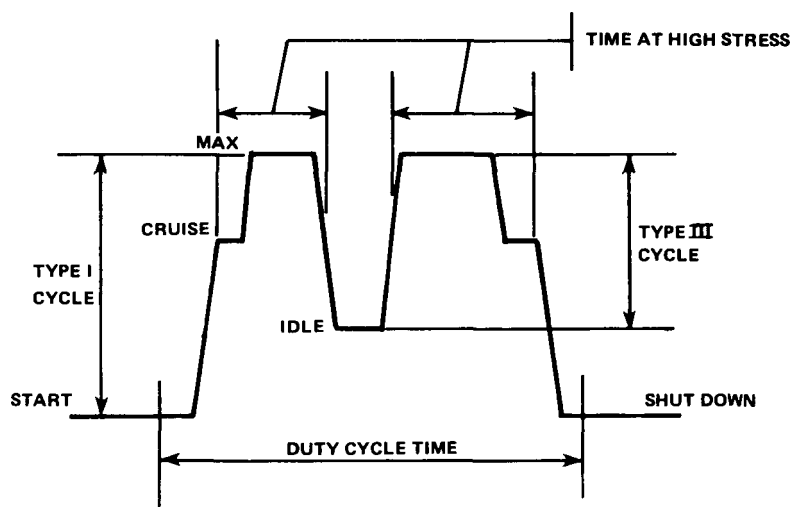


Figure 4-2 Demand Factors Used to Calculated Mission or Duty Cycle Factor, K - K factors are used in the Reliability Preduction Model to adjust for differences between the model base and the system under analysis.

The high stress demand factor is determined by dividing total time spent by the study engine at high power by the time spent at high power for the data base

engine. This factor comes into play when ultimate yield, or creep failure modes are involved.

The cycle demand factor is expressed as the ratio of the number of Type I power cycles (defined in Figure 4-2) plus 1/4 of the number of Type II power cycles occurring per hour for the new design to the same function of Type I and Type II cycles for the data base engine. The thermal fatigue factor is expressed as the ratio of the number of missions per hour for the new engine design to the number of missions per hour for the selected data base engine. Cyclic and thermal fatigue factors are used where both fatigue failure modes are involved.

The Design K Factor accounts for the effect of differences in major engine design characteristics on engine failure rate between the new design and the data base engine. These differing characteristics might include critical design parameters such as rotor speed and turbine inlet temperature, or degree of design complexity as would be represented by mechanical configuration and parts count, design margin, materials and system redundancy.

The Maintenance K Factor accounts for the differences in the maintenance practices, failure definitions and support equipment of the new design engine relative to the data base engine. Maintenance practice differences can influence reliability by detecting an incipient failure before it becomes catastropic. Some of the items used in determining the Maintenance K Factor include part accessibility for repair or replacement, maintenance philosophy and schedules, availability of test and inspection equipment, and component exposure to visual inspection.

To test the accuracy of the Reliability Prediction Model for determining the reliability of an industrial gas turbine, the model was used to predict the reliability of the GG4C gas generator from the FT4C using United Technologies' engine reliability data base.

The reliability of the FT4 engine was first determined and analyzed from service records. Not only was statistical failure rate data analyzed, but also the types of failures were evaluated as a guide for the design of a Highly Reliable Gas Turbine Engine. Data from each of six different engines was used independently to make a prediction of the GG4 reliability. The technology represented by these six engines spans more than two decades and includes United

Technologies' most advanced production engine. The accumulated operating experience on these engines totals more than 277 million hours.

The GG4C reliability was estimated by identifying parts of each data base engine similar or corresponding to each of the important parts of the GG4 and adjusting the corresponding data base part failure rate by the appropriate K factors to obtain the GG4C part failure rate. The GG4C engine failure rate was then obtained by combining the adjusted failure rates of all of the component parts by summing the failure frequencies.

A typical work sheet for one part is shown in Figure 4-3. The figure illustrates how the base failure rate is adjusted by the various factors to predict the reliability of the GG4 engine roller bearings. The first line of the worksheet identifies the bearing in the data base Engine A which was judged most similar to the No. 1 bearing in the GG4C. The three factors that were used to adjust the base rate are also shown and the product of these K factors which is used as the multiplier for the base failure rate is given in the GG4C estimate column. The Mission Factor is greater than 1 since the GG4C spends a larger percent of its operating time at high stress than engine A. The Design Factor is less than 1 since Engine A operates at a higher rotor speed and loading than the GG4C. The Maintenance Factor is less than 1 because of the difference in the maintenance actions counted in the data base engine relative to the GG4. In this case, the base engine was charged for one unscheduled maintenance action every time a Number 1 bearing caused a module removal and replacement. On the other hand, unscheduled maintenance of the GG4 Number 1 bearing is usually carried out by the utility operator "in mount" and as such is not reported as an unscheduled maintenance action in the GG4 data base. This sort of procedure was carried out for the complete set of some ten worksheets representing the total component breakdown for the engine.

Prediction Method Corroboration. The Reliability Prediction Model served as the keystone to the statistical reliability analyses. Using the Reliability Prediction Model, a rationale was developed to provide the bridge between the current FT4 used in peaking applications and an all-new 100 MW class design for future base load application. The Reliability Prediction Model as a basis for analyzing and projecting reliability (failure rates) for new engine designs was already in existence and had been extensively used by United Technologies. As such, a wealth of basic component failure rates was available in the Reliability

ROLLER BEARINGS		BASE RATE /10^6 HRS.	K MISSION	K DESIGN	K MAIN-TENANCE	GG4C ESTIMATE /10^6 HRS.	REMARKS
ENGINE A	GG4C						
No. 1 BRG	No. 1 BRG	BASE	1.1	0.25	0.8	0.23 BASE	HIGH LOAD & RPM CAUSE FAILURE. LESS SEVERE IN GG4C THAN IN ENGINE A.
No. 1 BRG	No. 3 BRG	BASE	1.1	0.1	1.0	0.10 BASE	GG4C No. 3 BRG HAS LESS ROLLER SPEED COMPARED TO ENGINE A No. 1 BRG. USE DESIGN FACTOR 0.1.
No. 4 BRG	No. 5 BRG	BASE	1.1	0.5	1.0	0.51 BASE	LESS SEVERE DESIGN IN GG4C. HOT ENV, HIGH POWER, RPM ALL CAUSE FAILURES. B10 LIFE 100% GREATER IN GG4C.
No. 5. BRG	No. 6 BRG	BASE	1.1	0.2	0.8	0.09 BASE	B10 LIFE RATIO ~ 0.16. LESS SEVERE DESIGN IN GG4C.

Figure 4-3 Example of GG4C Component Failure Rate Estimate Logic - The system uses K factors to adjust the estimates for design and application differences from the baseline.

Prediction Model data base. It remained in the High Reliability Gas Turbine conceptual design study to first verify the validity of the model by applying the various factors to operating experience from several in-service engines and then comparing the results with independently collected FT4 field experience.

The technique used to validate the method consisted basically of first using the reliability prediction method to predict the theoretical reliability of the FT4 engine, and then comparing the theoretical prediction with actual field experience. This was done for six different cases, starting with a different set of components taken from the complete data base in each instance. To best relate the true engine experience, each component set represented a set from a specific real engine. The technology represented by these six engines spans more than two decades and includes United Technologies' most advanced production engine. The accumulated operating experience on these engines totals more than 277 million hours. The reliability of the FT4 engine was then determined and analyzed from service records.

The GG4C reliability was estimated from the six other engines by identifying parts of each data base engine similar or corresponding to each of the important parts of the GG4 and adjusting the corresponding data base part failure rate by the appropriate K factors to obtain the GG4C part failure rate. The GG4C engine failure rate was then obtained by combining the adjusted failure rates of all of the component parts by summing the failure frequencies.

Typical values for the duty cycle factors are given for the six data base engines in Table 4-4. (The engines are identified by the letter code A through E to protect United Technologies Corporation's priority rights while at the same time allowing for reporting completeness.) Note that the GG4 spends more time at high stress than Engines A and E since these engines operate most of the time at part power. An examination of the cycle factors indicates that the GG4 experiences more cycling duty than Engines B and F, which operate over long periods of time at constant power with little or no cycling. The same situation is also reflected in the thermal fatigue factors.

TABLE 4-4. TYPICAL VALUES OF DEMAND FACTORS FOR PREDICTING GG4C RELIABILITY

Demand Factors	Engine A	Engine B	Engine C	Engine D	Engine E	Engine F	GG4C
High Stress Factor - K_{HS}	1.3	0.9	1.0	1.0	1.3	0.9	1
Cycle Factor - K_C	0.3	1.6	0.3	0.4	0.2	1.3	1
Thermal Fatigue - K_{TF}	0.8	1.6	1.0	0.4	0.6	1.3	1

The results of the prediction of the GG4 reliability using data from each of the six data base engines are shown in Figure 4-4. The GG4 field data for each of the five engine component categories is indicated as 100% in the right hand bar.

It was recognized early in the process of preparing the predictions that the data base for Engine E was of very poor quality. Engine E is the oldest of the engines and the early field data collected on this engine is highly suspect. The results obtained are shown here to emphasize the fact that the quality of the field data upon which the Reliability Prediction Model is based will of course determine the quality of the prediction to be expected.

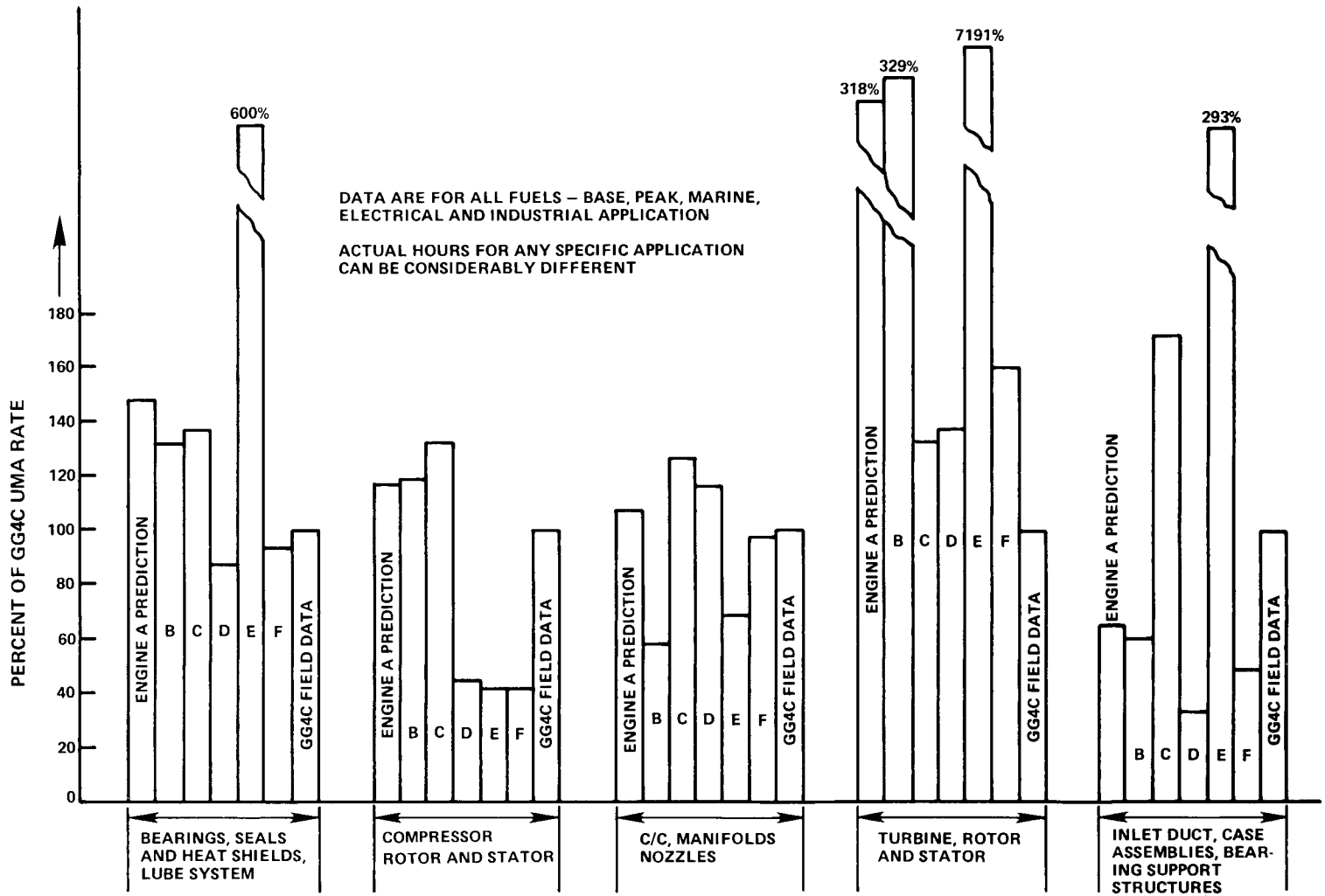


Figure 4-4 GG4C Reliability

The turbine section reliability predictions based on data base Engines A and B illustrate another point related to the use of the Reliability Prediction Model. In the case of these two engines, the turbine design is much more sophisticated than the design of the turbine in the GG4 and these turbines have little similarity to the GG4 turbine. As stated earlier, the effectiveness of the Reliability Prediction Model depends upon selecting components from the data base which are quite similar to the engine components to be evaluated. The greater the physical difference between the data base component and the component to be evaluated, the greater the probable error in the prediction. In the cases of Engines A and B, the design differences are so great that the estimation of a meaningful design K factor is very difficult and the method breaks down.

If data from Engine E are ignored and the Engine A and B turbine predictions are eliminated, the agreement of the prediction of the Reliability Prediction Model with the GG4C field data is remarkable and the method appears to be more than adequate for the purpose of estimating the reliability of the SSBL engine for use in the trade studies.

SSBL Reliability Prediction. The estimated reliability of the SSBL engine is required as the reliability baseline for the tradeoff studies. The Reliability Prediction Model was used to predict the reliability of the SSBL reference engine as shown in Figure 4-5 using the United Technologies reliability data base of actual field information.

The first step of the analysis was to divide the SSBL engine configuration into components, subassemblies, and parts to the same level of detail as the GG4C in the prediction model assessment. Components having the greatest similarity to the SSBL engine components were selected from the data base to provide the best baseline component or part failure rates.

These rates then were adjusted by the procedures described earlier using the adjustment K factors for duty cycle, design and maintenance. The individual component failure rates expressed as unscheduled maintenance action (UMA) rates calculated by the Reliability Prediction Model were then summed into a total gas turbine UMA rate per million hours. The mean time between unscheduled maintenance actions or MTBUMA was calculated as the reciprocal of the UMA rate.

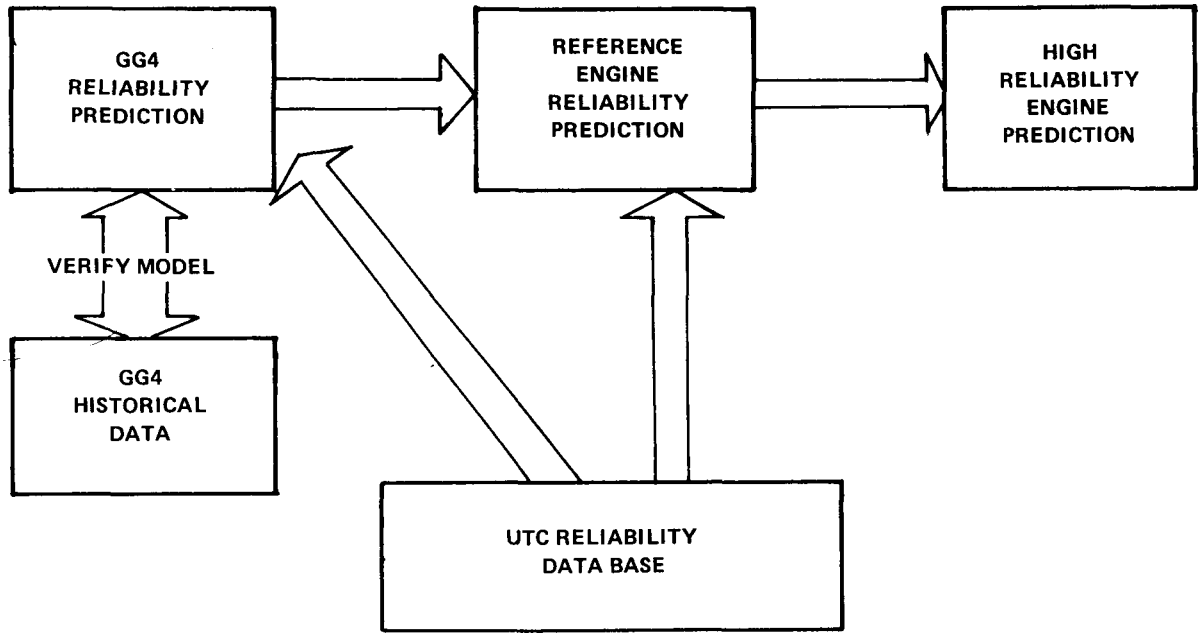


Figure 4-5 Gas Turbine Reliability Analysis Flowchart - The Reliability Prediction Model was used to predicted the reliability of the SSBL reference engine.

The results of this prediction are displayed in the right hand bar chart on Figure 4-6 and these results are compared to the results of the GG4C prediction. The GG4C historical reliability data presented in Figure 4-1 has been redistributed in Figure 4-6 by distributing the "unknown" category UMA's evenly among the five major categories. No unknown UMA's were assigned to the accessories category. This permits a more meaningful comparison of the GG4C data with the SSBL engine prediction.

This comparison is of somewhat limited validity because the categories of failures charged to the engine are not the same for the GG4C and the SSBL engines. The SSBL prediction is intended for use in an overall combined cycle powerplant reliability analysis and includes two "non-engine chargeable" UMA categories identified in Figure 4-6 as "nonchargeable" and "miscellaneous". These categories include events such as foreign object damage, mount vibration, and assembly errors such as misrigging of the variable geometry system. On the other hand, accessories are not included in the SSBL engine prediction because they are not engine mounted and are considered separate from a reliability analysis standpoint.

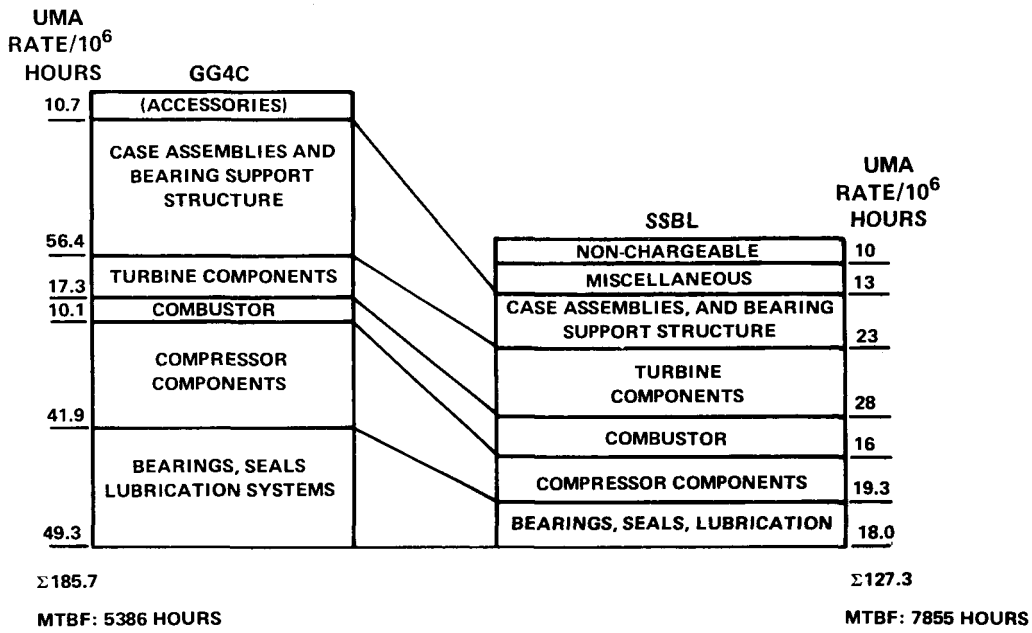


Figure 4-6 SSBL Reliability Prediction - The SSBL provides substantially higher reliability than the baseline GG4C.

However, some valid differences between the two engines are apparent in Figure 4-6 and are explained as follows. First, the case assemblies and bearing support structures of the SSBL engine are predicted to be substantially more reliable than those of the GG4C. This reflects a significant reduction in rotor induced vibration resulting from the use of a welded drum rotor, a simply supported turbine and more rigid cases.

The apparent higher combustor and turbine UMA rates shown for the SSBL engine relative to the GG4C can be explained by the fact that in-mount UMA's are not usually reported for the GG4C as previously discussed. Unreported actions include removal or replacement or repair of combustor cans, transition section, fuel manifolds, fuel nozzles and first turbine nozzle guide vanes.

The compressor components UMA rate for the SSBL engine is substantially lower than for the GG4C as a result of the previously mentioned lower rotor induced vibration level and the use of lower aspect ratio blading.

The improvement in SSBL engine bearing, seal, heatshield, and lubrication system reliability stems from an industrial design approach and the use of fewer

parts. The simple design and low rotor speed of the SSBL led to the selection of long life tilting pad hydrodynamic bearings. In addition, the requirement for fewer seals and the use of labyrinth seals in place of carbon seals for the bearing compartments reduces the UMA rate. As discussed earlier, the SSBL lubrication system is designed in accord with the best industrial practice. Most of the components of the system including the supply and scavenge pumps are floor or skid mounted in a location removed from the engine where the failure rate is reduced by lower environmental temperature and vibration levels, and, where maintenance is easier.

Summing the UMA rates for the several SSBL component categories yields an overall engine UMA rate of 127.3 per million operating hours. This is equivalent to the MTBUMA of 7855 hours. This value is used as input to the overall plant reliability estimate.

SSBL Maintainability Prediction. The maintainability requirements for the SSBL engine were established by developing a maintenance philosophy based on GG4C experience, aircraft engine experience and work done in support of the United Technologies FT50 large industrial engine program. Applying this philosophy to the conceptual design of the baseline powerplant and making use of aviation engine and GG4C experience as well as results of the FT50 maintenance studies, task time estimates were made for SSBL preventive and corrective maintenance. Preventive maintenance time estimates were made for inspections and water wash. Corrective maintenance task time estimates were made for module and engine removal and replacement and for in-mount component removal, replacement or repair. Both sets of estimates include cool down times. Assumptions were made for the on site maintenance equipment, component accessibility, spare parts, and maintenance crew expertise. Operation on Number 2 distillate fuel was assumed.

A list of the corrective maintenance developed for the SSBL design is listed in Table 4-5. This maintenance includes removal and replacement of the five modules, removal and replacement of the engine and removal, replacement or repair of components on-site while the engine remains installed. The table shows the maintenance times estimated for the SSBL engine in each of these categories. All of the corrective maintenance task elapsed times shown include coastdown and cool down times. These times were derived from a detailed analysis of the tasks required. The times for performing each step of the replacement or repair tasks were computed using GG4C experience and results of

the FT50 maintenance time estimates. The results of the SSBL engine reliability and maintainability predictions are summarized in Table 4-6. The table shows the UMA rates distributed into the sections of the engine where they occur and into the type of maintenance (module, engine or in-mount) that will be done to correct the problem.

TABLE 4-5. SSBL PREDICTED MAINTENANCE TIMES

Corrective Maintenance	Interval	Manhours	Elapsed Hours
REMOVE AND REPLACE			
Engine (Engine Split at Compressor/Diffuser Plane)	As Required	248	90
Inlet Case Module	As Required	62	45
Compressor Module	As Required	143	71
Diffuser Case Module	As Required	162	78
Turbine Module	As Required	121	63
Turbine Exhaust Case Module	As Required	96	54
IN MOUNT REPAIR			
Replace No. 1 Bearing	As Required	35	27
No. 1 Bearing Seal - Front	As Required	23	19
Replace No. 3 Bearing	As Required	53	32
Replace No. 3 Bearing Rear Seal	As Required	38	22
No. 2 Bearing	As Required	146	74
No. 2 Front Outer Airseal	As Required	132	69
No. 2 Bearing Front Oil Seal	As Required	134	70
No. 2 Bearing Rear Oil Seal	As Required	136	69
No. 2 Rear Outer Air Seal	As Required	134	68
Combustion Cans (Assumed 8)	As Required	213	112
Transition Sections (Assumed 8)	As Required	205	108
Turbine Nozzle Guide Vanes (30)	As Required	142	72
Replace Fuel Manifolds and Fuel Nozzles	As Required	64	42
Repair Fluid Line Leaks	As Required	12	12
Compressor Blades (Stages R_6, R_7, R_8)	As Required	159	84
Assembly Errors	As Required	13	13

At the bottom of Table 4-6, the Unschedule Maintenance Action (UMA) rate and Mean Corrective Time (MCT) are summarized for each type of maintenance. All of the MCT estimates include coastdown and cooldown times for one-half the maintenance actions, assuming the remaining one-half are done in conjunction with scheduled maintenance actions.

TABLE 4-6. SSBL RELIABILITY/MAINTAINABILITY PREDICTIONS

	MODULES					ENGINE	INMOUNT
	Inlet	Comp	Diffuser	Turbine	Exh Case		
	(UMA's per Million Engine Operating Hours)						
NON CHARGEABLE							
Comp FOD	1.5	4.0					
Fire/Expln (Stand)		0.5		2.0		1.0	3.0
Excess Mount Vib						1.0	
MISCELLANEOUS							
Vib Unbalance	0.2	2.0	1.0	2.0	1.3		
Assy & Manuf Error	0.3	1.0	0.4	0.9	0.3	0.6	3.0
MAIN STATIC STRUCTURE							
Inlet Case	3.0						
Comp Case		3.0					
Diff Case			3.0				
C/C & Trans Duct			3.0				
Turb Case				8.0			
Turb Exh Case					3.0		
TURBINE							
Blades				7.0			
V & S				3.0			4.0
Outer Rings				4.0			
Misc & Rotor				4.5			
Spacers				2.5			
Cover Plates				3.0			
COMBUSTORS							
Cans/Transition Sec.							7.0
Fuel Nozzles							5.0
Manifolds							4.0
COMPRESSORS							
Blades		5.0				5.0	
V & S		3.5				2.3	
Misc & Rotor		3.0					
BRGS, SEALS AND LUBE							
Bearings	0.3			0.4	0.3		2.0
Seals	0.5	0.5		0.5	0.5		4.0
Heat Shield				0.8			
Lub Sys	0.2	0.2	0.2	0.2	0.2		2.0
Ex Oil Consumption	0.5		1.0	1.7	0.5		1.5
Sum Total UMA's/10^6 EOH	6.5	22.7	8.6	40.5	6.1	2.5	40.3
MTBUMA (EOH)	153846.	44053.	116279.	24691.	163934.	384615.	24814.
MCT (Elapsed Time in Hrs)	40.4	66.4	73.1	58.0	48.0	90.2	52.3
Overall Engine Total UMA's/10^6 EOH	127.3						
MTBUMA TOTAL ENGINE	7855						

Use of this information along with the scheduled preventive maintenance time estimates in Table 4-5 established the engine portion of the original plant reliability calculation. Analysis of these results, presented in Section 6, led to design strategies in the conceptual design process which ultimately resulted in the final selected configuration.

REFERENCE PLANT DESCRIPTION AND RELIABILITY SOURCES

The reference plant established around the SSBL engine was a 282 MW combined cycle plant shown schematically in Figure 4-7. The plant consists of two

100-megawatt SSBL gas turbine generators. The exhaust from each SSBL gas turbine is ducted to a waste heat steam generator. Steam from the two waste heat steam generators is piped to a single 86 megawatt steam turbine generator. The condenser associated with the steam generator is cooled by a closed circulating fresh water loop which rejects heat to the atmosphere in a mechanical draft wet cooling tower. The overall plant produces 286 megawatts of net plant output. The reference plant heat rate is 10,350 Btu/kWH (ISO) in the simple cycle mode and 7350 BTU/kWH when operated as a combined cycle plant.

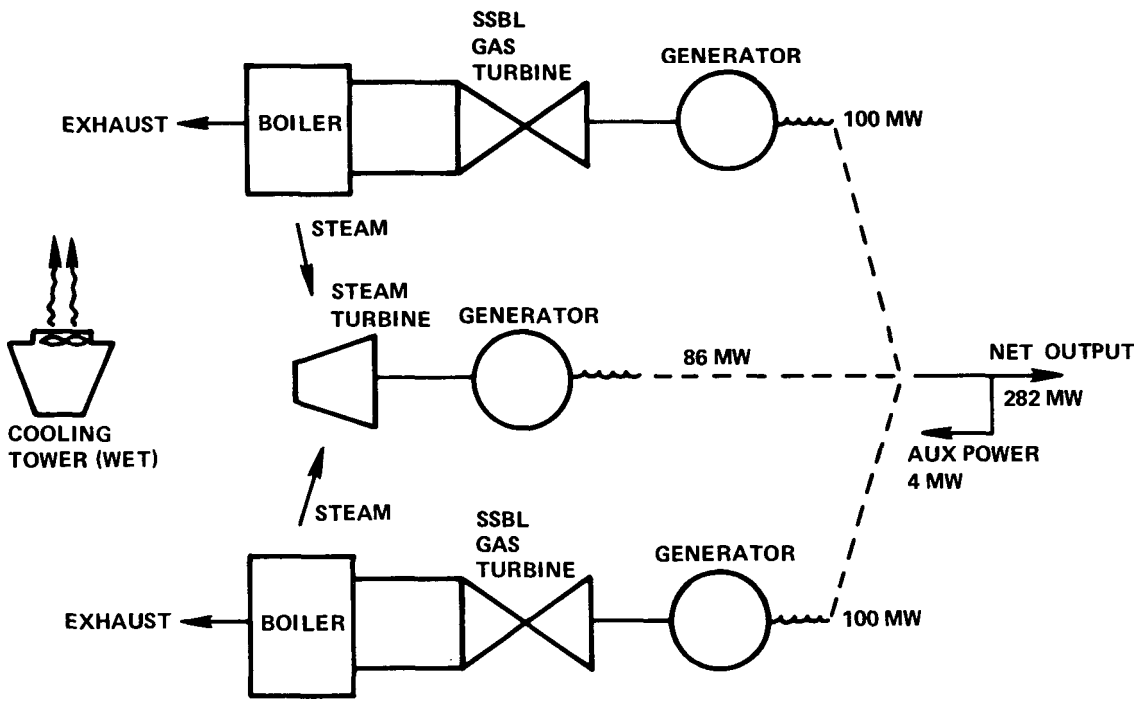


Figure 4-7 Combined Cycle Reference Plant with SSBL Engine - The SSBL Engine plant was used as a data base for tradeoff studies leading to the HRGT candidate conceptual designs.

Table 4-7 shows the overall combined cycle plant performance at ISO conditions. Heat rate is based on fuel lower heating value with higher heating value shown in parenthesis.

An arrangement plan for the overall SSBL combined cycle plant prepared by Chas. T. Main is shown in Figure 4-8. Included are the two enclosures for the SSBL

TABLE 4-7. PLANT PERFORMANCE

	ISO	NEMA
Gas Turbine Output, MW	100	90
Overall Plant Output, MW	282	256
Heat Rate Simple Cycle, BTU/kWh (HHV)	10350 (11,130)	10550 (11,340)
Heat Rate Combined Cycle BTU/kWh (HHV)	7350 (7,890)	7550 (8,040)

gas turbines with their generators and auxiliary systems, the bypass dampers, steam generators, the steam turbine building, the deaerator and feedwater pump enclosure, the cooling tower, buildings for the circulating water pumps, and the step up transformers. The plant using two SSBL gas turbines occupies about 4.7 acres exclusive of the cooling tower.

SSBL Auxiliary and Ancillary Systems

The recommended generator design from Brush Electrical Machines Ltd. is a model BDAX 9.45 air cooled generator with an efficiency of 98%. Brush has 9 of these units presently under construction.

The generator is cooled by a closed circulating air loop which has reliability benefits particularly in installations where there are extreme dust conditions. In normal environmental conditions an open air cooled generator taking outside. air through a filtering system can be used with no anticipated problems. This is the type of air cooled generator generally used in FT4 installations.

Reliability information was provided by Westinghouse and Brush on automatic synchronizing units for the gas turbine generators.

Reliability of the power control and voltage regulator is based on EEI data for these types of controls on gas turbine engine generators.

Reliability of air cooled generators for the study is based on air cooled generator experience at both Brush and Power Systems Division with 30 to 70 MW generators. Generator failures were separated into major and minor failures. The major failures covered experience from 82 operational units over a total of 2.8

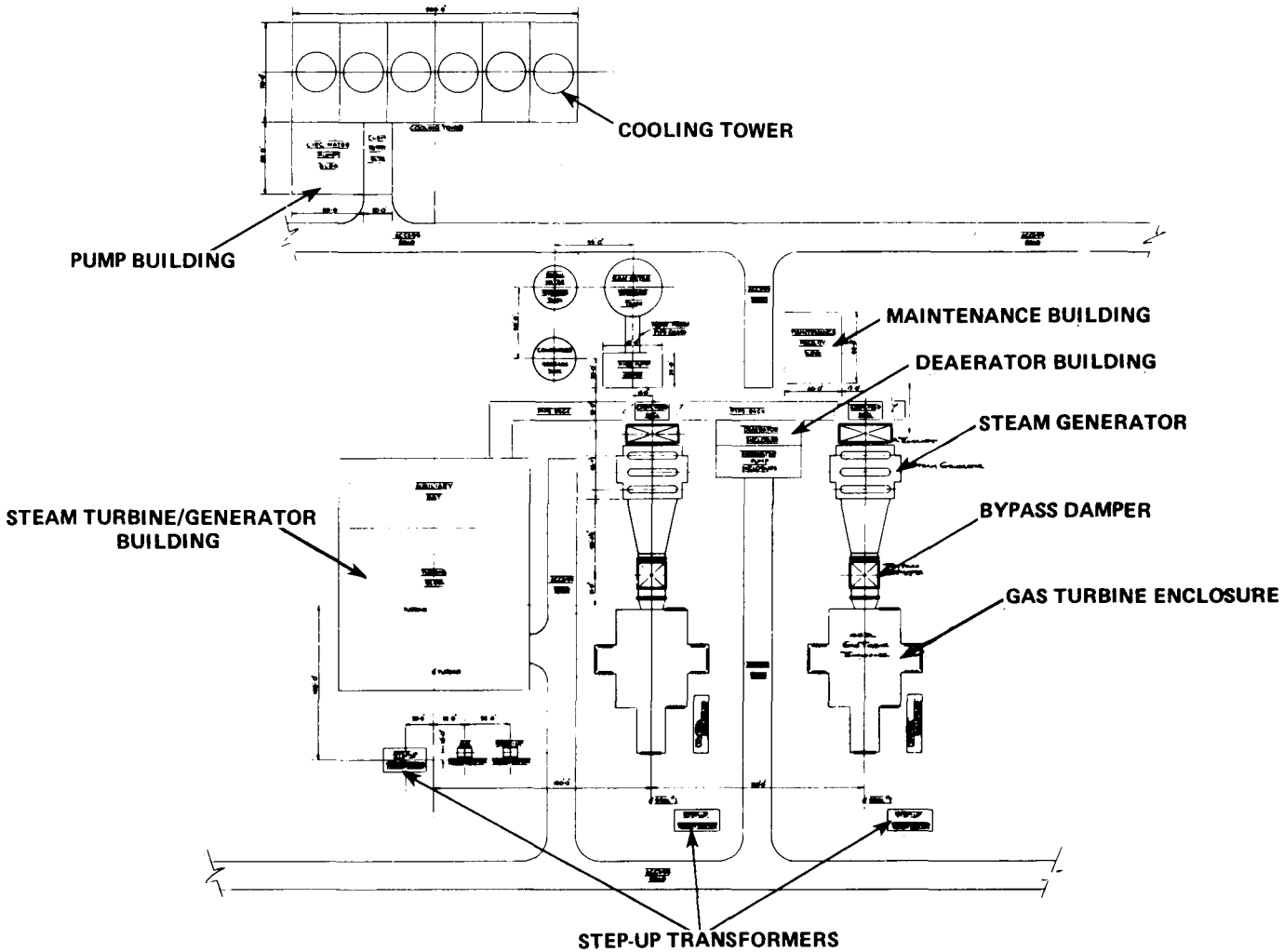


Figure 4-8 SSBL Plant Arrangement

million service hours resulting in a MTBF of 933,000 hours and an MCT of 960 hours. Minor failures in generators are based on operating experience with 22 aircooled generators covering 48,000 hours of operation resulting in a MTBF for minor failures of 9600 hours and a mean corrective time to repair of 96 hours. EEI data for jet engine generators shows an MTBF of 222,000 hours and an MCT of 227 hours overall, with no distinction between major and minor types of failures.

Gas turbine controls include a relay-actuated sequencer which provides all of the sequencing of the components for startup, normal operation, and equipment monitoring for alarms and shutdowns. The control and instrumentation also include an electronic solid state fuel control, a vibration monitor, and numerous sensors throughout the subsystem.

Reliability for the fuel control supplied by Hamilton Standard Division of United Technologies Corporation was quoted by the supplier as a mean time between failure of 25,000 hours.

As reliability information was not available on the relay-actuated sequencer, a failure rate was estimated using schematics of the sequencer and applying generic reliability data to the relays and switches.

A reference starting system concept for the SSBL engine selected was an electric motor and torque converter mounted on the end of the generator shaft. Direct air start through the compressor such as used in the FT50 was ruled out for the reference design because the SSBL engine has a front-end-mounted generator and an adaptation of the cone and ejector air start design for the FT50 would have required a complex seal around the generator drive shaft. Start system components were not included with the gas turbine auxiliaries in the estimate of running reliability since these components are inactive during normal operation.

The ancillary systems for the SSBL gas turbine are summarized in Table 4-8. Failure mode and effect evaluation of each of the gas turbine ancillary subsystems was conducted to determine if failures in any of the components in these subsystems would result in a gas turbine shutdown. Table 4-8 shows which of the systems are not critical. In the non-critical systems a failure would not shut the plant down. The critical systems for plant shutdown are the lubrication oil system, fuel system, and hydraulic system which operates critical controls in the fuel system.

TABLE 4-8. GAS TURBINE AUXILIARY SYSTEMS

CRITICAL	NON-CRITICAL[1]
Lube Oil	Water Wash
Fuel	Engine Heat
Hydraulic	Ignition
	Fire (CO_2)
	Start

[1] For operating reliability

Reliability information on specific components within these gas turbine auxiliary systems was not available, so generic failure rate information based on the nuclear plant reliability data system and IEEE standard 500 was used as a basis for reliability estimates. Table 4-9 is a summary of the generic failure rates used to estimate reliability of components in these subsystems.

SSBL - Balance of Plant

The balance of plant includes the steam plant including the waste heat steam generators and bypass dampers, the steam turbine generator set, main steam condenser, steam plant auxiliary systems, cooling tower, station control system and, electrical systems.

Determination of the inherent reliability of an SSBL gas turbine combined cycle power plant design required a reference plant design which included all ancillary equipment supporting the SSBL gas turbine, and all of the equipment in the steam plant generally referred to as the balance of plant.

The basis for the balance of plant definition was the 330 MW combined cycle plant currently being installed by United Technologies, Power Systems Division for the Korean Electric Company (KECO) at Ulsan Bay, Korea. This plant is representative of a modern gas turbine combined cycle plant. It served as the basis for defining the plant installation layout and steam bottoming equipment.

TABLE 4-9. FAILURE RATE ASSUMPTIONS
(WHERE SPECIFIC DATA NOT AVAILABLE)

	FAILURES PER 10^6 HOURS	REFERENCE
PUMP		NUCLEAR PLANT RELIABILITY DATA SYSTEM. 1976 ANNUAL REPORT, MAY 1977
CENTRIFUGAL	28	
VANE, POSITIVE DISPLACEMENT	37.5	
COMPRESSOR, POSITIVE DISPLACEMENT	145.4	
HEX	9	
VALVE		
PNEUMATIC, GLOBE	14.6	
RELIEF	17.1	
S/O, SOLENOID, BALL	8.0	
BALL, HYDRAULIC	7.8	
SPOOL, PNEUMATIC	12.4	
CHECK, MANUAL, FLOW	1.5	
SOLENOID, GLOBE	4.5	
4-WAY SOLENOID	8.0	
FILTER, MECHANICAL, MESH, LIQUID	2.1	
FAN	4.2	
REGULATOR, PRESSURE	14.6	
LOUVER (LUBE OIL COOLER)	11	
LOUVER MOTOR	2.9	
SENSORS		
TEMPERATURE ELEMENT	3.0	IEEE STD 500 - 1977
TEMPERATUER TRANSMITTER	.9	
PRESSURE SWITCH	2.9	
LEVEL	4.9	

Table 4-10 presents a tabulation of the differences between the combined cycle plant using four TWIN PAC's being installed for the Korean Electric Company at Ulsan Bay, Korea, and the reference plant selected for this study with two SSBL gas turbines. The differences are due almost entirely to the substitution of the SSBL gas turbine for the FT4 and the desire to make the plant non-site specific.

The waste heat steam generator design was provided by the Henry Vogt Machine Company. The Henry Vogt Machine Company has delivered steam generators of this type to approximately 50 installations, many of which have multiple units. This unit is larger than most of the units delivered but is within the present design range of modular equipment supplied by the Henry Vogt Machine Company.

TABLE 4-10. SUMMARY OF DESIGN DIFFERENCES
REFERENCE HRGTCC STUDY VS. KECO ULSAN PLANT

DESIGN ITEM	KECO PLANT	ASSUMED FOR HRGTCC REFERENCE
Gas Turbine	4 FT4 Twin Pacs	2 - SSBL Engines
General Site	Adjoins Existing Plant	Stand Alone Site
Cooling	Direct Seawater from Ulsan Bay	Mechanical Draft Wet Cooling Tower
Aux Steam for Start	From Adjoining Plant	Aux Fired Boiler
Main Condenser Degassing	Vacuum Pumps	Steam Jet Ejectors
Auxiliary Transformer	5 MVA	7.5 MVA

Experience of the Henry Vogt Machine Company with waste heat steam generators has been extremely good. In all of the units delivered to date, there have been only three instances of failure of boiler pressure parts, one because of internal corrosion, the others caused by damage in shipment.

Reliability of the waste heat steam generator is based on experience with this type of unit at the Salt Grass station in Freeport, Texas where the waste heat steam generator has 45,000 hours of operation with no boiler failures. Several shutdowns due to problems with the transition ducting to the boiler have occured, however. Another installation by Henry Vogt Machine Company for Southen Cal Edison has 15,300 hours total operation with no failures attributed to the boiler or ducting. A steam generator at Applied Energy Facility in San Diego, California has had one failure of a drain pipe after 5,800 hours, resulting in a four hour shutdown.

Reliability of the bypass damper is based on experience at the Diamond Shamrock station in Texas where two failures occurred after 2.5 years when malfunction of the limit switches caused cable failures on the bypass dampers. The MTBF was 6500 hours and repair time was 20 hours in each case. The experience of Henry Vogt Machine Company indicates no warping or distortion of the bypass damper provided exhaust gas temperatures are limited to the design condition. At one

installation where exhaust temperature exceeded 300°F above design conditions, warping of the bypass damper did occur requiring repair. However, this kind of failure is not included as inherent to the bypass damper. Additional reliability of the bypass damper and ducting can be achieved by using materials with considerably higher temperature capability than the temperature rating at the normal exhaust conditions of the gas turbine.

Design of the steam turbine/generator as well as cost and reliability information were provided by Toshiba International, which is supplying the steam turbine generator for the KECO plant at Ulsan Bay. The selected model TCDF-23 Toshiba International Steam turbine generator is a tandem compound double flow 23" last stage blade steam turbine generator which has been in operation at various sites for over 20 years. Experience on 7 units operating from 2 to 20 years with 665,000 service hours has provided an MTBF of 166,000 hours and an MCT of 357 hours. Experience on scheduled outages has been 967 hours per year average for 5 of the units over 53 unit years. However, Toshiba feels this is due to longer scheduled outage periods in early installations and their recommended 300-600 hours per year should be adequate for purposes of inherent reliability estimates. Therefore, 600 hours was used for the POH in this reliability study. In addition to the steam turbine generator itself, controls on the steam turbine include an EHC control unit (electrohydraulic) and an automatic sequencer. Reliability information on these units was also provided by Toshiba International. The calculated MTBF and MCT of the steam turbine generator and controls from Toshiba data are 8040 hours and 22 hours, respectively. EEI data from fossil units in the 90-120 MW range taken from the 1965-74 report shows MTBF and MCT values of 8423 hours and 96 hours respectively.

The main steam condenser in the SSBL plant design is cooled by a circulating fresh water loop from a mechanical draft wet tower. Reliability information on the main steam condenser is based on Toshiba International data covering a range of cooling conditions on condensers for plants larger than 75 MW size. This experience includes 4.5 million service hours with only 3 forced outages for a total of 356 hours or 118 hours per occurrence resulting in a failure rate of 0.67 failures per million hours. EEI data shows a failure rate of 3.4 failures per million hours and an average outage time of 43 hours per occurrence.

Systems in the reference steam plant are based on detail piping and instrumentation diagrams available for the combined cycle plant being supplied

to KECO at Ulsan Bay. A failure mode and effect evaluation of each of the fluid systems was conducted and criticality status affecting the availability was established as shown in Table 4-11. Non-critical systems are those in which the component failure would not require plant shutdown for repairs and where the repair could be accomplished in sufficiently short time that the equipment and operating systems would not be damaged or endurance measurably reduced. In each of the critical systems, probable failure modes for each component were established and categorized as major or minor and a spares assumption and MCT established. Several vendors were contacted in connection with reliability estimates on major components in the steam plant auxiliary systems such as boiler feed pumps and circulating water pumps, but these vendors indicated a lack of statistical reliability data and referred us to EEI data. The general comment by vendors on our use of generic data from the nuclear plant data system was that failure rates appeared reasonable in terms of their normal experience, equipment warrantees, and recommended period before major overhaul.

TABLE 4-11. CRITICALITY STATUS OF BOP SYSTEMS

CRITICAL FOR OPERATING AVAILABILITY	NON-CRITICAL
MAIN STEAM	MAKE UP WATER PRETREATMENT
FEED WATER	CHEMICAL FEED
CONDENSATE	AUXILIARY STEAM*
CIRC WATER	WASTE WATER NEUTRALIZATION
AUXILIARY COOLING WATER	SERVICE AIR
HYDROGEN	FIRE PROTECTION AND SERVICE WATER
INSTRUMENT AIR	N_2
	CO_2

* Critical for starting

Figure 4-9 shows a typical piping and instrumentation diagram prepared by C. T. Main for the circulating cooling water system between the cooling tower and the main condensers in the steam plant. This particular subsystem is different from the KECO Plant which uses the water from Ulsan Bay to cool the main condensers.

However, for this study, cooling towers were selected for heat rejection so that the results would not be site-dependent on a cooling water source, such as a river.

The cooling tower was specified by Chas. T. Main and information on size, cost and reliability was provided by the Marley Company. C. T. Main established that the plant can operate at rated output with 5 of 6 cooling tower cells at ambient conditions up to a wet bulb temperature of 72.5 ^{o}F. Therefore, one of the cooling tower cells in the design is considered as an operating spare in the reliability estimate. The Marley Company provided reliability estimates on the major components in the cooling tower including the fan gear reducer, the motor, the fan itself, and the drive shaft.

The reference plant has been designed with separate and independent controls for the gas turbine generator, steam turbine generator, and the steam system. The operation of the independent sections of the reference plant is accomplished by the station operators from the central control room located on the mezzanine in the steam turbine/ generator building.

Controls in the steam plant are based on the KECO plant design. The Foxboro Company which is supplying the control system for the KECO plant participated in this study by estimating the reliability of each of the 15 steam plant control loops. A failure mode and effect evaluation was conducted by Power Systems Division to determine which of the steam plant controls were critical from the standpoint of plant shutdown.

Reliability of the steam plant controls was estimated by Foxboro for the automatic mode. Sufficient information was provided on individual component failure rates to permit evaluation of each individual control loop in a manual mode and for common mode failures. In the failure mode and effect evaluation of the 15 critical steam plant controls, Power Systems Division determined which controls could be switched from an automatic to manual mode and operation of the plant continued manually despite a control component failure. Difficulty in diagnosing a failed component in the control system and the time for the operator to react before an automatic plant tripout occurs were considered in the controls evaluation. Table 4-12 shows typical steam plant control failure rates as estimated by Foxboro for the automatic mode and adjusted by Power Systems Division for operation in the manual mode for those controls where the FMEA indicates that is possible.

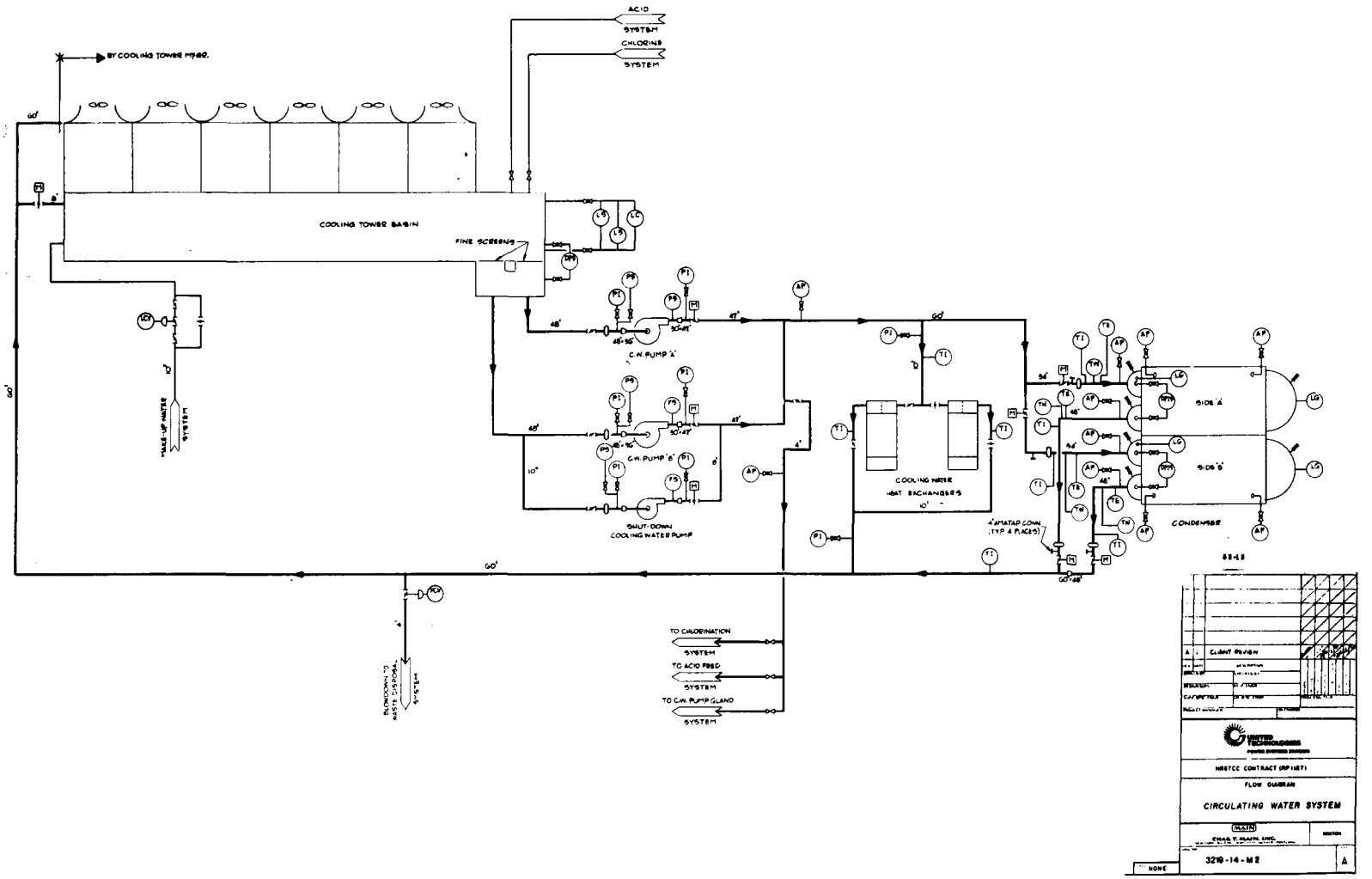


Figure 4-9 Piping and Instrumentation Diagram for SSBL Plant - A circulating cooling water system is used to preclude dependence on a cooling water source.

TABLE 4-12. TYPICAL STEAM PLANT CONTROL FAILURE RATES (NUMBER OF FAILURES PER MILLION OPERATING HOURS)

	Automatic From Foxboro	Adjusted by PSD For Operation in Manual Mode
Level Control	123	39
Pressure Control	35	25
Temperature Control	47	44
Flow Control	54	38

The electrical systems for the SSBL combined cycle plant were defined based on the KECO plant. The electrical schematic of the combined cycle plant is shown in Figure 4-10. The electrical systems for the SSBL gas turbines were defined based on the FT50 system.

A failure mode and effect evaluation of the electrical systems was completed for both the output power electrical system and the auxiliary power electrical system. The output power electrical system evaluation covered electrical buses, step up transformers, circuit breakers, disconnects and protective relays from the output of the generators up to and including the 2 parallel 154Kv switch-yard buses. The auxiliary power system evaluation covered the step down transformers, circuitbreakers, buses, protective relays, fuses and motor control center buses. The failure rate for a motor starter was added to the failure rates for each of the pumps in the subsystems. Failure rates for electrical equipment were based on IEEE Standard 500.

SUMMARY

The stage for the actual conceptual design studies conducted during Phase I was set by the reliability assessment initiated early in the program. This assessment consisted of a series of steps that started with the current FT4 field experience and led towards establishing a reference balance of plant. An intermediate step involved taking advantage of a pre-existing design concept (Single Shaft Base Load) whose characteristics were better suited to future combined cycle base load application in the 100 MW class. The Single Shaft Base Load (SSBL) served in establishing the reference plant, along with an existing (KECO ULSAN, KOREA) installation for the balance-of-plant.

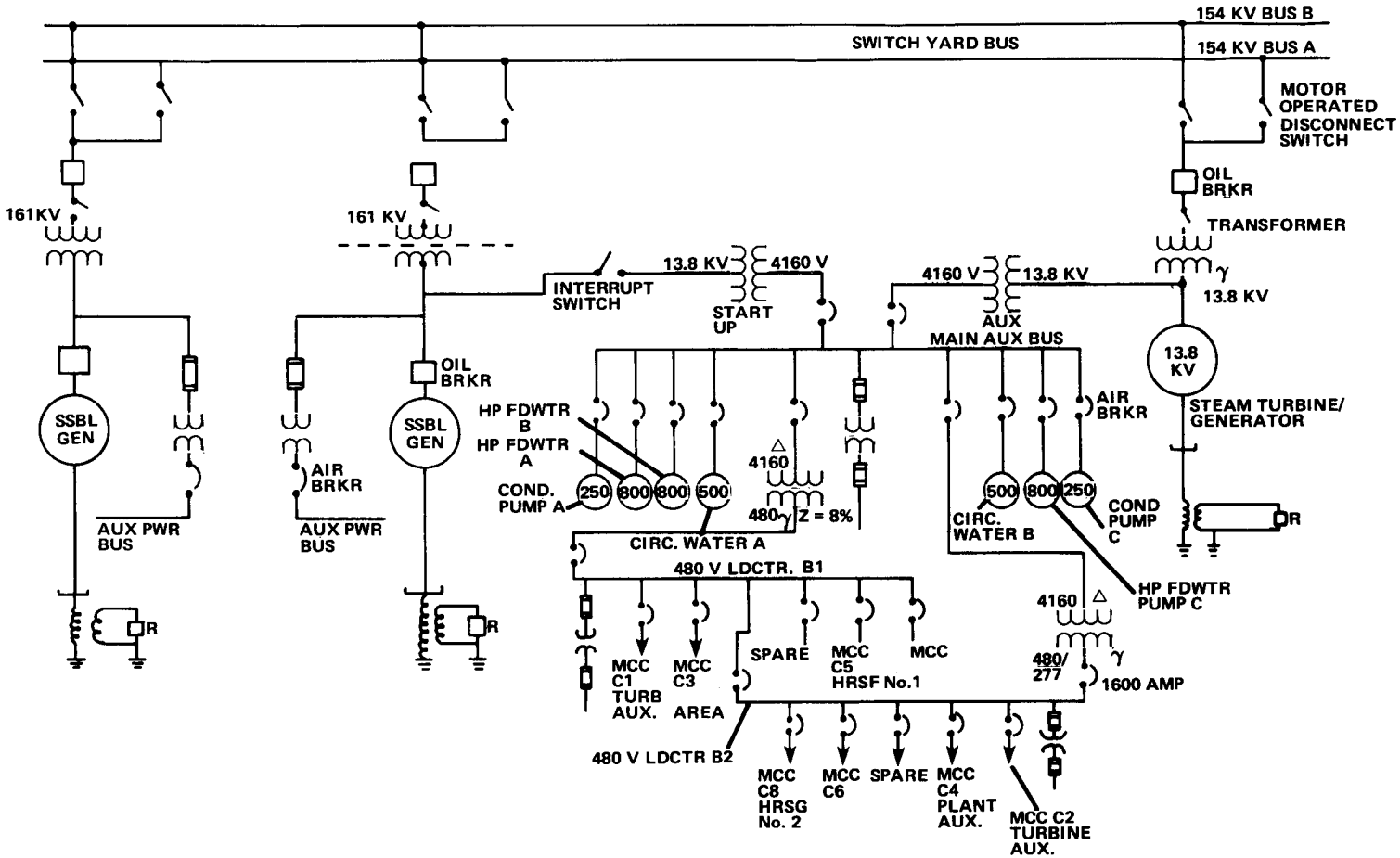


Figure 4-10 Electrical Schematic - The electrical system was based on the system for the FT50 powerplant.

The reliability connection between the SSBL and the FT4 was made by using United Technologies' Reliability Prediction Method, which embodies the entire gas turbine reliability data base, including applicable flight experience. The data base helped corroborate the prediction method, which was used in turn to project the reliability for the final HRGT design.

Reviewing the FT4 experience itself, along with analysis of the base of reference plant, provided insight in preparing the new centerline design. The reference also served as the basis for the trade-off studies covered in Section 5 of this report. Section 6 covers the reliability analysis itself.

Conducted later in the program, the on-site survey of utility operator experience provided further valuable insight for the HRGT design, especially with respect to design for reliability of the gas turbine and balance-of-plant auxiliary equipment.

SECTION 5.0

RESULTS OF STUDIES LEADING TO DEVELOPMENT OF THE GAS TURBINE DESIGN CONCEPT

SUMMARY OF APPROACH

General Approach

Having established, via the SSBL, a benchmark of reliability both for the engine and completed power plant, the next major step leading to the HRGT conceptual design consisted of a series of trade-off studies. This included cycle selection, aerodynamic parametrics for the turbomachinery, and combustor arrangement and technology selection.

The trade-offs investigated and optimized the key design parameters and features of the engine, including cycle parameters, pressure ratio, turbine inlet temperature (along with turbine airfoil metal temperature and cooling scheme), turbomachinery aerodynamics, combustor configuration, mechanical/structural configuration, gas turbine engine auxiliaries and balance of plant equipment (especially heat recovery steam generators), size class, and starting methods.

The cycle selection effort determined an optimum level for pressure ratio as a separate study. The key parameter, turbine inlet temperature, was determined in a combined parametric study which also included selection of the airfoil metal temperatures and cooling schemes, all closely interrelated.

The compressor and turbine aerodynamic designs evolved from broad parametric evaluations which covered combinations of key aerodynamic design parameters such as blade aspect ratio, hub and tip geometry, and throughflow velocity. Here use was made of the United Technologies' empirical aerodynamic data base and computerized design system to evaluate some 81 different compressor designs and about 30 different turbine designs.

The combustor selection studies involved trade-offs of reliability vs. fuel flexibility technology and emissions capabilities. In addition, studies were

made of various configuration arrangements as the combustor, together with its diffuser section connecting with the compressor in the front and transition section connecting with the turbine at the rear, might best integrate with the overall engine.

Throughout the trade-off studies reliability and COE were used as measures of merit to guide optimum choices. The SSBL reference base was used extensively to that end. Each change was quantified as a difference from the reference.

Cost of Electricity as a Trade-Off Factor

From the operator's point of view, reliability is vital because of its impact on generating costs. Ultimately, allowances for unavailable equipment must be made by planning for extra system capacity, which is a long term, capital expense. Additional costs are incurred in the short-term when an efficient plant is not operating and its power must be supplied instead by a less efficient plant from standby reserve. The reliability vs. cost trade study recognized that trade-offs in initial conceptual design of plant equipment which emphasized reliability should also include overall cost as a trade factor in a way that relates one to the other. Relating reliability to cost ensures that the engine design represents a balanced optimum for the electric utility application.

Cost of Electricity (COE) trades were conducted in each of the separate trade studies. At the same time, plant equipment reliability and scheduled maintenance times were estimated and combined into overall power plant scheduled and forced outage rates (POH and FOH). It was possible to relate POH and FOH rates to power cost by means of the economic formulation provided in EPRI's Technical Assessment Guide, as supplemented with specific EPRI economic premises. This formulation proved to be a powerful tool in all of the trade studies conducted.

As suggested by EPRI, plant COE was calculated based on 30-year levelized costs for capital, and overhaul and maintenance (O&M) expenses. The EPRI formula takes into account not only capital and O&M expense, along with fuel costs, but also scheduled and unscheduled outage hours. The key to EPRI's method is the use of an "effective capacity" factor, (C_{eff}). This factor includes an accounting for a utility system characteristic parameter, (m), is the difference between the rated capacity of a prime mover and the reserve capacity required to maintain a

fixed margin, on a statistical basis (loss of load probability). These considerations are further discussed in Appendix A.

As also suggested by EPRI, parameter trade-offs were calculated for different values of (m) to determine the influence of different utility generation mix characteristics.

Maintenance costs, as a parameter variable, were developed by Chas. T. Main Engineers, Inc. based on their overall experience with actual plant operation and staffing.

More specific details, assumptions, and definitions actually used for cost and economic trades are provided in Appendix A. This appendix covers the power cost calculation itself, the plant capital cost background, and a discussion of Charles T. Main's methods for estimating O&M costs. The results of the COE analyses are reflected in the change in cost of electricity differences which occur as and result in the various trade-off studies conducted throughout the program.

Cycle Selection

The components for the engine were selected based on the use of existing technology to ensure that the reliability goals can be achieved and the engine will be available by 1986. The only component where the use of existing technology was not adequate was the combustor. Advanced technology is required to satisfy the emission requirements when the engine is operating on residual oils, or coal derived liquids or gases. An advanced combustor concept has demonstrated the potential of achieving the emission requirements with the desired fuels. Further R&D is required to meet the emission requirements, combustion stability, and liner durability over the full range of power, and to determine the allowable fuel property limits.

The question of turbine inlet temperature selection represented a key consideration in the design of the Highly Reliable Gas Turbine engine. Choice of turbine inlet temperature came about through a comprehensive trade-off study and included the choice of airfoil metal temperatures and cooling technique.

The selection process involved a parametric study which covered a broad range of gas temperatures, metal temperatures and cooling techniques. A total of ten

discrete combinations were investigated in detail in terms of row-by-row performance and reliability. Turbine rotor inlet temperatures were examined in the range from 1525°F to 2300°F, and metal temperatures ranged from 1100°F to 1550°F. For each combination, appropriate cooling technology schemes were selected for detailed evaluation. The total study matrix is summarized in Table 5-1.

TABLE 5-1. MATRIX OF TURBINE ROTOR INLET TEMPERATURES, AIRFOIL TEMPERATURES AND COOLING SCHEMES

Rotor Inlet Temp, RIT, °F Metal Temperature, TM, °F	1900 1100	2150 1100	2300 1100	1900 1330	2150 1330	1900 1450	(SSBL Base) 2150 1450	2300 1450	2300 1550	1525 1550 (V) 1450 (B)
1st Vane Cooling Scheme	W	W	W	F	F	F	F	F	F	C
1st Blade Cooling Scheme	C	C	C	C	F	C	C	C	C	U
2nd Vane Cooling Scheme	W	W	W	C	C	C	C	C	C	U
2nd Blade Cooling Scheme	C	C	C	C	C	U	C	C	C	U
3rd Vane Cooling Scheme	W	W	W	U	C	U	U	C	U	U
3rd Blade Cooling Scheme	C	C	C	U	U	U	U	U	U	U

C = Convective F = Film U = Uncooled W - Water

Results were measured in terms of cost-of-electricity (COE) differences and reliability in terms of failure rates. COE was determined by translating turbine performance to heat rate, and parts cost to plant capital cost. Reliability was determined by a statistical analysis which calculated failure rates for each blade row, integrated into an overall mean time between failure rate (MTBF) for the engine.

Selection of the pressure ratio followed after the selection of the turbine inlet temperature. The successive rather than simultaneous selection of these two key variables was appropriate for two reasons. First, and foremost, the selection of turbine temperature along with the cooling scheme exerts a paramount influence over the reliability of the new engine design. Pressure ratio has much less influence on overall reliability, as evidenced from past gas turbine operating experience. Second, for a combined cycle plant, the variation in performance is not strongly influenced by cycle pressure ratio variations,

unlike simple cycle plants. This basic trend is explained by the fact that as pressure ratio is increased to improve simple cycle heat rate, the engine exhaust gas temperature is reduced at the expense of bottoming cycle efficiency. Thus one efficiency balances the other with the net effect that the combined efficiency remains about constant over a wide range of pressure ratio variation.

To determine an optimum level for pressure ratio around which to design the engine, preliminary conceptual engine layouts were completed for three discrete designs, representing 10:1, 14:1 and 18:1 pressure ratios. These layout designs comprised an elemental parametric analysis where overall COE, through heat rate and cost, was determined along with reliability projections.

Component Selection Process

Compressor. The selection of the compressor for the gas turbine engine was based on a broad aerodynamic design parametric study covering the significant ranges of the important compressor design parameters. These were followed by mechanical design studies to establish the mechanical configuration of the compressors selected from the aerodynamic studies with emphasis on reliability and maintainability. At the same time, reliability studies were conducted to confirm the design selection as optimum from the reliability point of view.

Combustor. The combustion section presents a unique set of problems to the Reliable Engine. Program goals require low emission operation using a wide range of fuels including heavy residual and coal derived liquids. The technology for environmentally acceptable methods to accommodate the low hydrogen and high nitrogen contents of these fuels is in the early stages of development. Even with clean fuels containing no bound nitrogen, the technology of dry NOx control has progressed only to engine-level demonstrations. Production dry NOx combustors have yet to enter the field.

Several combustor configurations were analyzed for their applicability to the HRGT design. Options for burner type, ignition and liner cooling were considered.

Turbine. The turbine design evolved from a parametric study which examined the effects of several important turbine aerodynamic parameters on efficiency and

numbers of airfoils. The parameters which were varied for this study were: 1) mean turbine diameter; 2) inlet annulus area; 3) airfoil chord; 4) airflow levels. The study investigated diameter and annulus area because these variables generally have the strongest influence on turbine efficiency. Because the effect of chord generally has a weaker effect on efficiency, but affects airfoil count, its influence was examined to a more limited extent. Increased power output with increased airflow will probably be desirable at some future time. For this reason, the effect of increased airflow was also examined.

Mechanical Layout Design of Candidate Concepts

As the third major step of the evolutionary process leading to the final selected gas turbine design concept, the completed trade-off analyses were used as the basis for selecting four candidate concepts to be investigated from an overall mechanical arrangement standpoint. Overall reliability and maintainability assessments were made of the candidate concepts as well as mechanical feasibility evaluations. The final selected configuration incorporated the best features of the four candidates as well as improved features uncovered during this final part of the conceptual design process.

TRADE-OFF STUDY RESULTS

Turbine Inlet Temperature and Cooling Method

The cost of electricity evaluation results turned out to be most predominantly influenced by the turbine inlet temperature effect on heat rate. In a combined cycle, increased turbine inlet temperature tends to raise the exhaust temperature, which in turn serves to improve the bottoming cycle efficiency. The cycle result is shown in Figure 5-1.

The net specific power output is increased with increased turbine temperature which results in an improvement in dollars per kilowatt capital cost. The cost improvement taken together with the improved heat rate resulted in a continuous reduction of cost of electricity over the full range of turbine inlet temperatures evaluated.

Similarly, although to a lesser degree, heat rate and specific output improved as metal temperature increased. This result directly reflects the performance

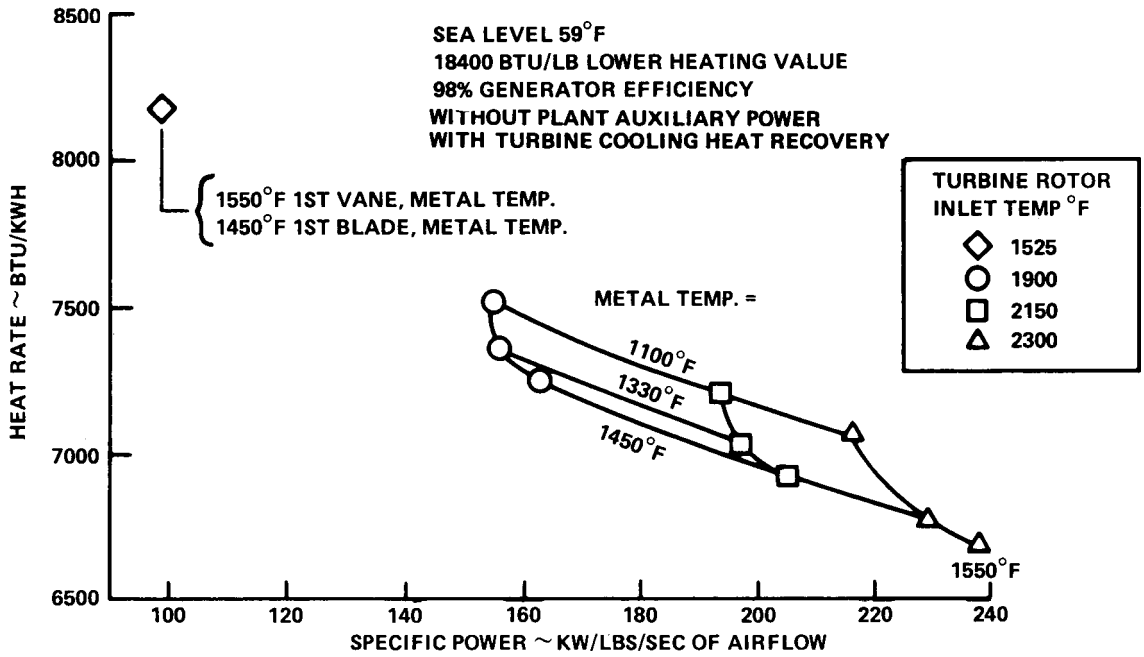


Figure 5-1 Results of Turbine Parameter Trade-off Study showing improved heat rate and specific power both with increasing turbine inlet temperature and airfoil metal temperature.

penalty associated with increased cooling flows required to maintain reduced metal temperatures.

The overall cost of electricity result is represented in Figure 5-2. Based on cost of electricity alone there is strong incentive to keep turbine inlet temperature and metal temperature high. However, reliability (failure rates), cooling scheme technology, coatings durability and maintenance goals limit turbine inlet temperature.

A statistical reliability analysis was conducted for the parametric matrix described in Section 3. The analysis used historical turbine airfoil failure rate correlations as a means for predicting the new designs. Results of this analysis are shown in Figure 5-3. When considering all possible failure modes there resulted an optimum level of metal temperature, which occurs in the range 1300°F to 1450°F, depending on turbine inlet temperature.

A major program goal has been to schedule major maintenance overhauls at intervals of no less than 18,000 hours of operation. A critical limitation on

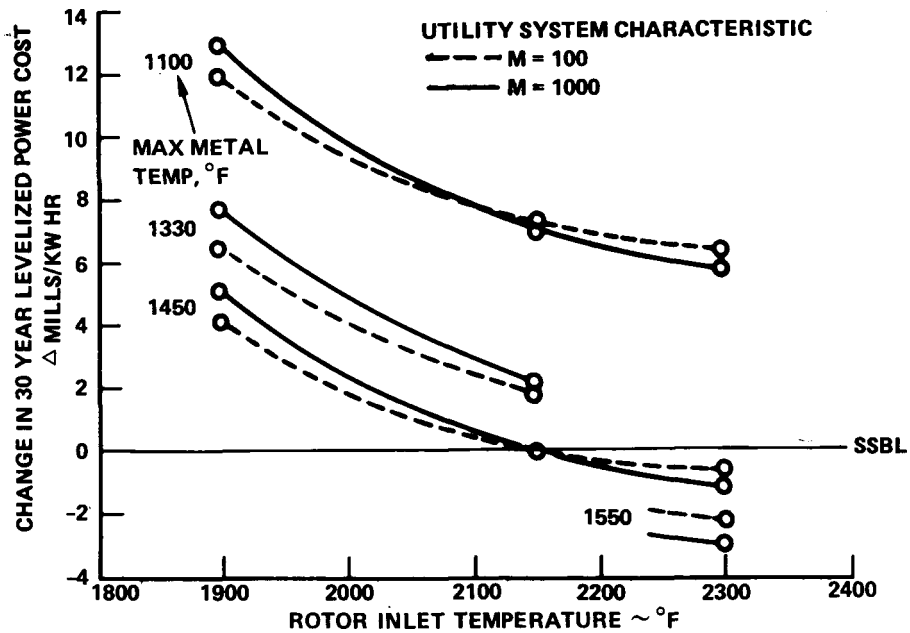


Figure 5-2 Rotor Inlet and Maximum Metal Temperature vs 30-Year Levelized Power Cost - Consideration of cost of electricity alone would indicate that turbine metal temperatures should be high. (See Appendix A for explanation of the change of cost of electricity background, including the parameter "M".)

the maintenance interval will be the turbine airfoil coating life. The surface coating will gradually be consumed by sulfidation attack until the base metal is exposed. Before reaching that point the blades and vanes should be removed from the engine and recoated and/or refurbished. Using the best available coating technology, projected to the 1985 time period when the power plant would be operational, the 18,000 hour point was derived for a 1450°F airfoil metal temperature.

Figure 5-4 shows the cooling scheme arrangements used for the parametric matrix. The parametric study analysis assumed water cooling for the stationary vanes and convectively air-cooled wafer-construction vanes and blades. These cooling selections result in low metal temperatures while avoiding the small holes associated with film cooling that are likely to plug when heavy fuels are used. However, in both instances, a technology is assumed that needs further development before proven useable in a production environment. Thus, technical risk becomes a judgement factor as a part of the selection process.

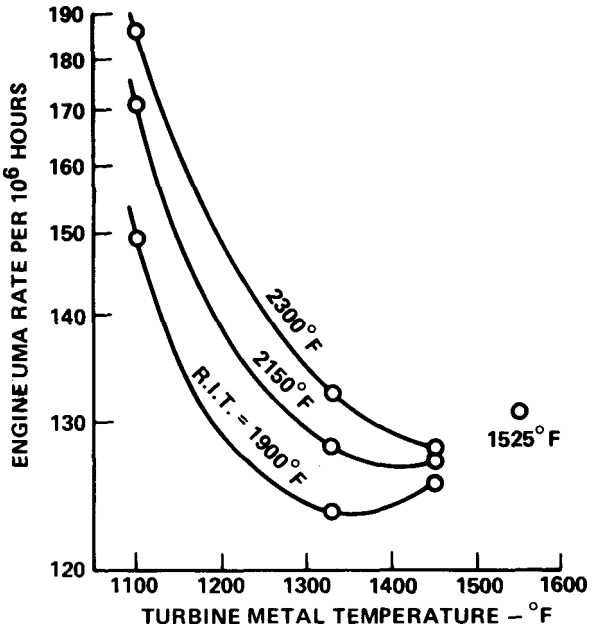


Figure 5-3 Trade Study Effect on Engine Unscheduled Maintenance Action Rate - Unschedule maintenance considerations indicate an optimum turbine metal temperature in the range of 1300°F to 1450°F.

Water cooling was not considered for the rotating blades because such technology represented a risk well beyond the 1985 time period and was not considered suitable for the HRGT design. However, as a consequence, when metal temperatures as low as 1100°F were assumed, a major heat load was shifted to the exposed blades when the stationary vanes took advantage of the water cooling. This results in not only a performance penalty but also a reduction in overall turbine reliability.

The candidate concept chosen for the rotating blades consisted of a new two-piece construction technology. This technology was not evaluated in the initial trade-off studies. However, towards the latter portion of the design studies, independent but parallel United Technologies studies identified the promising potential of two-piece construction. This technique is currently being perfected for United Technologies aviation gas turbines. As such, it is expected to be available for production use in the HRGT. The two-piece construction allows for the fabrication of fairly intricate internal cooling passages which allow high heat transfer. The technique accomplishes some portion of the heat transfer effectiveness gain available with the wafer construction technique.

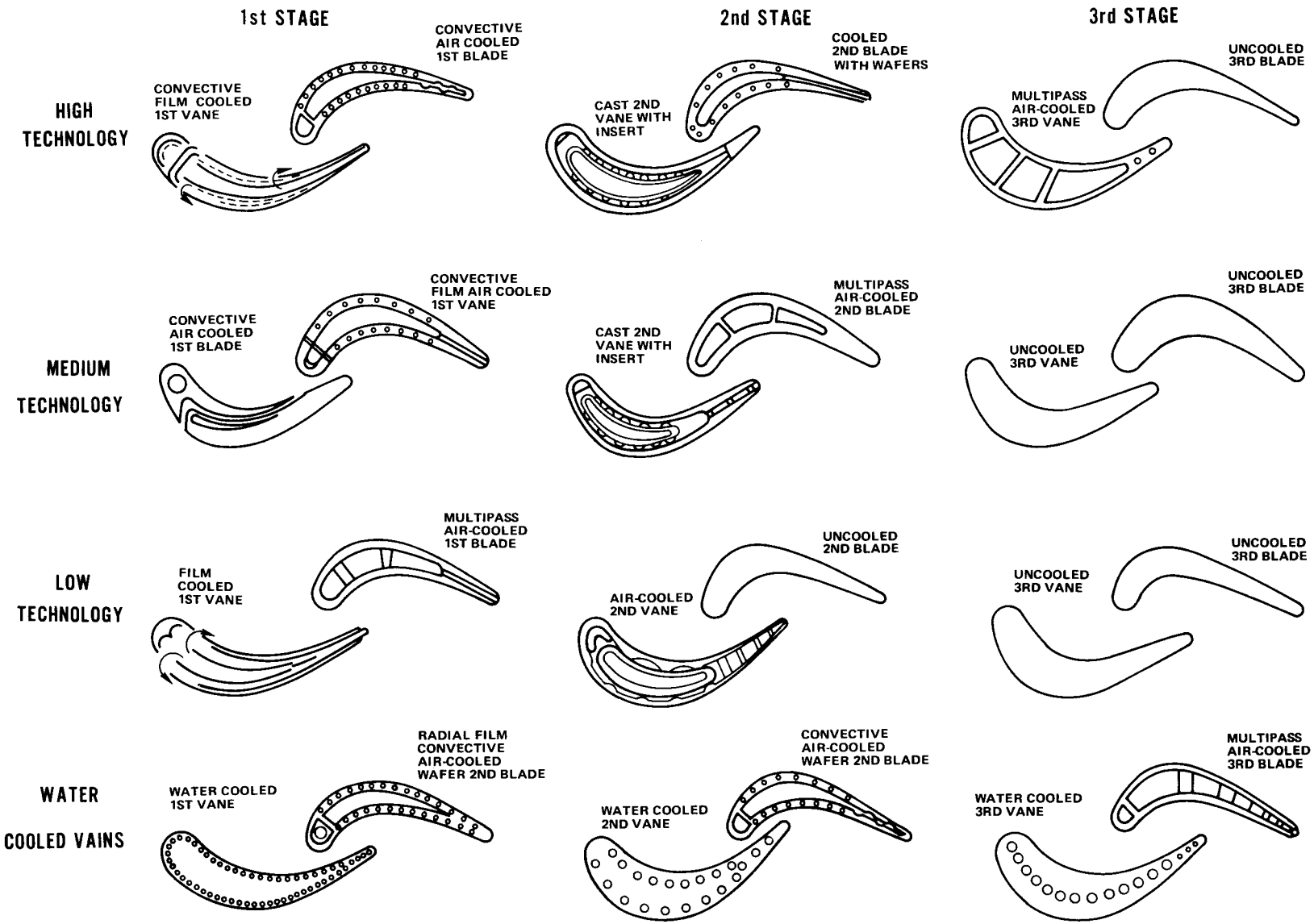


Figure 5-4 Candidate Turbine Cooling Schemes Used in Trade-off Studies

A final turbine inlet temperature selection can be made using the foregoing factors. The 1450°F metal temperature represents a near optimum in reliability (parts failure rates) and also allows an 18,000 hour time interval between airfoil recoating. As shown in Figure 5-5, the two-piece technology limitation intersects the 1450°F metal temperature at 2000°F turbine inlet temperature (which corresponds to 2150°F combustor exit, or firing temperature). The resulting combination provides low cost of electricity, which can in the future be further improved by using more sophisticated future cooling techniques as would be available with the wafer construction.

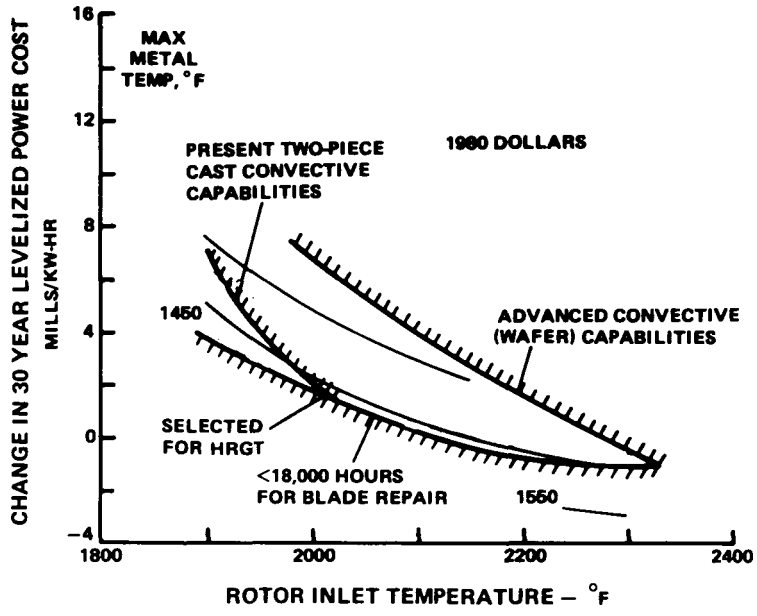


Figure 5-5 Rotor Inlet Temperature Determination - Rotor inlet temperature was determined on the basis of blade life and current cooling technology capabilities.

Pressure Ratio

To determine an optimum level for pressure ratio around which to design the engine, preliminary conceptual engine layouts were completed for three discrete designs, representing 10:1, 14:1, and 18:1 pressure ratios. These layout designs comprised an elemental parametric analysis where overall cost of electricity, through heat rate and cost, was determined along with reliability projections.

There were two obvious drawbacks associated with the higher pressure ratio case. First, design for more than a 14:1 pressure ratio on a single spool results in off-design stage matching difficulties. The off-design matching problem is only alleviated through incorporation of variable geometry stages, which can become a liability from a reliability and maintainability standpoint. Second, the greater compressor length exceeds the bounds where the rotor support can be accomplished with only front and rear bearings. The added bearing and internal bearing compartment pose additional reliability and maintainability detriments. As a final consideration, while a pressure ratio less than the near optimum 14:1 might not represent any appreciable cost of electricity penalty, the simple cycle efficiency would suffer, making the engine less attractive for peaking load or other potential applications.

The cost of electricity analysis results are shown in Figure 5-6. The results show that there is little significant difference over the range considered. However, 14:1 does appear to be near optimum. Since on the basis of the preliminary design 14:1 appeared much more feasible from a mechanical and aerodynamic design standpoint than the 18:1 case, this level was selected as the basis for the HRGT design.

STARTER SYSTEM

Table 5-2 presents a summary comparison of the 6 alternative starting system concepts.

Each start drive system concept was appraised in sufficient detail to establish size and cost. Equipment vendors were consulted regarding available size, weight, reliability, and cost information. Pertinent published data in the general literature were reviewed as well for information on starting reliability. Cost and reliability data obtained were compared along with other pertinent factors, for the six starting methods considered.

Electric Motor

The electric motor start concept is generally considered to have very good starting reliability and has the advantage of considerable background and experience in both design and support from equipment manufacturers. In addition, the electric start motor can be used during water wash of the engine and a

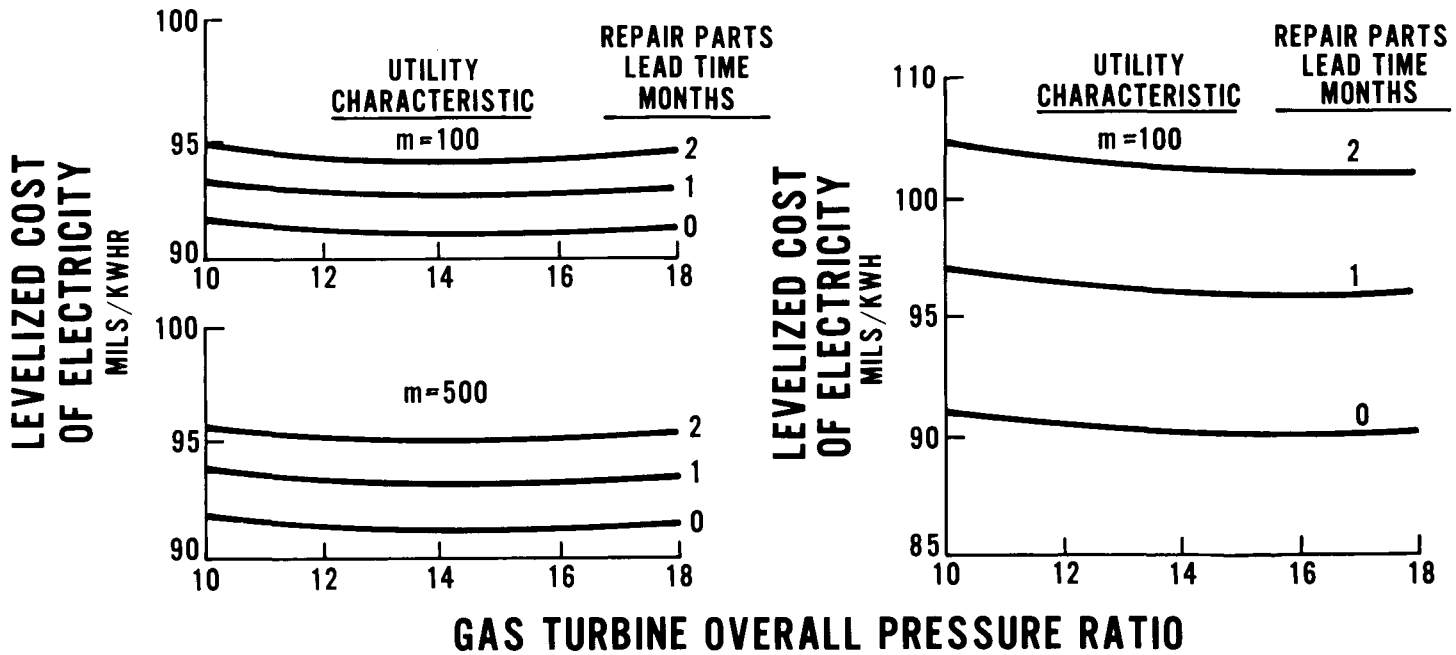


Figure 5-6 Effect of Pressure Ratio on Cost of Electricity

TABLE 5-2. ANALYSIS OF STARTING SYSTEMS

	(Reference Design) AC Motor	Generator Convert to start motor	Diesel	Gas Turbine	Steam Turbine	Compressed Air
Relative Cost	1.0	1.21	1.15	1.38	1.64	1.91
Inherent Reliability	Very good 99.72%	Very good	Good 99%	Good 94%	Very good (used in nuclear safety systems)	Very good 98%
Restrictions on Engine	None	None	None	None	None	Ejector design restricts configuration to generator on turbine. Air motor design requires shaft extension thru generator or out the turbine
Restrictions on Generator and Exciter	Shaft extension thru generator	None	Shaft extension thru generator	Shaft extension thru generator	Shaft extension thru generator	
Integration of barring Motor	Simple integration with start drive torque converter	Can accomplish barring function	Simple integration with start drive torque converter	Simple integration with drive clutch	Simple integration with drive clutch	Must be accomplished with separate system
Speed Control for Engine Wash & Cooldown	Limited to on-off use torque converter	Variable over very wide range	Limited to narrow range use torque converter	Limited to narrow range	Variable over wide range	Variable
Maintainability	Weighs 4000 lbs	No mechanical equipment beyond generator	Weighs 4800 lbs.	Weighs only 423 lbs.	Weighs 7900 lbs.	Heavy hood but small mechanical controls
Development Status	Well developed Widely used on large frame gas turbines	New concept No actual experience beyond proposals	Well developed Occasionally used on large frame gas turbines	Well developed Not used for this application due to cost	Well developed Not generally used for this application due to steam requirement. Used extensively in nuclear plants for safety system pump drives	Used on FT50 GT-200 FT4
Impact on Gas Turbine Enclosure	Extended enclosure 12.4'	No extension of enclosure - Equipment located in main bay	Extends enclosure 3.2'	Extends enclosures 8'	Extends enclosure 9'	Extends enclosure 12'
Impact of Black Start Requirement	Requires auxiliary power supply	Requires auxiliary power supply	Self starting	Self starting	Self starting from fired boiler	Self starting

measure of speed control can be provided by accelerating the gas turbine to a desired speed during the water wash, then cutting out the torque converter and allowing the gas turbine to decelerate under its own inertia.

Disadvantages of the electric motor start include the requirement for an on-site auxiliary power supply in the event that a black start is required, as well as the requirement of an extension of the drive shaft through the generator and exciter for connection to the electric start motor. This adds about 12.4 ft. to the length of the gas turbine enclosure building.

A hydraulic torque converter is generally used (and was included in the final selection). An alternative is the use of a magnetic variable speed drive which offers more flexibility in speed control for water wash. However, this approach still requires a mechanical clutch and a drive system, and it is about 30 percent higher in cost than the hydraulic torque converter.

Conversion of Generator to a Start Motor

A relatively new approach for consideration is the conversion of the generator on the gas turbine for use as a start motor using a variable frequency converter.

The primary feature of this system is its mechanical simplicity. It does not require extension of the drive shaft beyond the generator and exciter and therefore does not affect the gas turbine enclosure. It also eliminates additional requirements for mechanical components for the installation. A further advantage of the generator as a start motor is the wide range of speeds available, which provides greater flexibility during cooldown and water wash. Although the cost for a single gas turbine installation is relatively high with a variable frequency start drive system, the cost of this equipment can be shared in a multiple gas turbine installation. When the cost can be shared between two gas turbines, the cost is only slightly higher than conventional electric motor start drives. However, use of a single variable frequency drive to accomplish barring in a system with two gas turbines could present a problem if one engine needed to be started while the other needed to be barred.

While this approach does not have extensive development experience in this country, the design of the variable frequency drive has been extensively studied and is presently offered by certain gas turbine manufacturers in Europe.

Diesel Engine

The diesel engine is often used for starting large gas turbines. Several diesel engine manufacturers were contacted in the course of this study including Detroit Diesel Division and Cummins Diesel Inc. The diesel engine drive system includes a torque converter and clutch, for which information was provided by Twin Disc, Inc.

The diesel engine has the advantages of black start capability without an auxiliary power supply and an available on-site fuel system. The diesel engine has a relatively narrow range of speeds but deactivating the torque converter can provide speed flexibility during water wash and cooldown. The diesel start drive system is only slightly more expensive than the electric motor. As in the case of the electric motor, a separate system is required for each gas turbine. Start up of the diesel engine can be accomplished by a small ac motor, or a DC motor and batteries can be used where black start is a requirement.

Small Gas Turbine

A small gas turbine has the same advantage as the diesel in terms of flexibility for black start. It has limited speed control capability for use in powering the large gas turbine for water wash and variations of speed during cool down.

One example of a small gas turbine which might be used is the PT6A-45A supplied by Pratt & Whitney Aircraft of Canada. This machine has a free turbine and built in gear box, requires no torque converter and weighs only 423 lbs. It can provide the 1100 horsepower required for start up of an 100MW size gas turbine, has about the same mechanical complexity as the diesel start system, and requires a smaller extension of the gas turbine building to accommodate the start system. This approach has not found its way into applications to date because of its relatively high cost.

Small Steam Turbine

Information for a 1,000 horsepower steam turbine drive system and controls was supplied by Terry Steam Turbine, Inc. The steam turbine shown in Figure 5-7 is a single-stage, solid-wheel industrial unit operating up to 5000 rpm with a gearbox. As in the case of the small gas turbine, the steam turbine requires no

torque converter, but a clutch for disconnecting the system once the gas turbine is operational is required. The steam turbine has an exhaust pressure of 15 psig and can exhaust to the steam plant main condenser, deaerator, or condensate tank, rather than to a separate condenser because of the relatively infrequent use and short duration of operation.

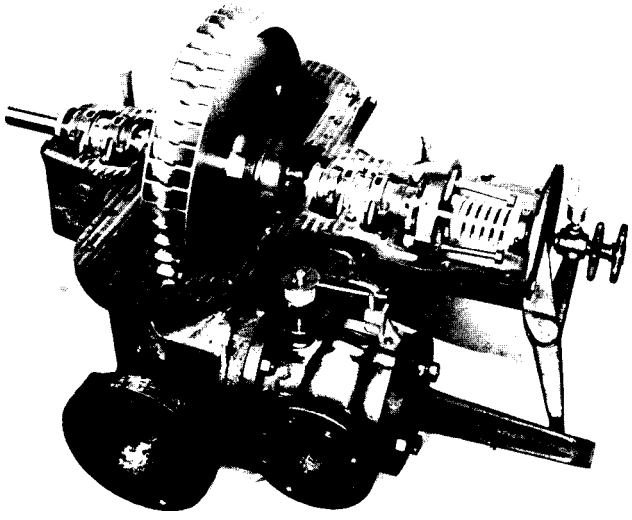


Figure 5-7 Steam Turbine Starter System - The steam turbine starting system would not be cost effective in view of the cost of modifying the auxiliary boiler system to drive the starter.

A particular requirement for the small steam turbine is a boiler for steam generation. One possibility is the auxiliary boiler anticipated in the study of the combined cycle plant design by Charles T. Main, to provide freeze protection, building heating, and warming of the main steam systems and steam turbine generator. However, increasing the boiler pressure from 125 psig to 650 psig and supplying an extra 24,000 pounds per hour of additional steam to a small steam turbine for starting the large gas turbines has a major cost impact.

Compressed Air System

Compressed air start systems are relatively common for gas turbine starting. They are used with the FT4, were used on the FT50, and are presently in use for the Stal Laval GT200 50 Hz version of the FT50. The method of utilizing the compressed air differs in the case of the FT4, FT50 and GT200. In the FT4, the compressed air drives a small air motor which is connected directly to the gas turbine through a gearbox. Both the FT50 and Stal Laval GT200 inject compressed air directly into the front end of the compressor of the gas turbine using nozzles. Figure 5-8 shows a schematic of the starter system for the FT50. Stal Laval, in its recent operation of the GT200, has conducted 89 starts with only one failure to start directly attributable to the start system.

The air start system has the advantage of providing considerable excess air through the gas turbine. This air provides cooling and a relatively stress free start condition in the turbine. In addition, compressed air start has the added advantage of energy storage for a black start and is a relatively reliable system in that once the compressed air receivers have been charged, activation of the system involves only a few control valves and movement of the mechanical ejector hood to engage with the gas turbine compressor.

The air start system has the disadvantage of high cost, particularly in installations where there is only one gas turbine. The high cost is due primarily to the large and expensive air storage tanks. For example, the air storage system for the FT50 included two tanks, 70 inches in diameter by 47 feet long, weighing 70,000 lb each. In a multiple gas turbine installation, the air compressor system and storage tanks can provide startup capability for both gas turbines. However, the cost, even under these conditions, is still considerably higher than either an electric motor or diesel start system.

The air start system using an ejector design as developed for the FT50 and GT200 large industrial turbines has the additional disadvantage that it restricts the configuration of the engine to a turbine-end mounted generator in order to leave the compressor end of the engine clear for the start ejector system. The system is also relatively expensive.

Another alternative for the air starting system is the use of a small air-start motor directly attached to the shaft beyond the generator and exciter. This

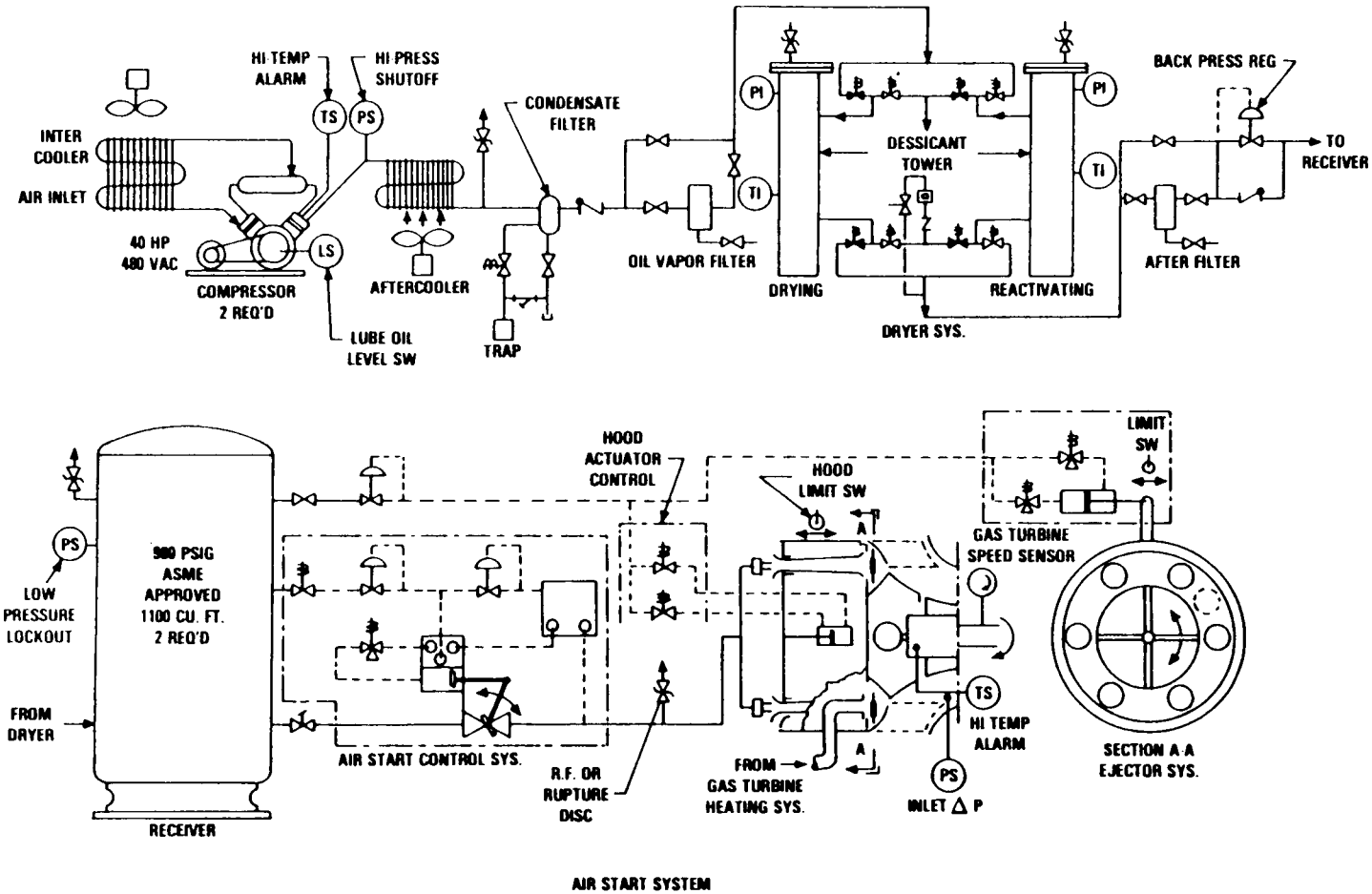


Figure 5-8 FT50 Compressed Air Starting System - Compressed air system provide a low stress start but are expensive and in some designs must be mounted at the turbine end of the gas turbine.

approach offers the convenience of a relatively small package and avoids the large hood and ejector system used on the FT50 and GT200. The primary advantage of this system is that it allows a front end mounted generator and keeps the turbine-end of the engine free for a direct exhaust to the waste heat steam generator.

A more detailed review should be made of using the variable frequency drive conversion of the generator as a start motor. Further consideration of this approach is recommended because of its relative mechanical simplicity and the promise of further cost reduction with improvements in the state of the art in solid state devices for variable frequency drives.

Heat Recovery Steam Generator System Selection

Selection of the three-pressure system was based on data provided by The Henry Vogt Machine Co., which designed and costed boilers for both a three pressure and a single pressure boiler system and on overall plant cost data provided by Charles T. Main. Reliability of the one-pressure boiler was adjusted to account for changes in boiler controls, feedwater pumps and the use of a deaerating condenser in place of a deaerator.

The study results indicated that the three pressure design offered significantly lower cost with only slightly lower availability (0.02 percent).

The study results showed that the combined cycle heat rate and net plant output power are both adversely influenced by the use of a single pressure boiler which does not recover as much waste heat. This is evidenced by the higher exhaust temperature of 404°F compared with 310°F with a three-pressure boiler. Boiler cost is significantly reduced as is the steam plant cost with a single-pressure boiler. However, the lower rated output power and higher heat rate more than compensate for the lower cost. The result is higher cost of electricity with a single-pressure boiler by 1.76 to 2.13 mills/kwh depending on the utility system characteristic, for a plant operating on No. 2 fuel. For a plant operating on No. 6 fuel, the cost of electricity increases by 1.59 to 1.99 mills/kwh with a single-pressure boiler.

The influence of plant capacity factor on the comparison of a single-pressure versus a three-pressure boiler plant is shown in Figure 5-9. The increase in

cost with a single pressure boiler is shown over a range of plant capacity factor, utility system characteristic and for both No. 2 and No. 6 fuel. The cost change is 30 years levelized busbar cost in 1980 dollars.

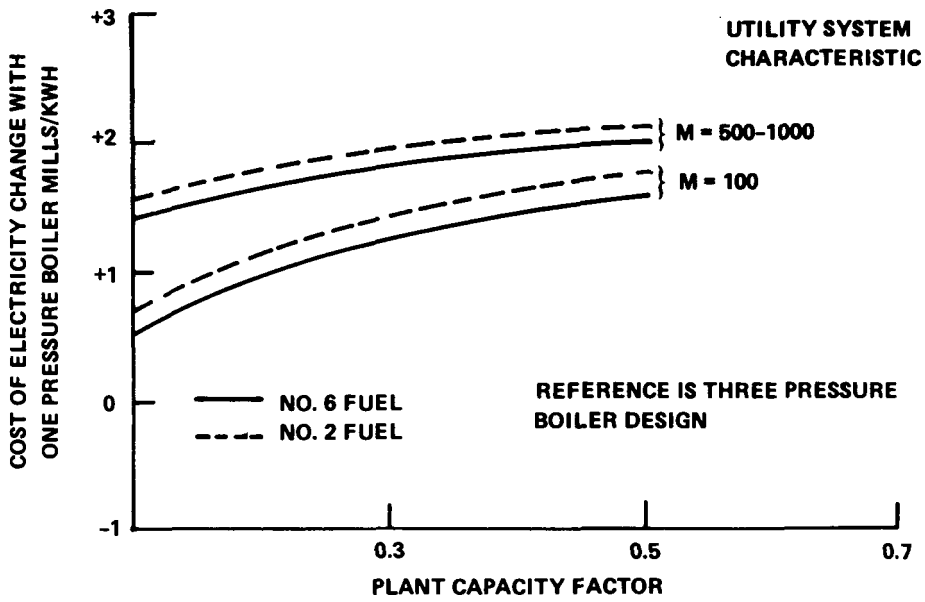


Figure 5-9 Effect of Plant Capacity Factor Using Single and Three Pressure Waste Heat Boilers - The cost of electricity with the single pressure system is significantly higher than that with the three pressure system. (See Appendix A for explanation of the change of cost of electricity background, including the parameter "M".)

Effect of Heavy Distillate Oil on Reliability

The trade off of using alternate fuels vs SSBL engine reliability was investigated separately since heavier distillate oil fuels could have various detrimental effects on the gas turbine reliability. The factors that influence reliability related to fuel flexibility include heat release residence time, peak temperatures, and the probability of unburned hydrocarbons in the turbine section. Fuel atomization assisted by high pressure air is assumed in the reliability study.

The study was conducted for three different fuels, and two combustor configurations. The fuel types considered are No. 2 home heat (distillate), an intermediate grade no. 6, and fuel composed of No. 2 home heat plus intermediate

grade No. 6 blended to a Redwood 600 viscosity. The SSBL baseline engine combustor configuration and a translating burner case design were compared to determine whether it is any maintenance advantage.

An early trade off study which determined the choice of combustor configuration involved considering reliability vs. fuel flexibility and emission capabilities. At the outset it was pre-supposed that the HRGT engine should be designed to meet all proposed EPA emissions requirements for an engine starting operation in the mid-1980's. This left the question of fuel flexibility vs. reliability to be determined.

Two alternate fuels were selected for evaluation: Number 6 residual fuel oil and a blend of Number 6 with Number 2. Results from research and development programs funded by EPRI, EPA and NASA, the United Technologies FT50 program and United Technologies FT4 field experience were used to establish requirements for gas turbine design changes and fuel treatment. Information on fuel treatment and atomizing air systems was furnished by suppliers who specialize in these kinds of equipment.

The effect of the use of heavier fuels on gas turbine reliability was established by adjusting the maintenance frequency, based on field experience, to maintain the engine failure rate the same as for the light distillate design.

The results of this study are summarized in Table 5-3.

As shown in the table, the failure rates are the same as for No. 2 fuel, but the hot section inspection interval has been reduced and the time required for hot section inspection is greater with heavier fuels to accommodate replacement of hot section parts. The repair time associated with in-mount maintenance has also been increased to accommodate the burner cans for heavier fuels.

The data shown in the table is predicated on fuel treatment to reduce the concentrations of undissolved solids, sodium, and potassium. Therefore, the reliability of the fuel treatment system itself was evaluated. The results predicted that the treatment system would have a relatively minor effect on the forced outage rate. Two approaches were investigated for fuel treatment system reliability estimates.

TABLE 5-3. GAS TURBINE MODULE FAILURE RATES WITH ALTERNATE FUELS

	#2 Home Heat		#2 and #6 Blend		#6 Residual	
	Failure Rate	MTTR	Failure Rate	MTTR	Failure Rate	MTTR
Hot Section Interval	9000 hours with fuel nozzle cleaning as required		5700 hours		4500 hours	
Hot Section Inspection Time	60 hours		75 hours*		100 hours*	
Diffuser Case Module	8.6	73	8.6	73.1	8.6	73.1
Turbine Module	40.5	58	40.5	58	40.5	58
In mount Maintenance	40.3	52.3	40.3	56.2	40.3	56.2

Note:
1. Failure rate is per million operating hours
2. MTTR is in hours
3. Blended residual fuels assume special equipment for fuel washing, and introducing fuel additives
4. Blended fuels require specially designed fuel nozzles
5. Residual fuels require specially designed dual fuel nozzles

*Includes allowance for repairs during hot section inspection

In one approach, one of the six 2.25×10^6 gallon tanks on the site, assumed in the reference plant for a 30 day supply, is used as a treated fuel tank. With a 2.25×10^6 gallon treated fuel tank, the power plant can continue to operate despite failure in the fuel treatment system because the inventory exceeds the longest fuel treatment system component MTTR. With this approach the forced outage rate increases from 6.6% in the reference plant to 7.03% with the heavier fuel.

In the other approach the fuel treatment system delivers fuel to a 3,000 gallon day tank serving each engine fuel system. Due to redundancy in components within the fuel system the forced outage rate increased only slightly more, from 6.6% to 7.12%.

It was concluded from this brief study that fuel flexibility covering a wide range of distillate oils could be realized without an appreciable reliability penalty. This is true only if special provisions are made for (1) fuel treatment and (2) increased hot section maintenance frequency. Fuel flexibility from a broader perspective, including coal-derived fuels, is covered in Section 7.

Reliability Versus Engine Size

The reference Single Shaft Base Load gas turbine was sized to be in the 100 Mw power class, which has been assumed to be best-suited for the combined-cycle, base load application. It was the purpose of this study to investigate a range of smaller sized gas turbines as well and determine if an optimum size, with respect to both cost and plant availability exists. In considering the smaller sizes, the number of gas turbines was increased while net power plant output was held fixed. From the study results, a size class was selected as the basis for the conceptual layout design effort.

Two alternate engine sizes were selected for comparison with the reference Single Shaft Base Load engine. The two sizes are a 2/3 size engine and a 1/2 size engine. A conceptual engine design for each of these alternatives was prepared in the form of an aerodynamic flowpath. Performance, reliability, and cost were then established. For each of the two alternate engine sizes, a conceptual design was also developed for the overall combined cycle plant. Plant performance, reliability, and cost were used to determine cost-of-electricity and availability for comparison with the reference.

Table 5-4 compares performance, reliability, relative cost, and cost of electricity for the reference engine with the two alternate engine sizes. There are two cases shown for the 1/2 size engine. In one case, there is a waste heat boiler to match each gas turbine. In the other case, the exhaust from two 1/2 size engines are combined with a single boiler. Plant reliability and cost of electricity are presented for three cases associated with gas turbine repair parts procurement time of 0, 1, and 2 months. Changes in cost of electricity are presented for three cases of utility system characteristic (m).

The mean time between failures, MTBF, of the reference engine is 7855 hours, the 2/3 engine mean time between failures is 8170 hours and the 1/2 size engine mean time between failures is 5810 hours, reflecting a twin spool design. The resulting overall plant Equivalent Forced Outage Rate (EFOR) is shown in Figure 5-10. Overall plant availability is shown in Figure 5-11. Availability profiles for each of the engine size alternatives is presented in Figure 5-12.

Both the 2/3 size and the 1/2 size engine were found to have a higher cost of electricity than the reference Single Shaft Base Load. This is shown in Figure

Table 5-4 Results of Engine Size Study

	Reference Engine	2/3 Engine Size	1/2 Engine Size	1/2 Engine Size
PERFORMANCE				
Gas turbine units/site	2	3	4	4
Gas turbine unit size (mw)	122.5	75.56	56.31	56.31
ISO (LHV) simple cycle heat rate btu/kwh	9710	10046	10210	10210
Turbine exhaust temp, °F	983	956	969	969
Boilers per site	2	3	2	4
Steam plant output, mw	95	90.7	93.6	93.6
Total plant net output (NEMA) mw	301	280.8	282.0	282.0
ISO (HHV) combined cycle heat rate btu/kwh	7615	7821	7754	7754
RELIABILITY, AVAILABILITY-PERCENT				
Gas turbine MTBF, hours	7855	8170	5810	5810
Overall Plant EFOR				
Gas Turbine Parts Lead Time				
0	6.60	6.50	6.90	6.90
1 month	14.50	14.20	17.30	17.30
2 months	21.20	20.70	25.60	25.60
OVERALL PLANT EQUIVALENT AVAILABILITY				
Gas Turbine Parts Lead Time				
0	87.35	87.43	87.06	87.06
1 month	79.97	80.26	77.36	77.36
2 months	73.73	74.16	69.57	69.57
RELATIVE CAPITAL COST				
Gas turbine cost	1.0	0.94	0.81	0.81
Gas turbine plant cost	1.0	1.123	1.337	1.337
Steam plant cost	1.0	1.014	1.004	1.044
Overall plant cost	1.0	1.078	1.111	1.139
CHANGE FROM REFERENCE COST OF ELECTRICITY (Mills/kwh)				
Repair Parts Utility Lead Time Characteristic				
0 m=1000	0	+4.24	+4.01	+4.39
= 500	0	+4.26	+4.05	+4.43
= 100	0	+4.17	+4.28	+4.75
1 m=1000	0	+4.40	+4.93	+5.36
= 500	0	+4.41	+5.08	+5.52
= 100	0	+4.43	+6.85	+7.49
2 m=1000	0	+4.54	+5.83	+6.31
= 500	0	+4.54	+6.08	+6.59
= 100	0	+4.49	+9.13	+9.92

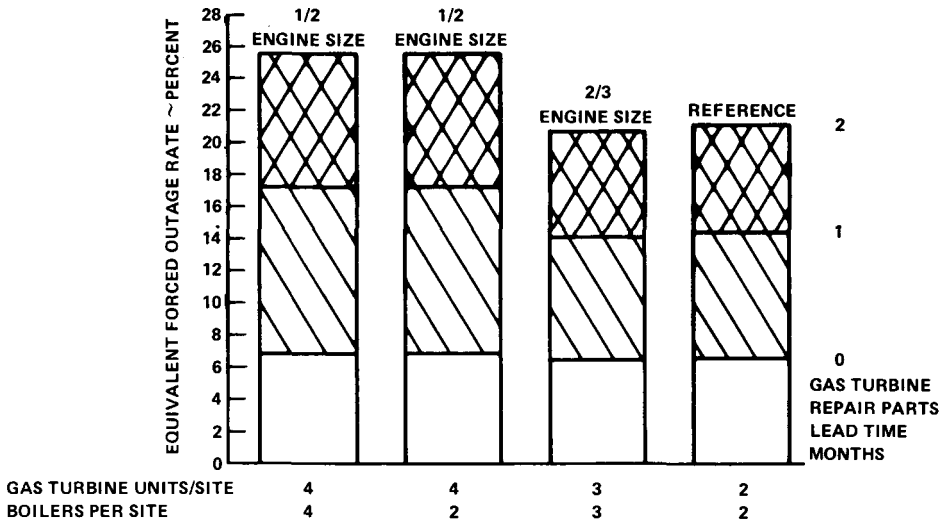


Figure 5-10 Effect of Engine Size and Repair Parts Lead Time On Equivalent Forced Outage Rate

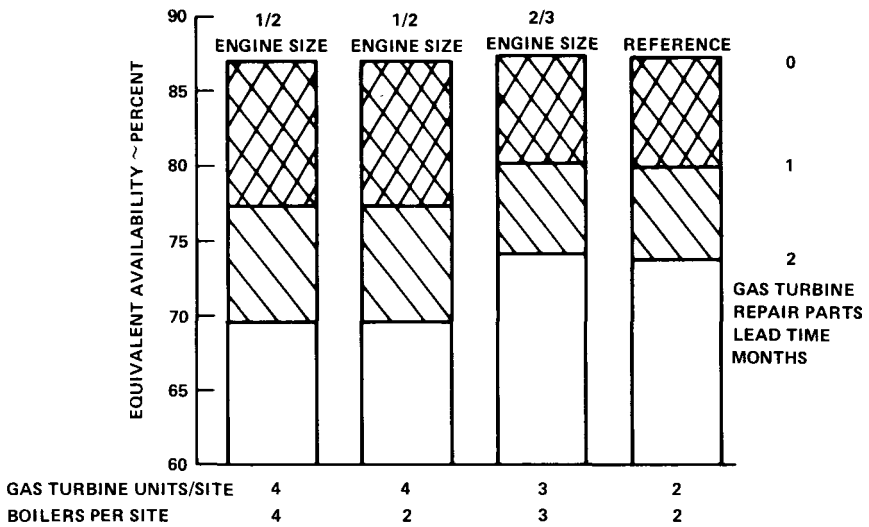


Figure 5-11 Effect of Engine Size and Repair Parts Lead Time on Equivalent Availability

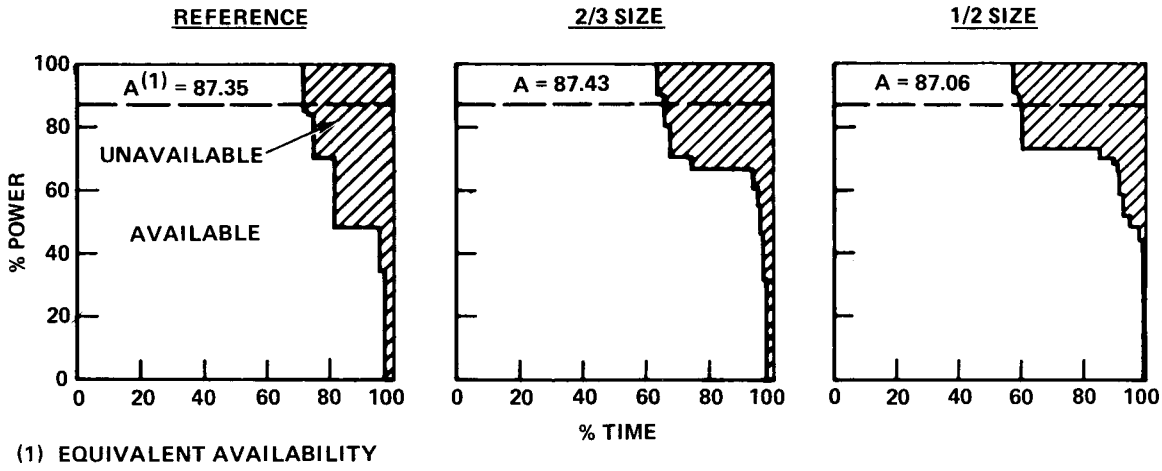


Figure 5-12 Availability Profile for Engine Size Alternatives

5-13. The plant arrangement which pairs a waste heat boiler with each 1/2 size gas turbine has a higher cost of electricity than the arrangement with one boiler behind two 1/2 size gas turbine.

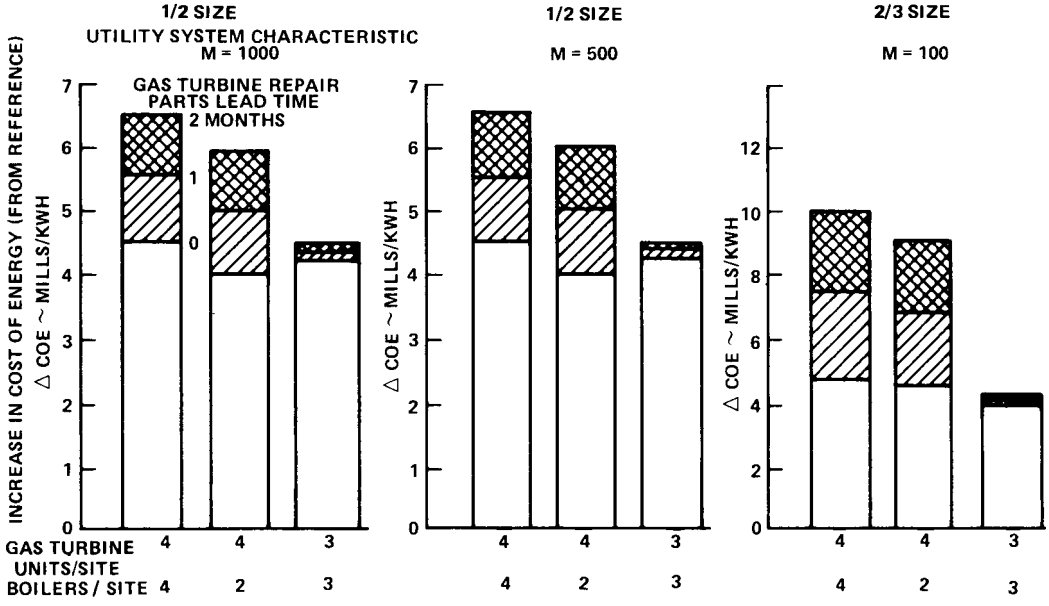


Figure 5-13 Effect of Engine Size and Repair Parts Lead Time on 30-Year Levelized Power Cost

It was established that a combined cycle plant with the 2/3 size engine has a 4 to 4.5 mill/kwh higher cost of electricity and the same availability as the reference engine. The 1/2 size engine plant has a 4 to 10 mill/kwh higher cost

of electricity than the reference engine and 0.3 percent to 4.2 percent lower plant availability, depending on the time assumed for spare parts procurement delay.

It was concluded from these results that an engine smaller than the Single Shaft Base Load offers no advantage. The benefit of even larger size, as limited by highway transportability or maximum allowable stress in the turbine last stage can be determined as the HRGT design layout progresses.

AERODYNAMIC STUDY RESULTS

Compressor

Aerodynamic Design Selection. A series of parametric studies covering a matrix of compressor design variables were conducted to provide the performance and configuration data necessary for the selection of optimum compressor configurations for use in the HRGT engine. A matrix of 81 cases covering three flow path configurations, three values of specific flow, three tip speeds and three aspect ratios was constructed as a basis for the study. This matrix is shown in Figure 5-14.

A statistical matrix of 25 cases was selected from the 81 parametric cases for the determination of the geometrical configuration and calculation of the compressor performance. In addition, the differential cost of electricity generated was calculated based on the SSBL engine as a reference.

The Pratt & Whitney Aircraft one-dimensional meanline compressor analysis program was used for the compressor performance analysis. This program utilizes the conical diffuser analogy supplemented with cascade correlations and predicts performance based on aspect ratio, thickness to cord ratio, solidity, Reynolds number, clearance, flowpath, and surface roughness. The results of this analysis are expressed in terms of efficiency and geometrical characteristics of the compressor including length, diameter, hub tip ratio, number of stages and number of airfoils.

The required results for the remainder of the 81 cases were then obtained by the application of the statistical regression analysis to the results of the 25 cases analyzed with the meanline program.

		AVG AR = 1.20			AVG AR = 1.50			AVG AR = 1.80		
	TIP SPEED	1150	1250	1350	1150	1250	1350	1150	1250	1350
COD	40.0	X		X				X		X
COD	37.5					X				
COD	35.0	X		X				X		X
CMD	40.0					X				
CMD	37.5		X		X	X	X		X	
CMD	35.0					X				
CID	40.0	X		X				X		X
CID	37.5					X				
CID	35.0	X						X		X
SPEC. FLOW										

X = CASES ANALYZED AERODYNAMICALLY
AR = ASPECT RATIO
CID = CONSTANT INNER DIAMETER
CMD = CONSTANT MEAN DIAMETER
COD = CONSTANT OUTER DIAMETER

Figure 5-14 Compressor Evaluation Matrix - A matrix of 81 possible combinations of design variables was analyzed using a statistical selection of 25 specific configurations.

The calculated efficiencies for the 25 cases are displayed in Figure 5-15.

Most of the high inlet specific flow cases can be ruled out on the basis of their low efficiencies. All of the 40 $lb/sec/ft^2$ cases and most of the 37.5 $lb/sec/ft^2$ cases were eliminated on this basis.

Bearing requirements were analyzed on the basis of the parameter L^2/D shown in Figure 5-16. This parameter is inversely proportional to the rotor stiffness and is important because its value is indicative of the number of engine bearings required to keep the rotor first critical speed above the operating

		AVG AR = 1.20			AVG AR = 1.50			AVG AR = 1.80		
	TIP SPEED	1150	1250	1350	1150	1250	1350	1150	1250	1350
COD	40.0	(87.4) 9		(86.3) 7				(87.3) 10		(86.3) 7
	37.5					(88.0) 8				
	35.0	(89.1) 9		(88.8) 7				(88.8) 10		(88.5) 8
CMD	40.0					(87.4) 9				
	37.5	(88.9) 10	(88.3) 9		(88.8) 11	(88.0) 10	(87.4) 8		(87.8) 10	
	35.0					(88.5) 11				
CID	40.0	(88.4) 11		(86.8) 10				(88.4) 12		(86.6) 10
	37.5					(88.7) 12				
	35.0	12		(88.7) 11		(89.3) 13		13		(88.1) 12
	SPEC. FLOW									

9 = NUMBER OF STAGES

Figure 5-15 Efficiencies of Parametric Compressor Configurations - The data show that the configurations with high inlet specific flow provide low efficiencies.

range. For a two bearing engine rotor, the value of L^2/D for the compressor cannot exceed a value of about 150 inches.

On the basis of these data, all constant ID cases were ruled out because of excessive values of L^2/D. In these cases, the low wheel speed in the rear stages necessitates the use of a large number of stages to attain the required overall pressure ratio. This results in a long compressor and an excessively large value of L^2/D.

		AVG AR = 1.20			AVG AR = 1.50			AVG AR = 1.80		
	TIP SPEED	1150	1250	1350	1150	1250	1350	1150	1250	1350
COD	40.0	(119)	(40)					(81)		(23)
	37.5					(51)				
	35.0	(129)		(44)				(88)		(33)
CMD	40.0					(89)				
	37.5	(251)	(134)		(217)	(118)	(53)		(91)	
	35.0					(156)				
CID	40.0	(364)		(162)				(351)		(116)
	37.5					(309)				
	35.0	(1315)		(262)		(435)		(789)		(168)
	SPEC. FLOW									

Figure 5-16 L^2/D Values for Parametric Compressor Configurations - Values below 150 are required for a two-bearing configuration.

The results of the Cost of Electricity analysis are given in Figure 5-17. These results are expressed as the differential between the cost of electricity generated in the SSBL engine as designed and the same engine in which the original compressor is replaced by the subject compressor. Values are quoted in mills per kilowatt hour. They are also approximately equal to the differential in percentage points since the cost of electricity generated by the SSBL engine is about 90 mils per kilowatt hour.

It can be seen that the lowest aspect ratios and lowest tip speeds give rise to the lowest electricity costs and the smallest number of airfoils. However, the limit of somewhere between 125 and 150 inches imposed on L^2/D by the

		AVG AR = 1.20			AVG AR = 1.50			AVG AR = 1.80		
	TIP SPEED	1150	1250	1350	1150	1250	1350	1150	1250	1350
COD	40.0	(.170)		(.700)				(.341)		(.787)
COD	37.5					(–.117)				
COD	35.0	(–.705)		(–.595)				(–.434)		(–.354)
CMD	40.0					(.223)				
CMD	37.5	(–.583)	(–.286)		(–.457)	(–.067)	(.200)		(–.092)	
CMD	35.0					(–.316)				
CID	40.0	(–.343)		(.509)				(–.24)		(.724)
CID	37.5					(–.387)				
CID	35.0			(–.455)		(–.672)				(–.011)
SPEC. FLOW										

Figure 5-17 Cost of Electricity - Values represent the difference in cost relative to the cost for the SSBL engine.

two-bearing engine requirement restricts the minimum aspect ratio which can be selected at a given tip speed for each compressor configuration and inlet specific flow.

Carpetplots of the results from all of the 81 cases were prepared. These plots consist of curves of constant COE, L^2/D, and number of airfoils, plotted on an aspect ratio versus tip speed coordinate system. Four such plots are displayed in Figures 5-18 through 5-21. Data for other configurations was not plotted on the basis of the results obtained for efficiency and L^2/D values.

From the matrix of 81 cases, three compressor configurations were selected as representative of the best performance attainable consistent with the technical

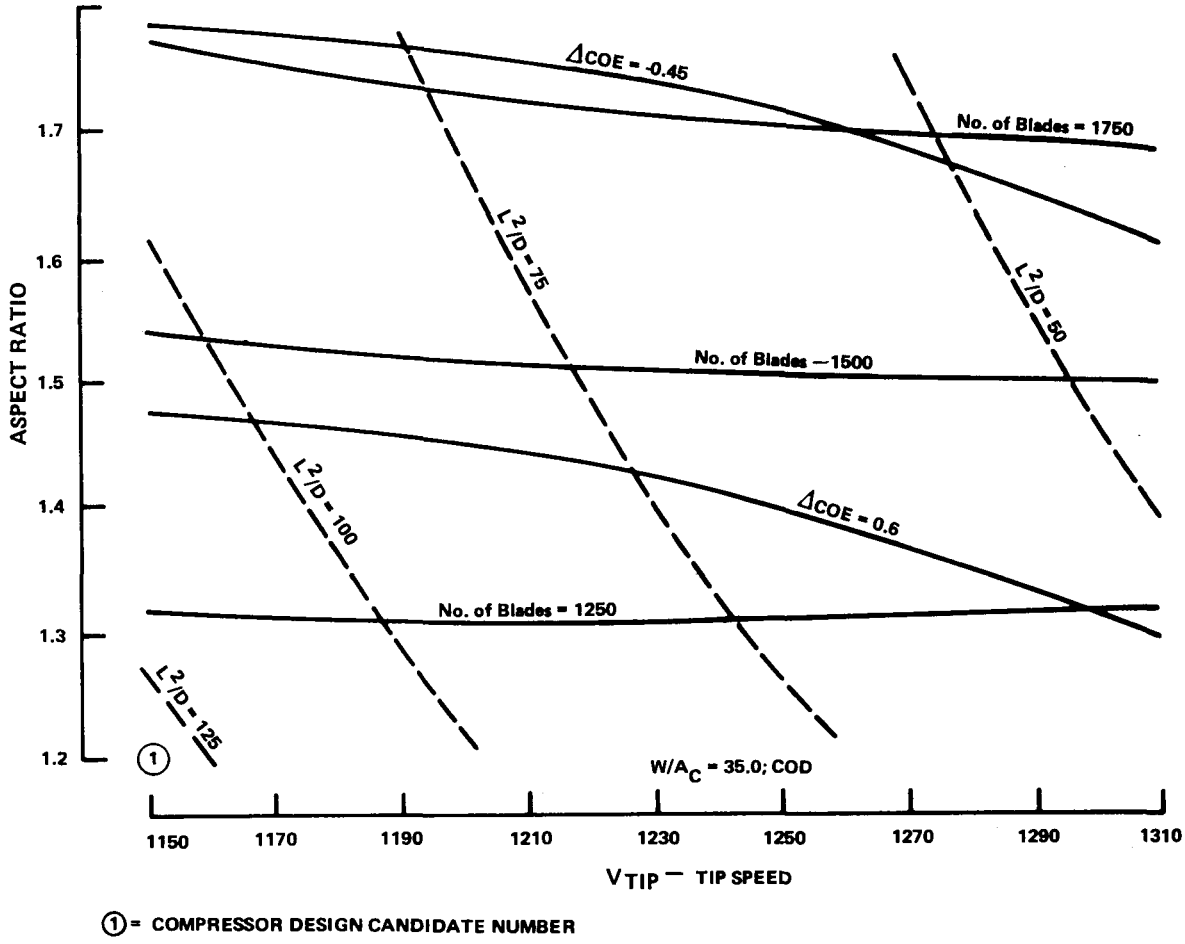


Figure 5-18 Carpet Plot for Parametric Compressor Configurations with Constant Outer Diameter and Flow Rate of 35 lb/sec ft^2.

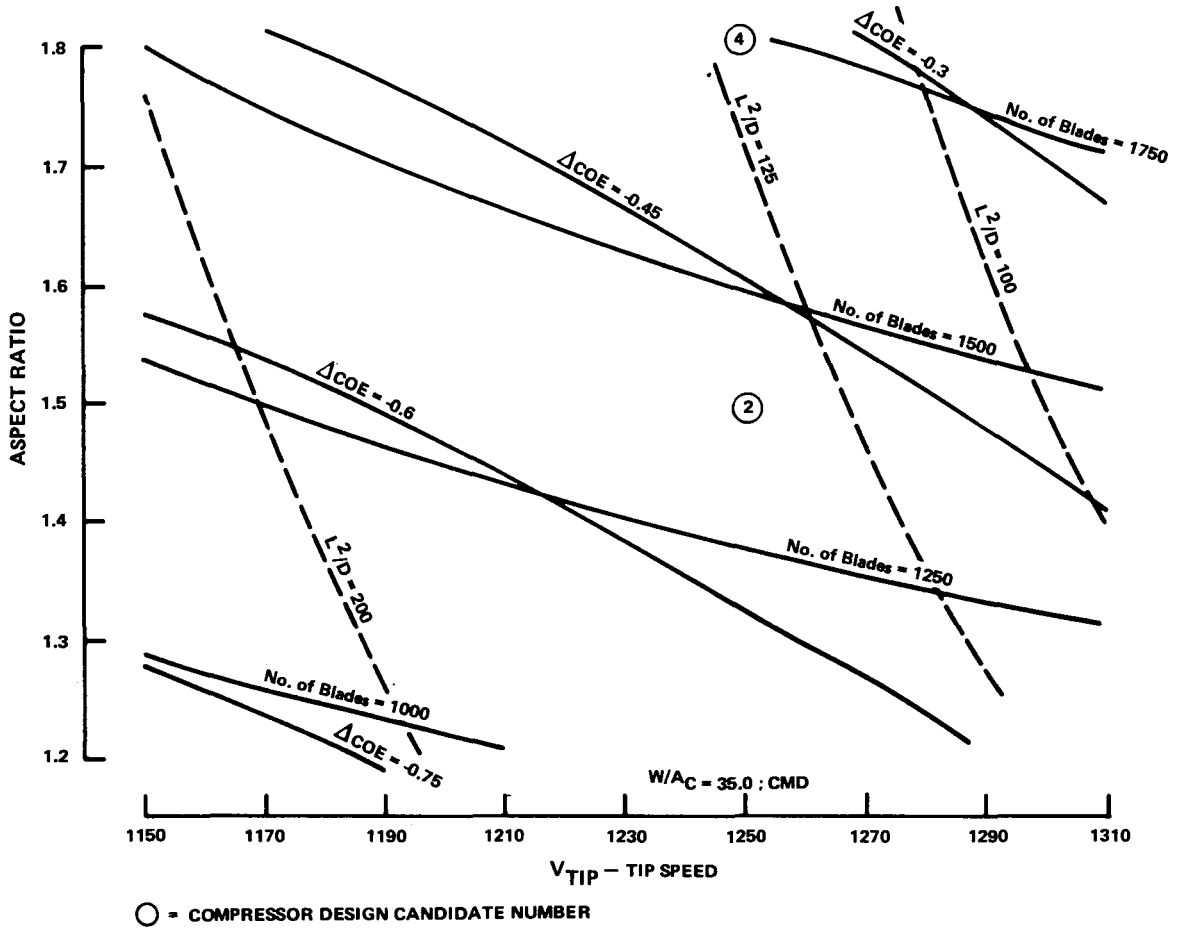


Figure 5-19 Carpet Plot for Parametric Compressor Configurations with Constant Mean Diameter and Flow Rate of 35 lb/sec ft^2.

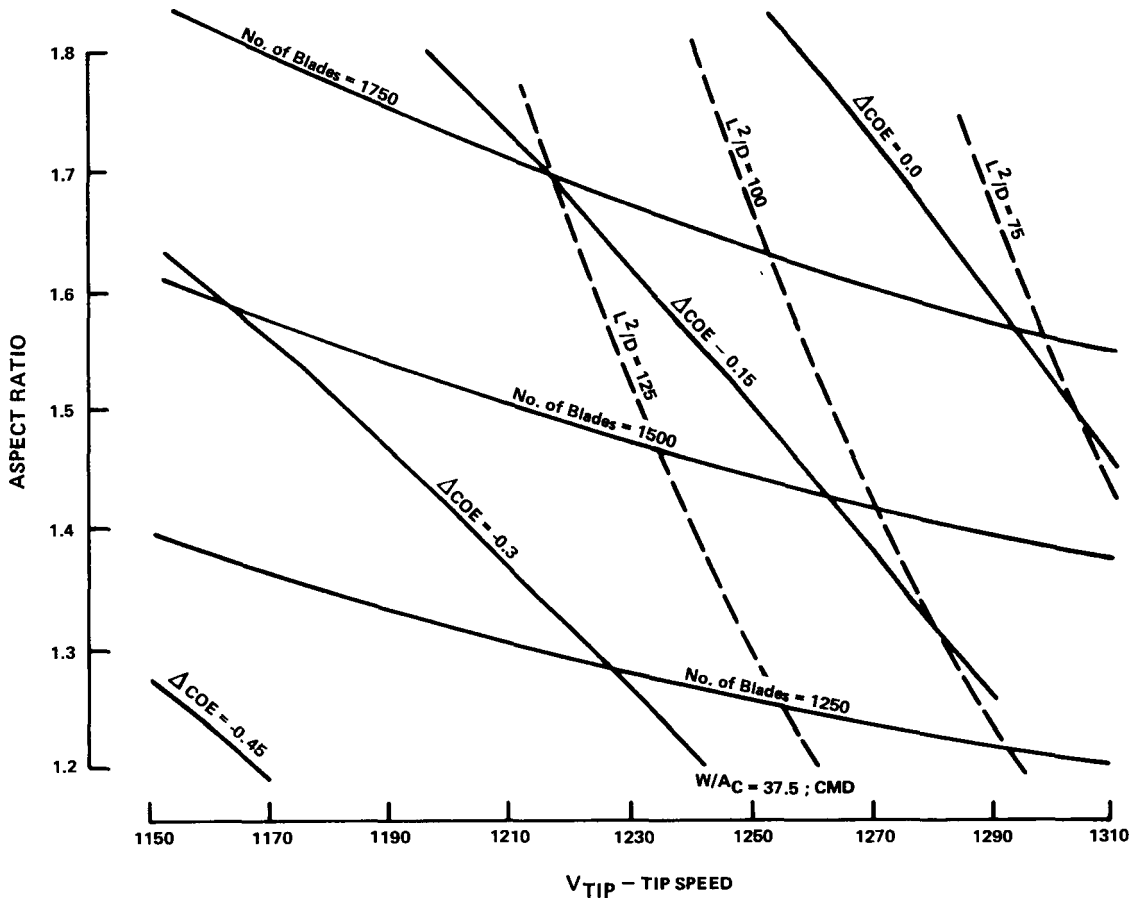


Figure 5-20 Carpet Plot for Parametric Compressor Configurations with Constant. Mean Diameter and Flow Rate of 37.5 lb/sec ft^2.

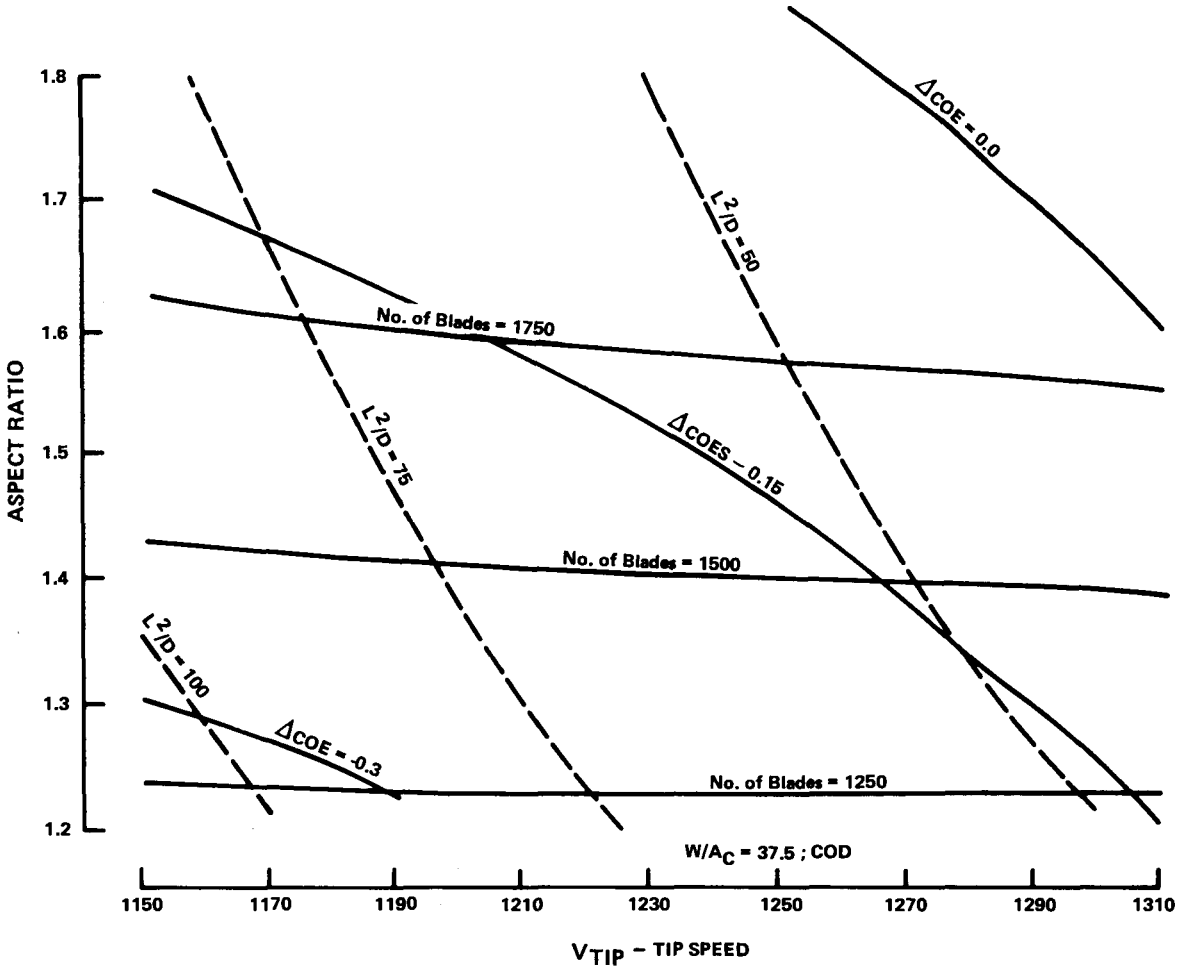


Figure 5-21 Carpet Plot for Parametric Compressor Configurations with Constant Outer Diameter and Flow Rate of 37.5 lb/sec ft^2.

risk level for the design over a range of technical risk levels. A fourth free turbine case was also calculated for use in the conceptual design of a free turbine version of the engine for evaluation relative to the single shaft engine concepts.

These four candidate compressor configurations are displayed in Table 5-5. The first candidate was selected on the basis of offering the lowest COE of any case considered; however, it must be pointed out that the COE is relative insensitive to aspect ratio and tip speed. This compressor has an aspect ratio of 1.2 and a hub tip ratio of 0.94. The very low aspect ratio and high hub tip ratio of this design give rise to a relatively high level of risk in the reliability of the

performance prediction for this case. This selection is plotted on Figure 5-18. Note that the L^2/D for this configuration is about 130 inches. This compressor is a constant OD design with an adiabatic efficiency of 89.1%.

TABLE 5-5. COMPRESSOR DESIGN CANDIDATES

CANDIDATE NO.	1	2	3	4
CONFIGURATION	Single Shaft	Single Shaft	Free Turbine	Single Shaft
FLOW PATH	COD	CMD	CMD	CMD
FLOW/AREA	35	35	37.5	35
ASPECT RATIO	1.2	1.5	1.2	1.8
HUB/TIP RATIO (LAST STAGE)	0.94	0.91	0.90	0.90
TIP SPEED	1150	1250	1350	1250
NO. OF STAGES	9	11	11	14

The second candidate selected is a constant mean diameter configuration which is only slightly inferior in terms of cost of electricity to the first selection but carries a considerably reduced risk in the attainability of the predicted performance. In this case, the aspect ratio is within the range of available experimental data and the hub tip ratio has a more conservative value of 0.91, which results from the constant mean diameter design. This candidate is plotted on Figure 5-19. The L^2/D for this case is very close to that for the first candidate, about 135 inches.

The third candidate (designated Candidate Number 4 in Table 5-5 to be consistent with the candidate engine design designations) was chosen as representative of conventional technology. It has an aspect ratio of 1.8 and a conservative hub-to-tip ratio of 0.90. The efficiency of this configuration is also excellent. The cost of electricity is only about .3% higher than for the first candidate chosen. The performance predictability risk in this case is very low since this configuration is based on an enormous background of available technology. This configuration is also a constant mean diameter configuration.

The free turbine compressor design was not one of the 81 matrix cases but rather one of two special cases which were analyzed. This compressor is designated Candidate Number 3 in Table 5-5. In this case, the compressor shaft speed is not limited by the synchronous speed of the generator and a high speed design was selected to take maximum advantage of this situation. A constant mean diameter, low aspect ratio configuration was chosen. The technical risk in this case is lower than for the first single shaft candidate because the high rotational speed permits the use of a reasonably low hub tip ratio in the high pressure stages.

High Through Flow Concept

The parametric studies for the selected HRGT compressor configuration were based on a conventional exit Mach number of 0.35. Similar matrix type parametric studies were also carried out for exit Mach numbers of 0.45 and 0.55, the so-called "high through flow" regime. The results of these studies failed to show significant benefits in efficiency over the conventional Mach 0.35 designs. However, there is indication that further optimization of the high through flow configuration might show greater benefits.

The range of variables covered in the high through flow parametric studies was approximately the same as for the Mach 0.35 design, except that a variation in exit Mach of 0.35, 0.45, and 0.55, with fixed flow path shape was substituted for variations in flow path shape with fixed Mach number. The complete matrix is shown on the chart in Figure 5-22.

For each case, the flow path was held to a constant mean diameter shape. This constraint was necessitated by limiting exit hub-to-tip ratio considerations. As a general rule, an increase in exit Mach number level in the rear most stages reduces the flow passage height, ultimately reaching an unacceptable value. Even with the inner hub diameter for the aft stages kept near the centerline, as would be the case for a constant mean diameter flow path shape, exit hub tip ratios tend to exceed 0.94 at exit Mach numbers beyond 0.45 and tip speeds over about 1200 ft/sec, and designs with hub-to-tip ratios in this range suffer severe aerodynamic efficiency penalties because the tip clearances become a significant portion of the passage height. The effect of Mach number of hub-to-tip ratio is illustrated in Figure 5-23.

SPEC. FLOW	TIP SPEED	AVG AR = 1.20 1150	AVG AR = 1.20 1250	AVG AR = 1.20 1350	AVG AR = 1.50 1150	AVG AR = 1.50 1250	AVG AR = 1.50 1350	AVG AR = 1.80 1150	AVG AR = 1.80 1250	AVG AR = 1.80 1350
.55	40.0	X		X				X		X
	37.5					X				
	35.0	X		X				X		X
.45	40.0					X				
	37.5		X		X	X	X		X	
	35.0					X				
.35	40.0	X		X				X		X
	37.5					X				
	35.0	X		X				X		X

Figure 5-22 High Through Flow Compressor Study Matrix

Two typical design cases taken from the study are illustrated in Figure 5-24 along with a high through flow design representing a separate free power turbine spool arrangement. Generally, higher through flow Mach numbers permit fewer stages for a given flow path shape. High Mach number, however, cannot be used in conjunction with a constant outer flowpath diameter because of the high consequent hub-to-tip ratios. In contrast, with lower Mach numbers, a constant outer flowpath diameter can be used without exceeding the hub-to-tip ratio limitation, and the increase in mean flowpath diameter provides increased stage loading as a result of the higher wheel speed. The net effect, therefore, is that the high through flow design does not provide much reduction in the number of stages.

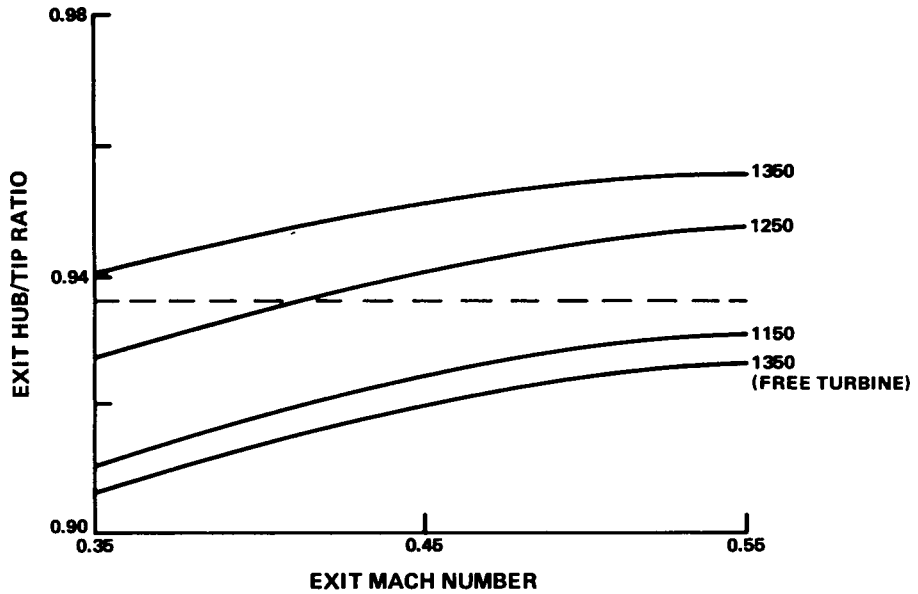


Figure 5-23 Effect of Mach Number on Exit Stage Hub-to-Tip Ratio

A further effect of the high through flow Mach number is additional aerodynamic losses over and above the hub-to-tip ratio effect, at the expense of efficiency. This is shown in Figure 5-25, which presents calculated efficiencies for the complete matrix.

Finally, as a key design parameter, L^2/D results are presented in Figure 5-26, and numbers of stages are presented in Figure 5-27, as calculated for the complete matrix of 25 cases. L^2/d trends tend to follow the same pattern regardless of Mach number; that is, flow path shape, tip speed, and aspect ratio tend to exert the dominant influence. Consequently, higher through flow Mach number provides no special advantage over the base level of Mach 0.35.

As an overall conclusion, it is clear from the parametric results that the aerodynamic efficiency penalty associated with design for an exit Mach number of 0.55 is excessive for any possible cost or reliability benefits. However, the trade off between a constant outer diameter design at Mach 0.35 (which was the level selected for the final HRGT design) and a constant mean diameter design at about Mach 0.45 is not so clearly established. The choice of narrowing to a more optimum combination must remain as the subject for more detailed study and evaluation, as is being considered under the DOE-sponsored contract now being undertaken.

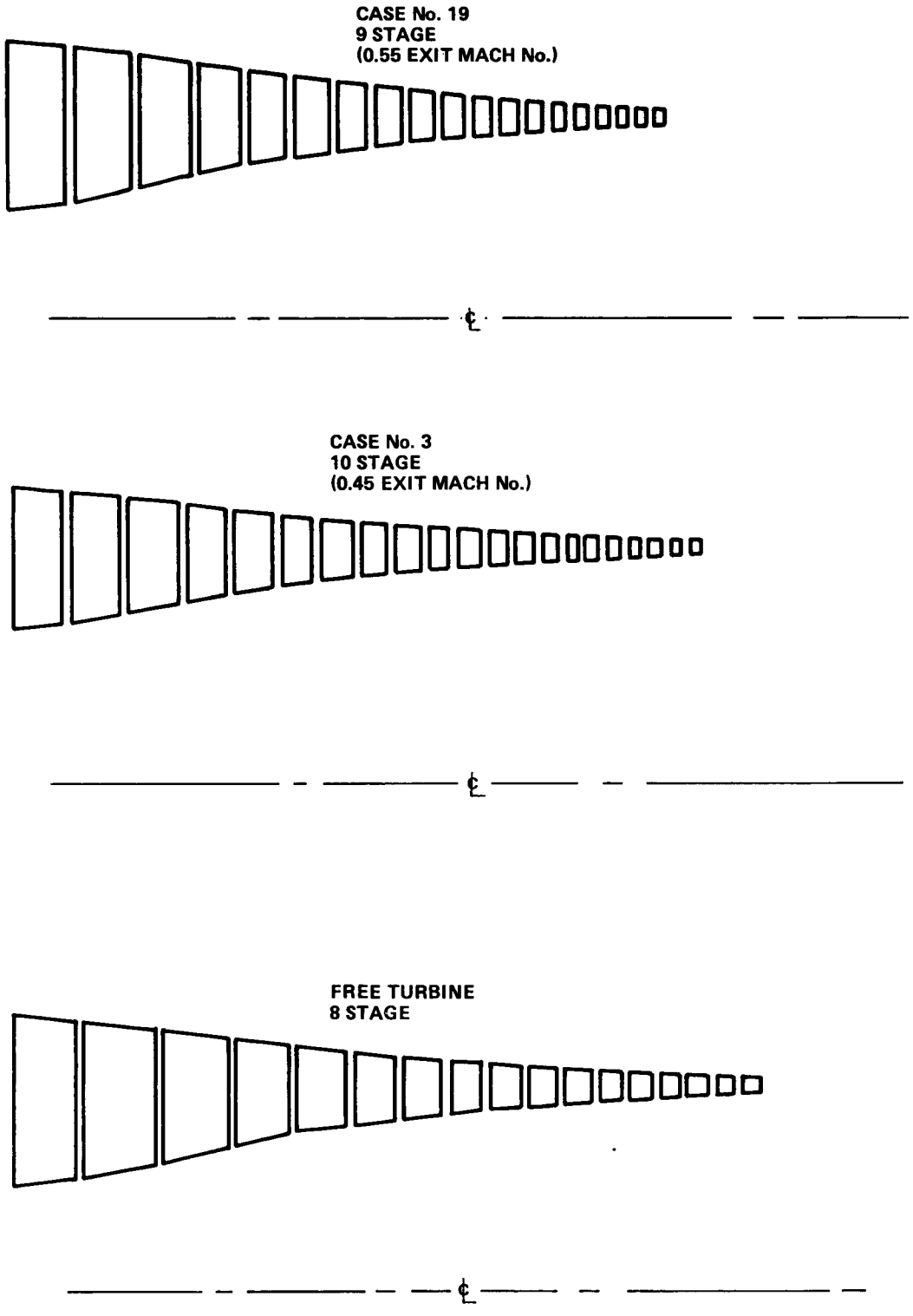


Figure 5-24 High Through Flow Compressor Flow Path Comparison

EXIT MACH No.	SPEC FLOW	AVG AR = 1.20 TIP SPEED 1150	1250	1350	AVG AR = 1.50 1150	1250	1350	AVG AR = 1.80 1150	1250	1350
.55	40.0	.869		.850				.864		.848
	37.5					.869				
	35.0	.883		.866				.886		.866
.45	40.0					.868				
	37.5		.877		.883	.877	.869		.876	
	35.0					.882				
.35	40.0	.881		.866				.874		.863
	37.5					.883				
	35.0	.897	.888					.886		.881

Figure 5-25 High Through Flow Compressor Study Matrix Showing Efficiency Values

EXIT MACH No.	SPEC FLOW	AVG AR = 1.20 TIP SPEED 1150	1250	1350	AVG AR = 1.50 1150	1250	1350	AVG AR = 1.80 1150	1250	1350
.55	40.0	115.4		40.1				76.4		29.6
	37.5					73.2				
	35.0	190.3		58.8				124.7		41.5
.45	40.0					68.2				
	37.5		154.3		173.3	91.8	51.0		71.5	
	35.0					126.6				
.35	40.0	226.6		69.1				150.4		50.3
	37.5					119.8				
	35.0	351.9	80.2					263.7		57.5

Figure 5-26 High Through Flow Compressor Study Matrix Showing L^2/D Values

EXIT MACH No.	SPEC FLOW	AVG AR = 1.20 TIP SPEED 1150	1250	1350	AVG AR = 1.50 1150	1250	1350	AVG AR = 1.80 1150	1250	1350
.55	40.0	7		6				8		7
	37.5					8				
	35.0	8		7				9		8
.45	40.0					8				
	37.5		8			9	8		9	
	35.0					10				
.35	40.0	10		8				11		9
	37.5					10				
	35.0	11		8				13		9

Figure 5-27 High Through Flow Compressor Study Matrix Showing Number of Stages

Compressor Reliability Parametric Study

Reliability calculations were made for the statistical matrix of 25 configurations discussed previously.

Reliability was evaluated by applying factor analysis techniques to the trade study results. Included were the number of stages, number of airfoils, total axial length, rotor diameter, and efficiency.

The factor analysis technique consisted of assigning weighted reliability "goodness" values to selected parameters, based on how they impacted foreign object damage, vibration, airfoil erosion, etc. Then the "goodness" values for

each case were summed, providing 25 relative reliability indicators for each trade study. When ranked, the best to worst configurations in the matrix became evident. Table 5-6 shows how and why each parameter was evaluated for relative reliability "goodness".

TABLE 5-6 COMPRESSOR RELIABILITY CONSIDERATIONS

PARAMETER	RANGE OF PARAMETER	IMPACT ON RELIABILITY	PARAMETER/ RELIABILITY RELATIONSHIP	EFFECT OF USING HIGH VALUE OF PARAMETER
*Inlet Specific Flow	35 to 40	small	inverse	More transonic airfoils with thin LE tips; more erosion.
*Average Aspect Ratio	1.2 to 1.8	large	inverse	**Thinner, more fragile, more numerous airfoils, cracks, FOD, erosion.
Number of Stages	6 to 13	medium	inverse	More change for cracking, FOD and erosion.
Number of Airfoils	482 to 2267	medium	inverse	Ditto. Also more thinner airfoils.
Length^2/Rotor Diam. (L^2/D)	23 to 1315	med.-small	inverse	Shaft critical speed parameter-affects stiffness & vibr. damage.
Efficiency	.863 to .891	none	-	No R impact
*Profile Geometry	Constant ID, MD & OD	none	-	No R impact (See other parameters)
Chord Length	52 to 200	none	-	No R impact (See L^2/D)
*Blade tip speed	1150 to 1350	none	-	No R impact

* Independent variables comprising the matrices.

** Conversely, low aspect ratio airfoils have more blade curvature with thicker and stronger sections having large LE radii. These features provide more flutter resistance as well as resistance to FOD, erosion and cracking. The low aspect ratio also shortens compressor axial length, contributing to rotor stiffness which helps avoid critical speed/vibration.

In these trade studies, it is assumed that the heavy construction cases and disks are equally reliable for every configuration because the aerodynamic parameters used have little influence on case and disk design features. However, the relative reliability of various case and disk designs is reported in the assessment of the five HRGT concepts. Figure 5-28 and Figure 5-29 show the relative reliability "goodness" factors as calculated for the original Mach 0.35 and high through flow matrices, respectively.

Combustor

Alternate Fuels. The HRGT engine is due to start service in the mid 1980's and, presumably, to continue a useful life well in the 1990's. Consequently, the engines' combustion system design must anticipate a wide latitude of possible fuels, with coal-derived liquids or gases of prime interest. Recognizing this need, the Phase I conceptual studies included a separate task which considered how best to allow for the coal derivative fuels in the HRGT. The results

BASELINE MATRIX No. 1 (EXIT MACH No. = 0.34 = CONSTANT)

GEOM. SPEC. FLOW		AVG AR = 1.20			AVG AR = 1.50			AVG AR = 1.80		
	TIP SPEED	1150	1250	1350	1150	1250	1350	1150	1250	1350
COD	40.0	53.4		55.7				32.2		35.0
	37.5					46.6				
	35.0	57.3		60.0				36.2		39.6
CMD	40.0					41.0				
	37.5		52.9		40.2	42.0	42.2		31.8	
	35.0					43.3				
CID	40.0	47.6		47.6				27.1		27.4
	37.5					37.0				
	35.0	46.3		50.2				29.6		29.3

HIGHER FACTORS ARE IN DIRECTION OF BETTER RELIABILITY

Figure 5-28 Compressor Reliability Study Matrix Results for Mach 0.35 Design

included a comprehensive survey of potential fuel characteristics and associated on-going R&D programs. This work is presented separately in Section 7. Our conclusions relative to the HRGT design can be condensed to a few general observations.

Regarding the coal-derived gases, wide experience with natural gas in gas turbine engines has proved that the even relatively low Btu gases can be readily handled by present day combustion technology. This has been further verified by actual full-scale combustor rig tests using low Btu gas produced by a prototype coal gasifier. Allowance in the design need only be made for the increased space needed to feed the high volume gaseous fuel into the combustion chamber. This was done for the selected design.

HIGH-THROUGH-FLOW MATRIX No. 2 (GEOM. = CMD = CONSTANT)

EXIT MACH No.	SPEC FLOW	AVG AR = 1.20 TIP SPEED 1150	1250	1350	AVG AR = 1.50 1150	1250	1350	AVG AR = 1.80 1150	1250	1350
.55	40.0	42.8		43.5				22.2		21.6
	37.5					33.3				
	35.0	44.8		46.4				23.8		23.7
.45	40.0					36.0				
	37.5		47.4		35.2	36.4	37.6		25.8	
	35.0					37.6				
.35	40.0	47.7		51.9				26.5		30.3
	37.5					40.7				
	35.0	48.3		55.9				27.0		34.0

Figure 5-29 Compressor Reliability Study Matrix Results for High Through Flow Design

Coal-derived liquids impose much greater problems than the gases. However, as these problems relate to the internal combustor, the rich burn/quick quench technology has the potential for overcoming them. Equipping the HRGT with a rich burn/quick quench combustor, as is the case in our recommended design, provides the engine with the widest latitude of fuel flexibility.

Candidate Combustor Configurations. There are two basic types of onboard burners (can and annular) with three combinations or orientations (inline, offset axial and offset canted). The inline and offset canted orientations are shown schematically in Figure 5-30.

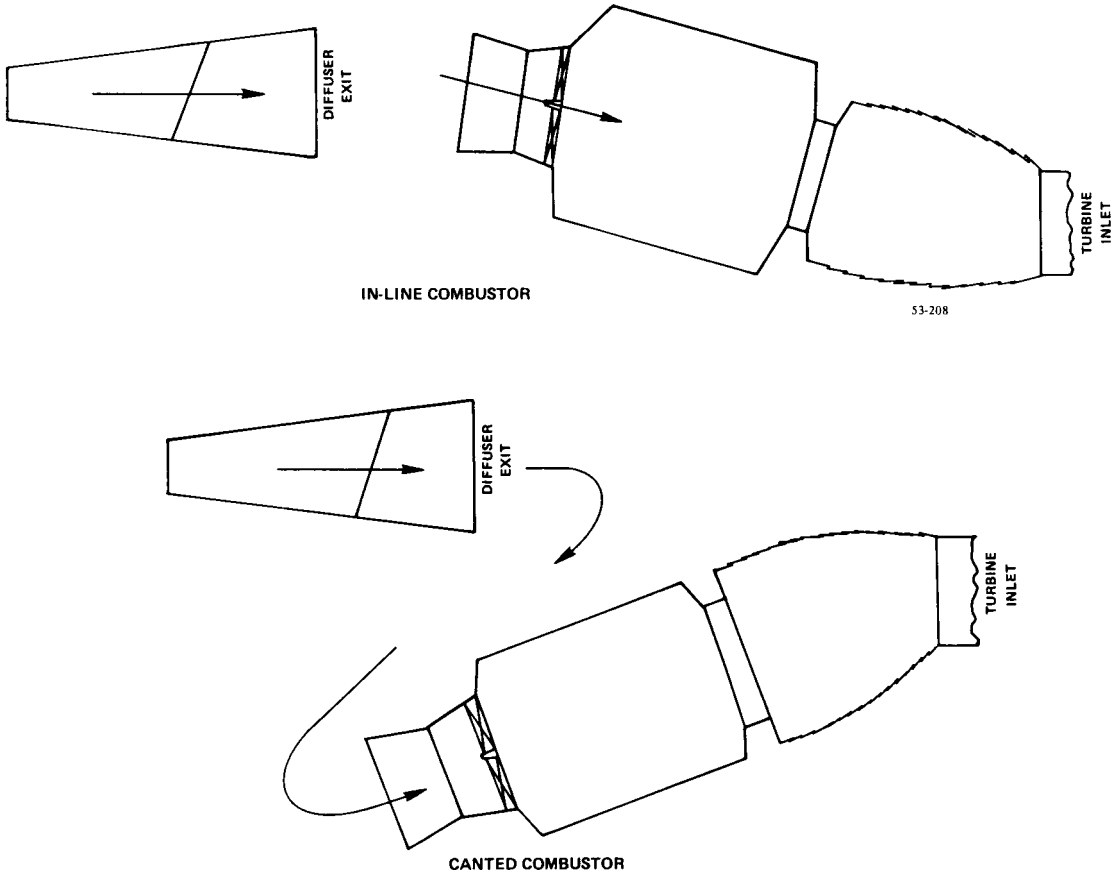


Figure 5-30 Inline and Canted Combustor Configurations - Both types of combustor were considered in the combustor trade studies.

As the name implies, an inline burner is centered about a straight line between the diffuser exit and the turbine inlet. The inline orientation captures diffuser exit total pressure which can be used to atomize fuel and enhance front end mixing. Inline combustors can add weight and length (which can result in additional bearings) if the combustion process employed requires large volume and/or length for completion.

The offset canted orientation is used when the available envelope is insufficient to accommodate an inline combustor with the volume necessary to complete the combustion process. Air entering the front of an offset canted burner has to make essentially two 180 degree turns, eliminating the possibility of capturing diffuser exit total pressure. Offset canted burners are more accessible and minimize overall engine length.

An inline annular burner provides more volume per unit surface area than inline can burners; however, the can has the higher volume to area ratio in the canted orientation. Can burners are generally easier and less expensive to develop than annular burners because a single can rig or segment of an engine can be used to evaluate modifications as opposed to a full annulus for the annular burner. Flame propagation presents more of a problem with can burners.

An offset axial burner can be used to gain burner length, without increasing overall engine length, when the turbine mean diameter is larger than the compressor mean diameter. This orientation does not capture diffuser exit total pressure and generally does not provide the unlimited length/volume envelope of the canted configuration.

The selection process used in the combustor analysis is illustrated in Figure 5-31. As shown, it involved selection of the basic type followed by selection of the basic configuration and finally the liner cooling scheme. Meanwhile, ignition schemes were studied and selected in an independent study.

Of the possible combinations of burner types and orientations, four were chosen for detailed study (in-line annular, canted annular, in-line can, and canted can). Table 5-7 presents the essential design features for each of the configurations.

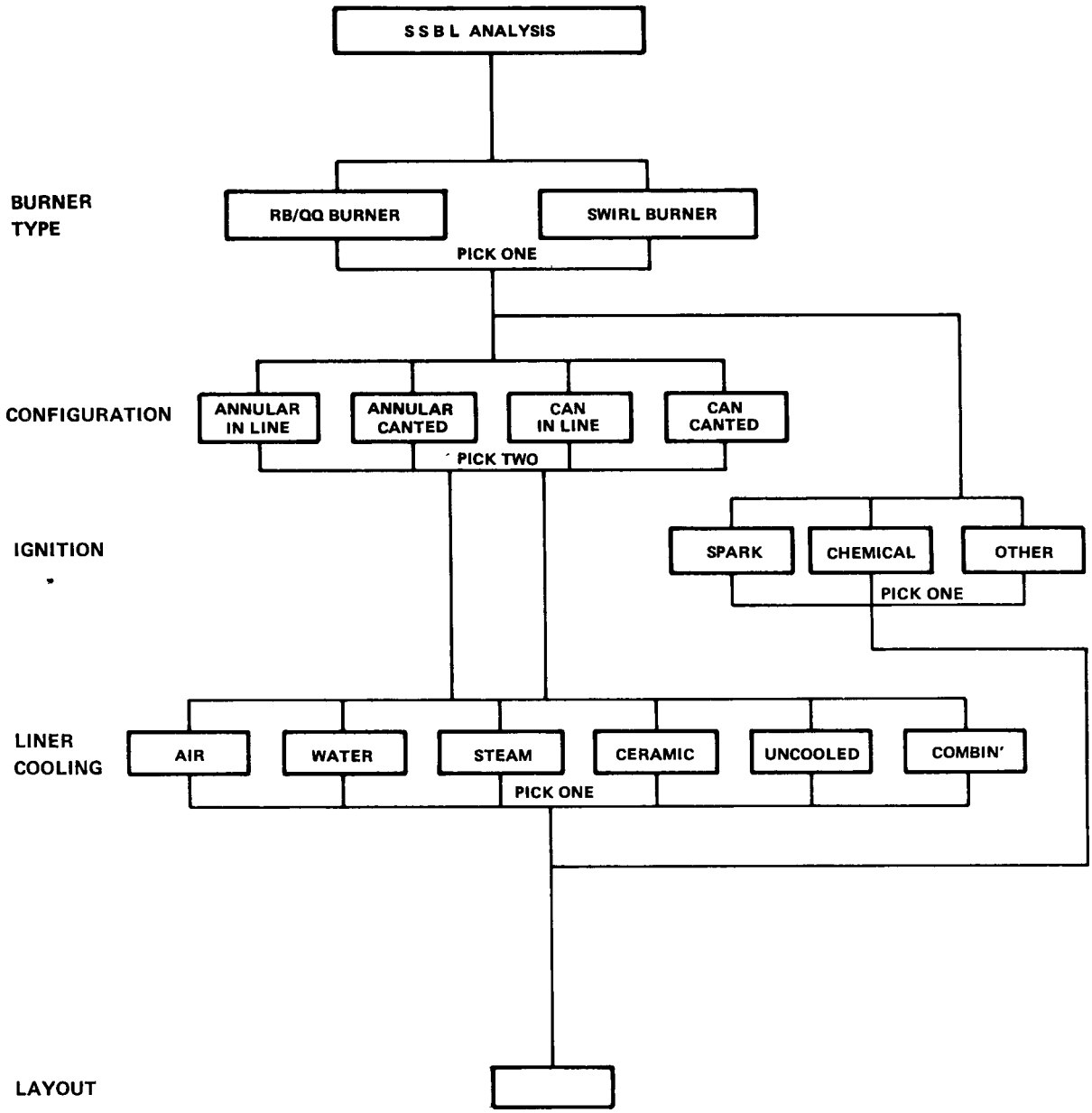


Figure 5-31 Combustor Selection Process - A sequential multistep process was used.

The design study revealed that the offset axial annular could not provide the necessary volume. The offset canted cans were chosen over the offset canted annular because they provided the larger volume for a given surface area.

TABLE 5-7. COMBUSTOR CONFIGURATION DESIGN SUMMARY

	COMBUSTOR CONFIGURATIONS			
	In-Line Annular	Canted Annular	In-Line Can	Canted Can
Length, IN	57.5	44.6	70.56	54.0
VOL_{main}, FT^3	18.9	20.4	18.5	18.5
VOL_{lean}, FT^3	15.7	14.8	35.9*	37.9*
VOL_{total}, FT^3	34.7	35.2	54.4*	56.3*
V_{main}, fps	23.7	22.0	41.8	41.8
t_{main}, sec	0.042	0.045	0.041	0.041
V_{lean}, fps	95.6	81.4	55.9	53.0
t_{lean}, sec	0.0105	0.0112	0.0239	0.025
$A_{surface}$, IN^2 main	5838	7057	8128	8128
$A_{surface}$, IN^2 lean	7040	7518	8889*	10050*
$A_{surface}$, IN^2 total	12878	14576	17018*	18179*

*Approximate due to nature of calculation.

A third possible type of burner, the external burner was also considered. Though United Technologies has had no experience with external burners, this was considered a viable alternate. An external burner would permit a large volume to be formed without a length penalty to the engine and the longer flowpath and combustion time would permit better gas mixing, thus a lower pattern factor. Further, the external burner could be cooled with steam or water.

Ignition Techniques. Two proven techniques were investigated. The first involves the use of cross-over tubes which rely on flow between two cans for ignition. This technique has had durability problems in the past.

The second technique employs igniters in each can and relies on redundancy for reliability. Two igniters are used for each can, requiring a total of twelve ignition systems. Each system is connected to two cans, and no two cans are

conected to the same system. A system for verifying ignition is included. The use of redundant igniters and ignition verification results in a more reliable engine.

Turbine

The turbine design evolved from a parametric study which examined the effects of several important turbine aerodynamic parameters on efficiency and numbers of airfoils. The parameters which were varied for this study were: 1) mean turbine diameter, 2) inlet annulus area, 3) airfoil chord, and 4) airflow levels.

The matrix covers diameter and annulus area variations because these variables generally have the strongest influence on turbine efficiency. Because the effect of chord generally has a weaker effect on efficiency, but affects airfoil count, its effect was examined to a more limited extent. Increased power output with increased airflow will probably be desirable at some future time. For this reason, the effect of increased airflow was also examined.

		-19 PERCENT			BASE			+15.5 PERCENT		
		INLET ANNULUS AREA								
		-20	BASE	+20%	-20%	BASE	+20%	-20%	BASE	+20%
BASELINE		X	X	X*	X	X*	X	X	X*	X
CHANGE IN CHORD	10%						X			X
	20%					X*	X			X
	30%						X			X
	40%						X			X
CHANGE IN FLOW RATE	10%			X			X			X
	20%			X		X*	X			X
	30%			X			X			X
	40%			X			X			X

* BLADE PULLS

The ground rules for this parametric study are summarized in Section 3. In brief, the ground rules for the turbine parametric study were that: 1) all turbines would be designed for 840 pounds per second of flow; 2) the rotor inlet temperature would be 2150°F and the inlet pressure would be 206 psia; 3) the coolant flow would be commensurate with a maximum metal temperature of 1450°F and would be the same for all turbines; (4) the maximum annulus area for the last stage blades would be the same as in the FT50 power turbine; (5) the

maximum rim velocity would be 1400 feet per second; (6) airfoils would have the same aspect ratio in a given row (except for chord variation studies); (7) constant mean diameter except that the ID flowpath divergence angle was limited to 14°; and (8) the work split between stages was 37/37/26 percent in all cases.

The resultant effects of changing the primary variables, diameter, and inlet annulus area on efficiency are shown on Figure 5-32.

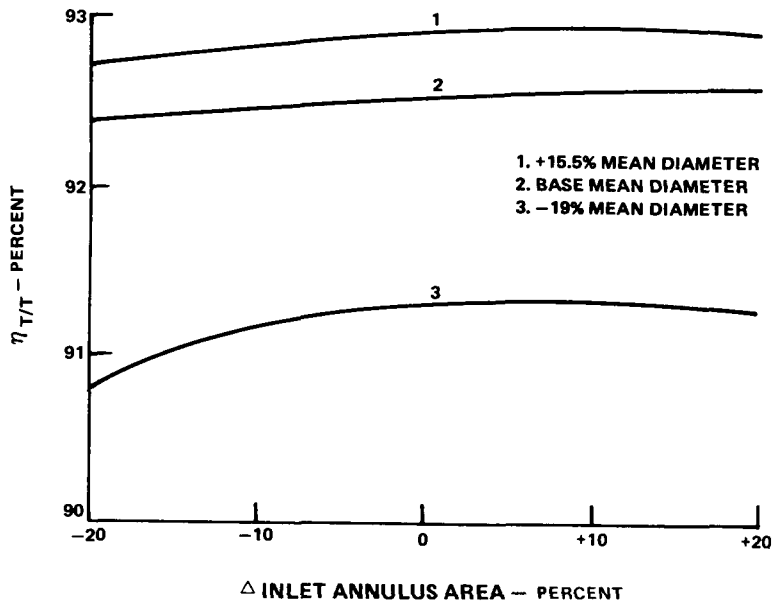


Figure 5-32 Effect of Inlet Annulus Area and Diameter on Efficiency - The effects are shown for a rotor inlet temperature of $2150^{o}F$ and a metal temperature of $1450^{o}F$.

The effect of changing mean diameter can be seen by examining the three curves. The +15.5% mean diameter curve represents the maximum diameter consistent with a 1400 feet per second rim speed. The -19% mean diameter curve represents the lowest velocity ratio at which good design experience was available.

It can be seen that increasing the diameter does not improve efficiency very rapidly. On the other hand a similar reduction in mean diameter reduces efficiency significantly. This indicates that the mean velocity ratio of the baseline turbine is near the knee of the curve where efficiency no longer improves rapidly with velocity ratio. Thus, the baseline diameter appeared to be the most suitable.

All of the turbines evaluated tend to improve in efficiency with increased annulus area. For the baseline mean diameter, however, this improvement is not

very rapid. Since there is an improvement in efficiency, and, as will be seen later, the total number of airfoils is reduced, the larger annulus area was selected.

The effect of airfoil chord is shown in Figure 5-33 for two different mean diameters. It can be seen that there is an almost negligible effect on turbine efficiency. Airfoil chord can then be varied purely from structural and airfoil count considerations.

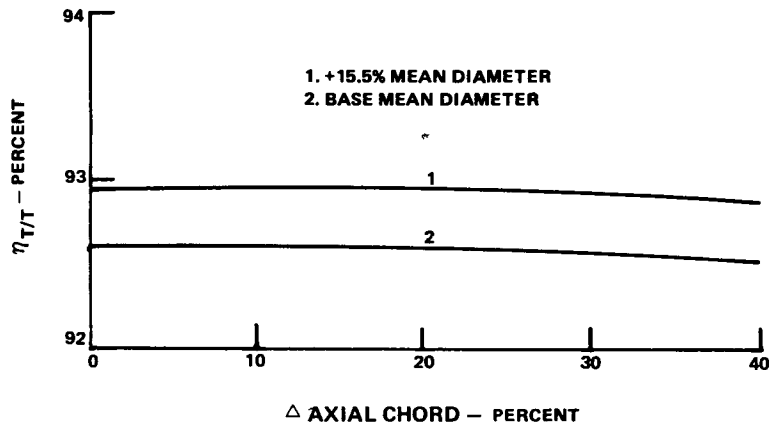


Figure 5-33 Effect of Axial Chord on Turbine Efficiency - The data are shown for a rotor inlet temperature of 2150°F, a metal temperature of 1450°F, and an inlet annulus area increase of 20%.

The effects of increased airflow are shown on Figure 5-34 for three different diameters. Note that in these curves the efficiency includes the turbine exit guide vane losses whereas Figures 5-31 and 5-32 did not. It is important to include the turbine exit guide vane losses in this case because one of the strong effects of increased airflow is to increase turbine exit guide vane losses when exit annulus area is held constant. It can be seen that for the most important case, the baseline mean diameter, the flow can be increased up to around 20 percent before efficiency penalties become significant.

In addition to the three-stage parametric study, one case of a four-stage configuration was also calculated. This single case of a four-stage turbine was calculated to ensure that significant advantages of a four-stage turbine were not overlooked. Because of the limited nature of the four-stage investigation, no strong conclusions can be drawn. However, it should be noted that at the calculated design condition of the same velocity ratio as the baseline three-stage turbine (0.58), the four-stage turbine had an efficiency slightly higher than the baseline turbine and contained only a slightly greater number of airfoils.

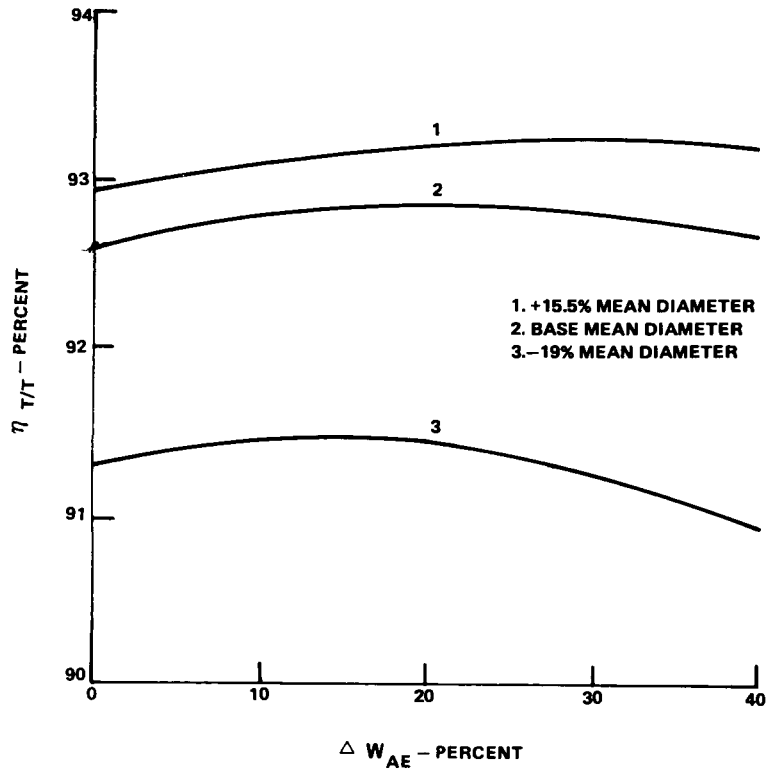


Figure 5-34 Effect of Increased Flow on Turbine Efficiency - The effects are shown for a rotor inlet temperature of 2150°F, a metal temperature of 1450°F, and an inlet annulus area increase of 20%.

Table 5-8 summarizes the results of the turbine parametric studies. It can be seen that the three stage turbines have an efficiency very close to each other, with the exception of the low efficiency of the 19 percent decreased mean diameter turbine. If the latter turbine is disregarded because of its low efficiency, then the choice of the turbine rests on the number of airfoils, which relates directly to cost, and to structural considerations and indirectly to reliability. The simplicity afforded by fewer airfoils means less chance for failure. The baseline mean diameter with 20 percent increased annulus and its low airfoil count appeared the most suitable choice. Accordingly, this design was selected for further mechanical layout evaluations. Structural considerations might later change the chord and airfoil count, but should not affect efficiency.

The flowpath for this turbine is shown in Figure 5-35.

TABLE 5-8. TURBINE PARAMETRIC STUDY RESULTS

Configuration	ηT-T²	VR	η Stage	P,T/PT	# Vanes/ Stage	# Blades/ Stage	Total Airfoils
Baseline	92.3	.585	88.9/90.4/92.3	12.09	40/76/40	76/82/66	380
19% Mean Diameter	90.9	.474	87.3/88.8/91.2	12.70	26/56/32	54/62/56	286
15.5% Mean Diameter	92.5	.676	89.1/91.0/92.7	11.93	52/76/42	88/94/76	428
Baseline 20%	92.3	.585	88.8/90.4/92.4	12.09	32/62/32	64/68/56	314

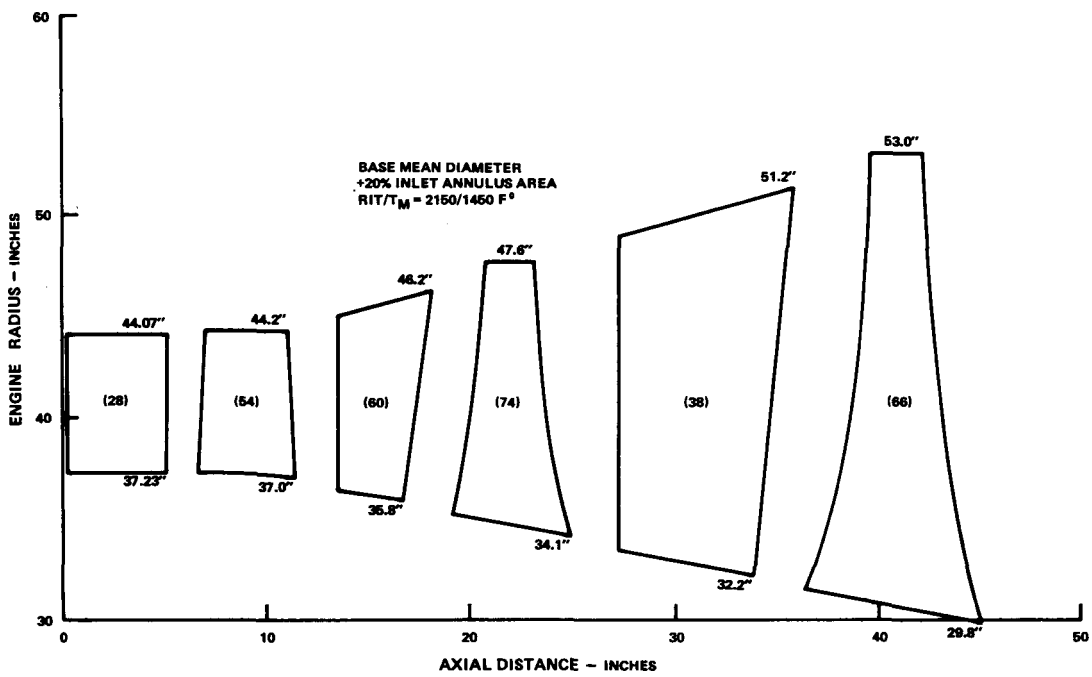


Figure 5-35 Turbine Flowpath Selected on Basis of Trade Studies - The selected configuration has the baseline mean diameter and a 20 percent increased annulus area.

EVALUATION OF GAS TURBINE CANDIDATE CONCEPTS

Four gas turbine engine concepts were originally selected on the basis of the trade aerodynamic studies for more detailed evaluation. Based on the evaluation

of the original four candidates, a final candidate was developed. The performance and descriptive characteristics of the selected configurations and the SBBL are summarized in Table 5-9. Each candidate is separately discussed in the following paragraphs.

TABLE 5-9. HRGT CONSTRUCTION COMPARISON

COMPONENT	SSBL	CANDIDATE #1	CANDIDATE #2	CANDIDATE #2A	CANDIDATE #3	CANDIDATE #4	CANDIDATE #5
Number of Main Bearings	3	2	2	4	4	2	2
Number of Compressor Stages	10	9	11	11	9	15	9
Number of Compressor Airfoils	1335	1117	1348	1348	827	1614	1117
Compressor Rotor Construction	Drum	Stacked	Drum	Drum	Drum	Drum	Drum
Compressor Case Construction	Split	Stacked	Split	Split	Split	Split	Split
Combustor Configuration	Can (8)	Can (8) Annular	Split	Annular	Can (8)	Can (8)	Can (12)
Transition Duct Configuration	Split	Split	Split	Split	Split	Split	Split
Number of Turbine Stages	3	3	3	3	2	4	3
Number of Turbine Airfoils	290	336	336	336	216	290	336
Turbine Rotor Construction	Stacked	Stacked	Stacked	Stacked	Stacked	Stacked	Stacked
Turbine Case Construction	One Piece	One Piece	One Piece	One Piece	One Piece	Split	Split
Number of Free Turbine (F/T)Stages	NA			NA	2	NA	NA
Number of Free Turbine (F/T) Airfoils	NA			NA	149	NA	NA
Free Turbine (F/T) Rotor Construction	NA			NA	Stacked	NA	NA
Free Turbine (F/T) Case Construction	NA			NA	One Piece	NA	NA

Assessment of Mechanical Features

Candidate Number 1. Candidate Number 1 is a single spool machine with a constant outer diameter, nine-stage compressor, and a three-stage turbine. The design is shown in Figure 5-36 and consists of five modules: the inlet case module, the compressor module, the diffuser/combustor module (includes the turbine case), the turbine module, the turbine exhaust module

From a maintainability viewpoint, the design features a translating burner case which will facilitate in-mount hot section maintenance by providing access to the transition duct, nozzle guide vanes and first-stage turbine blades.

However, there are a number of features that detract from maintainability. First, the diffuser/combustor module includes the turbine case which will make the module very awkward to handle during replacement. In addition, this scheme has a one-piece center shaft which will present handling problems during module replacement. The shaft should be designed in two pieces to facilitate maintenance. Further, the stacked compressor rotor and case assembly prevents in-mount maintenance. The compressor assembly would have to be removed and returned to a shop area if repairs are necessary.

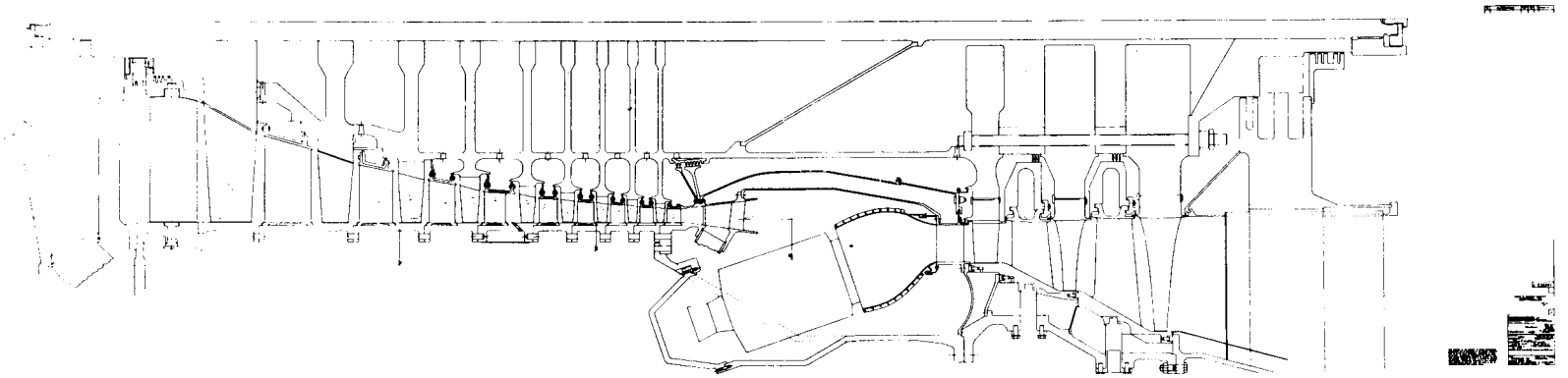


Figure 5-36 Design Layout for Candidate Number 1 - Candidate 1 is a single-spool machine with a constant outer diameter, nine-stage compressor, and a three-stage turbine.

Mechanical considerations include rotor length, and rotor balancing, and tie bolt construction. The design provides a short rotor system because of the relatively small number of stages. This is an important influence on shaft critical speed, and has been carried through to the final engine. The stacked rotor, however, will make rotor balancing difficult. Stacked rotors are prone to shifting between adjacent elements.

Candidate Number 2. Candidate Number 2 is a single-spool machine with a constant mean diameter, an eleven-stage compressor, and a three-stage turbine.

The design is shown in Figure 5-37 and consists of five modules, the inlet case module, compressor module, diffuser/combustor module (includes the turbine case), turbine module, and turbine exhaust case module

Similar to Candidate Number 1, Candidate Number 2 features a combustor with split cases. The translating burner case permits in-mount hot section maintenance by providing access to the transition duct, nozzle guide vanes and first-stage turbine blades. However, the access to the inside fasteners on the split annular combustor and transition duct is very poor. If this combustor scheme is selected for development, improved access to these fasteners should be a prime design consideration, as this will probably be a high maintenance area. Also, the diffuser/combustor modules includes the turbine case, resulting in a module that will be difficult to handle during replacement.

As a variation to investigate alternative maintenance concepts for the hot section, Candidate 2A was considered. In this case, the mechanical integrity of the annular section was maintained by eliminating the horizontal split. However, no feasibile way was found for ease of maintenance.

While Candidate Number 2 has some interesting features, it also was found to have several mechanical drawbacks. The eleven-stage compressor is more expensive and has a decreased reliability due to the larger number of parts. The in-line combustion chambers, while minimizing engine diameter, limit the development flexibility. The annular burner case is considered to be a less desirable feature for field maintenance in that its removal undermines the structural integrity of the static case system. To compensate requires a more massive mount system.

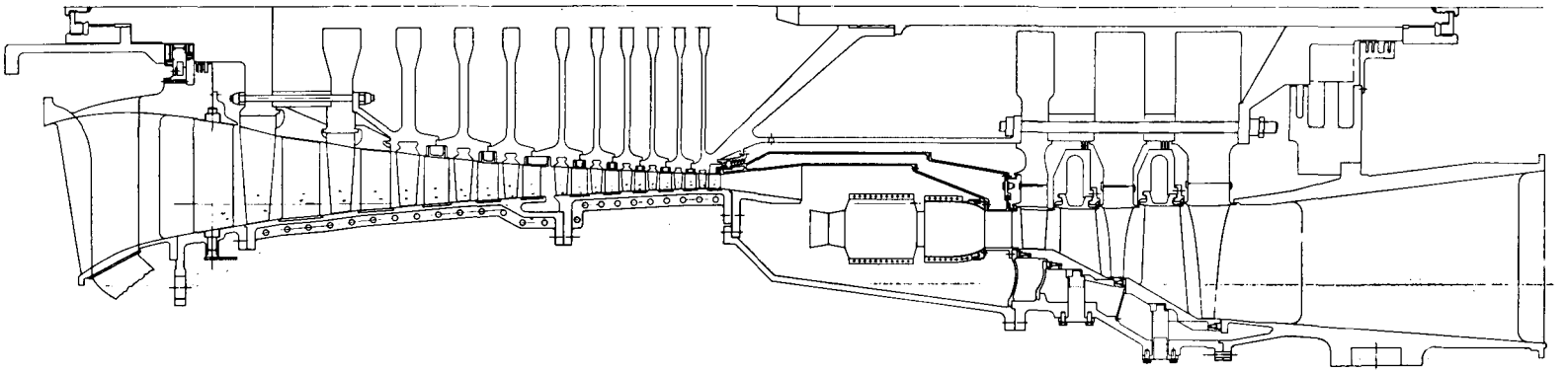


Figure 5-37 Design Layout for Candidate Number 2 - Candidate 2 is a single-spool machine with a constant mean diameter, eleven-stage compressor, and a three-stage turbine.

Candidate Number 3. Candidate Number 3 features a single-spool gas generator with a free turbine. It has a constant mean diameter, a nine-stage compressor, and a two-stage gas generator turbine, followed by a two-stage power turbine.

The design is shown in Figure 5-38 and consists of five modules: inlet case module, the compressor module, the diffuser/combustor module, the combustion turbine module, and the free turbine module.

Desirable maintenance features include split cases, which facilitate in-mount maintenance and a translating burner case which provides access to the hot section for in-mount maintenance. Detracting from maintainability are the facts that the combustor turbine module is replaced in three places which leaves the component susceptible to maintenance damage, and the 8 burner can covers will be difficult to handle during burner can replacement. In addition, the split outer duct is rather large and will be very difficult to remove and reinstall. The free turbine module adds another unit to the system that must be maintained.

Overall, Candidate Number 3 is more complex than the other engines and therefore more expensive to produce and maintain. From a mechanical standpoint, its main advantage is the flexibility to be able to furnish various turbine options. Therefore, conversion to reverse free turbine rotation and from 60 to 50 Hertz becomes feasible. However, it was not felt that this feature outweighed the other disadvantages.

Candidate Number 4. Candidate Number 4 is a single-spool machine with a constant mean diameter, a fifteen-stage compressor, and a four-stage turbine.

The design is shown in Figure 5-39 and consists of four modules: the inlet/front compressor case module, the compressor module (includes diffuser), the turbine module, and the turbine exhaust case module.

A maintenance advantage is that all compressor, diffuser, and turbine cases are split along the horizontal plane to provide in-mount access for repairs. However, the inlet/front compressor case module will be difficult to replace due to the split case flange arrangement.

Similary to Candidate Number 3, Candidate Number 4 has the cost disadvantage of a higher number of parts along with decreased reliability when compared with candidate numbers 1 and 2.

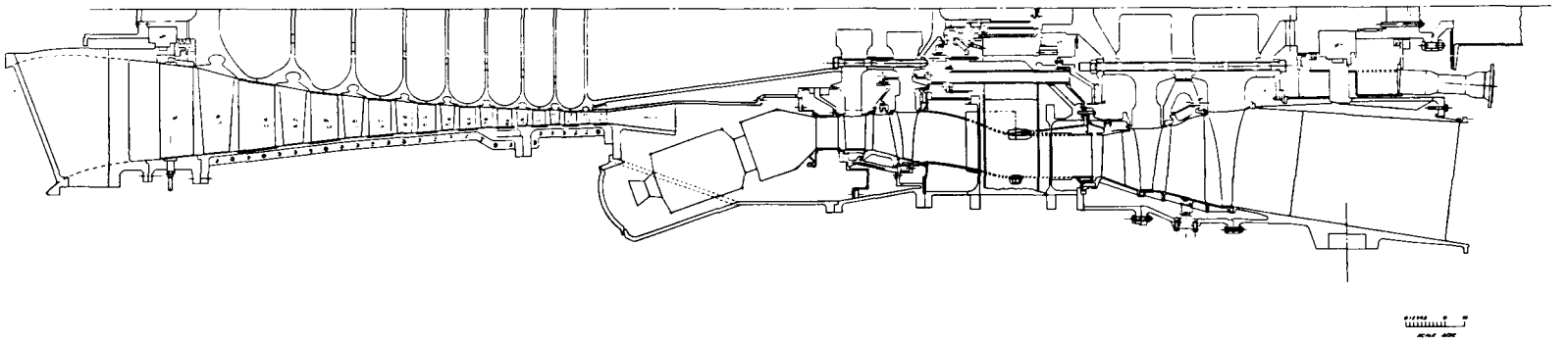


Figure 5-38 Design Layout for Candidate Number 3 - Candidate 3 is a free turbine machine with a constant mean diameter, nine-stage compressor, and a two-stage gas generator, and a two-stage power turbine.

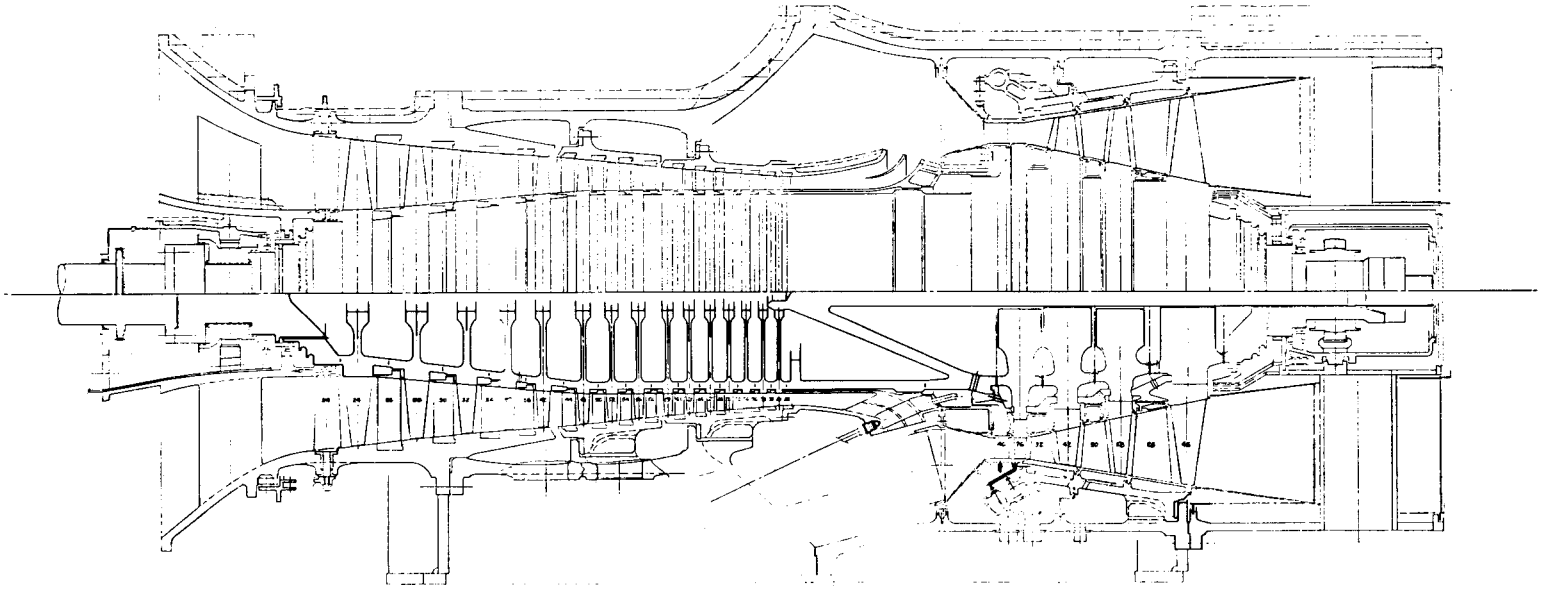


Figure 5-39 Design Layout for Candidate Number 4 - Candidate 4 is a single-spool machine with a constant mean diameter, a fifteen-stage compressor, and a four-stage turbine.

Candidate Number 5. Candidate Number 5 is a single-spool machine with a constant outer diameter, a nine-stage compressor, and a three-stage turbine.

The machine consists of four modules: the inlet case module, the compressor module (includes diffuser), the turbine module, and the turbine exhaust case module.

The combustor configuration is based on Candidate Number 5. However, more refined analysis increased the number of cans from 8 to 12. A maintenance advantage of this design is the use of split cases in the compressor, diffuser, and turbine which will allow in-mount maintenance. However, the twelve burner can covers will be difficult to handle during burner can replacement, and the split outer duct is rather large and will be very difficult to remove and reinstall.

Overall, Candidate Number 5 combines the good features of the SSBL with the best features of the other four candidate configurations. It offers the following features:

- Minimal number of aerodynamic parts
- Short rotor length with favorable diameter to length ratio
- Two bearing compartments
- Ease of horizontal maintenance
- Flexibility for burner development

Candidate number 5, which was developed as a result of initial evaluations of the original 4, was selected to be the final HRGT conceptual design.

Maintainability Evaluation

The six candidate conceptual designs were assessed for maintainability by comparing features and estimating the maintenance task times required to replace the gas turbine assembly, modules, and some major components.

The results of these studies are presented in Table 5-10. The SSBL is included in the table for reference. The mean corrective time (MCT) and mean protective time (MPT) for each component were based on vendor supplied information or experience available within United Technologies. The estimated task times include the time to gain access, remove and replace the item, and check it out. These data were used as input for the overall selection process.

Reliability Assessment

The single shaft baseload (SSBL) design was used as a comparative reference for the reliability analysis. As discussed previously, the SSBL has a ten-stage drum compressor with an axially split casing for the compressor module. The turbine has three stages assembled as a unit with tie bolts through the disks. The SSBL has a three bearing arrangement for rotor support which reduces the span lengths and the possibility of critical speed problems. The combustion arrangement consists of eight individually canted burner cans readily accessible from an axially split annular burner-diffuser.

Early in the Phase I study, a thorough reliability analysis was completed for the SSBL, using the Reliability Prediction Model method. The candidate concepts were evaluated from a reliability standpoint in terms of differences from the SSBL base.

The reliability analysis results for the basic configurations summarized in Table 5-9 are presented in Figure 5-40.

As shown, Candidate Number 1 and Candidate Number 2 were better than the baseline SSBL, while Candidate Number 3 and Candidate Number 4 were worse. In general, this can be attributed to overall mechanical complexity. Candidate Number 3 and Candidate Number 4 were more complex by virtue of extra shafts and bearings and added compressor and turbine stages, respectively.

Based on these results, a final configuration (Candidate Number 5) was developed to embody the best features from Candidate Number 1 and Candidate Number 2. Candidate Number 5 offers the following features when compared with the original SSBL reference design:

- The number of bearings have been reduced to two which provides mechanical simplicity of the bearing compartment and the engine lubrication system as well.

TABLE 5-10. HRGT MAINTENANCE TASK TIME ESTIMATE COMPARISON

COMPONENT	SSBL MMH	SSBL HRS	CANDIDATE #1 MMH	CANDIDATE #1 HRS	CANDIDATE #2 MMH	CANDIDATE #2 HRS	CANDIDATE #2A MMH	CANDIDATE #2A HRS	CANDIDATE #3 MMH	CANDIDATE #3 HRS	CANDIDATE #4 MMH	CANDIDATE #4 HRS	CANDIDATE #5 MMR	CANDIDATE #5 HRS
Gas Turbine	248	90	248	90	248	90	242	89	273	101	273	101	273	100
Inlet Module	62	45	62	45	62	45	72	48	62	45	138	75	85	55
No. 1 Bearing	35	27	35	27	35	27	35	27	35	27	141	77	88	57
Comp. Module	143	71	143	71	143	71	116	58	203	101	328	175	328	175
R6,R7,R8 Blades	159	84	236	125	156	84	156	84	156	84	152	86	248	128
Diffuser	162	78	172	83	172	83	88	50	203	101	209	112	187	101
No. 2 Bearing	146	74	57	34	57	34	90	52	169	84	65	40	65	40
Burner Cans	213	112	184	96	184	96	114	58	79	53	79	53	100	65
Trans. Duct	205	108	175	93	196	104	99	54	125	66	129	72	147	82
1st Turb. NGV	142	72	190	97	226	115	87	51	145	76	149	82	162	89
Fuel Nozzles	64	42												
Fuel Mfds.	64	42											--	--
Turb. Module	122	63	130	68	130	68	118	61	193	96	197	101	197	101
TEC Module	96	54	100	56	100	56	NA	NA	100	56	100	56	--	--
No. 3 Bearing	52	32	NA				86	49	154	69	NA		2A	3
No. 4 Bearing	NA						124	65	72	41	NA		3	2A
Comb. Module	NA						80	45	NA				--	--
F/T Module	NA								232	97	NA		--	--
F/T Tec	NA								197	88	NA		--	--
F/T Int. Case	NA								148	66	NA		--	--

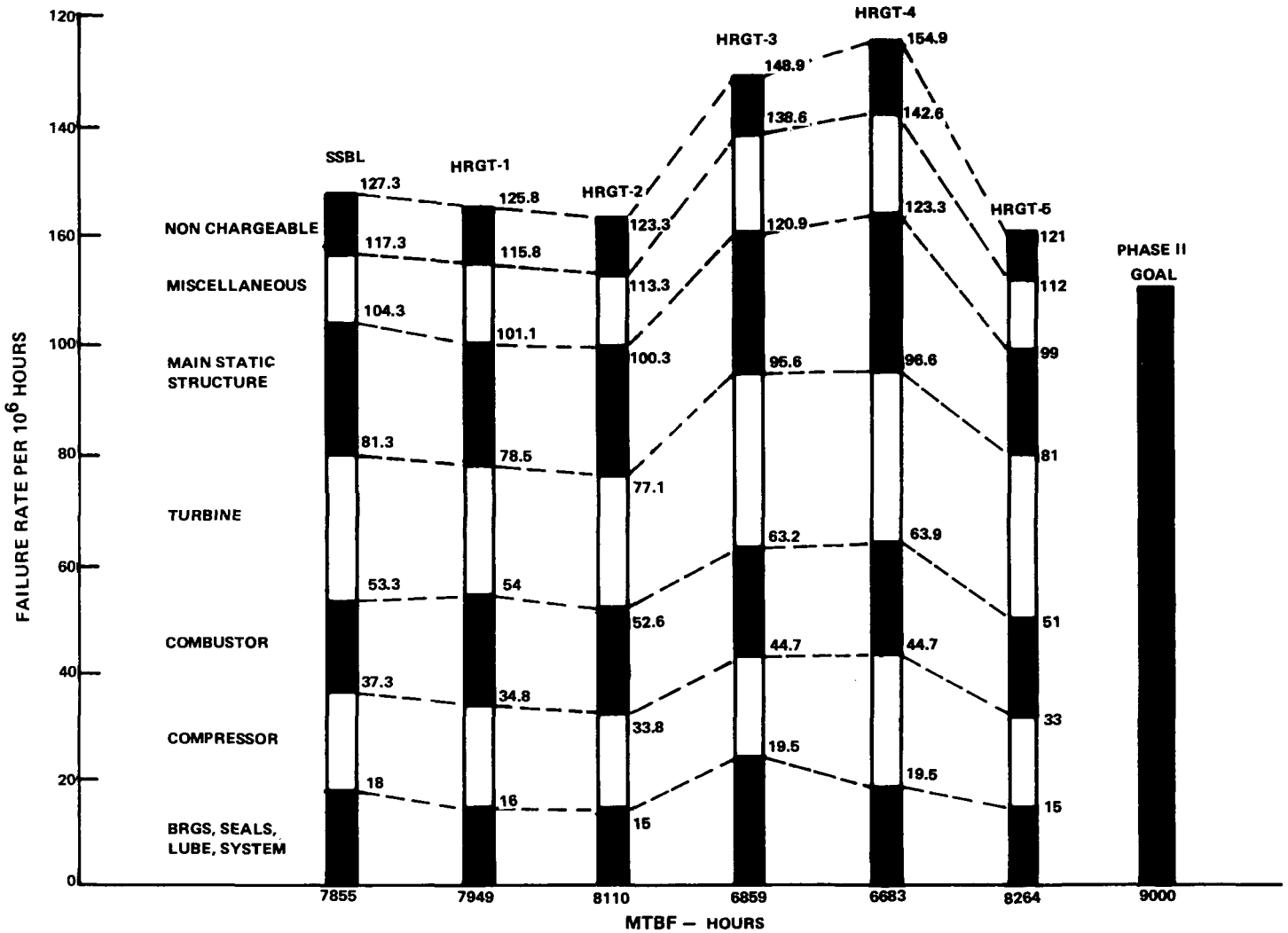


Figure 5-40 Results of Reliability Analysis for Five Candidate Conceptual Designs - Candidate Number 5 was designed to incorporate the best reliability features of the other candidates.

- Curvic couplings in compressor and turbine disks have been eliminated which greatly reduces the potential wear problems.

- Using virtually constant outside compressor blade diameters has assured a low aspect ratio, i.e., the ratio of blade height to average chord length. This results in thick airfoils of such rigidity to be out of the normal engine excitation range, blades not as sensitive to erosion and blades which normally will not require shrouds to avoid induce flutter. Since the number of airfoils per-stage is considerably reduced in a low aspect ratio blade design and the number of stages is minimized to nine, the reliability has been enhanced.

- The number of combustor cans was increased from 8 to 12 to meet diffusion requirements, which slightly increases the potential failure rate.

- The SSBL turbine casing is annular. A partially split design is used in Candidate Number 5 which causes an increase in the failure rate from 28 to $30/10^6$ hr.

- The turbine case split forward of the 1st-stage blades. This will provide in-mount access at a small sacrifice, in reliability.

- In the static structure there is no midspan support required in Candidate Number 5. This combined with the double wall construction of the casings causes the failure rate to drop from 23 for the SSBL to $18/10^6$ hr.

- Double wall construction in the compressor module improves the heat transfer characteristics and retains the structural stiffness.

- The quick quench combustor design has flexibility as far as fuels and NOx. In addition, the predicted failure rate is $18/10^6$ hour compared to the SSBL's $16/10^6$ hour predicted rate.

- The compressor uses the drum rotor concept. Having fewer stages in total than the SSBL results in the failure rate of $15/10^6$ hr compared to $18/10^6$ hr for the SSBL.

- A less complex bearing system with only two bearings reduces the failure rate for all bearings, seals and fabrication system from 18 to $15/10^6$ hr.

CONCLUSIONS

Aerodynamics and Cycle

Turbine inlet temperature was selected, concomitant with the limiting airfoil metal temperature and cooling scheme, through comprehensive trade-off studies. Trades took into account the cost of electricity and reliability. This was also true for a separate cycle pressure ratio study.

Preliminary selection of the turbine inlet temperature and cooling scheme, along with pressure ratio, enabled initiation of aerodynamic design.

Efficient minimum stage compressor and turbine design concepts were evolved by taking full advantage of current aerodynamic design technology. Aerodynamic parameters covered a wide range of variables which was facilitated by using United Technologies' computer design systems.

The compressor and turbine studies, along with studies of combustor schemes, led to the selection of candidate configurations for further mechanical layout and reliability evaluation.

Mechanical Design and Reliability

Several basic features in the original Single Shaft Base Load reference design turned out to be attractive from a reliability standpoint. This included the single-shaft, front-end drive arrangement and the modular maintenance concept. These features afforded major improvements in projected reliability over the current FT4. Mechanical design studies of candidate HRGT concepts chiefly served to refine and verify the basic Single Shaft Base Load approach. The final selected design is predicted to have a failure rate only slightly better than that projected for the Single Shaft Base Load.

However, mechanical design evaluations have identified several important aspects of the conceptual configuration arrangement. These appear as design features

that are readily apparent on inspection of the layout drawing of the final candidate No. 5, which was selected to serve as the HRGT design. The features include:

- The incorporation of two bearings rather than three as originally determined for the Single Shaft Base Load. By using the refined compressor and turbine aerodynamics, the rotor shaft length and weight were reduced to the point where simple front and rear bearing supports were feasible, as confirmed by preliminary dynamic analysis.

- Improved case construction. Double-wall cases enhance reliability. The case arrangement can also enhance ease of maintenance for a modular engine.

- Means for breaking apart the (single-shaft) rotor for fully modular maintenance.

- Refinement of hot section maintenance steps.

- Incorporation of new rich burn quick quench combustor.

- Definition of seals and bearings conceptual design.

The configuration that finally evolved can serve as a well-established basis for further mechanical and aerodynamic design refinement as the program proceeds into the follow-on program preliminary design effort to be conducted in Phase II.

SECTION 6.0

COMBINED CYCLE POWERPLANT RELIABILITY ANALYSIS

INTRODUCTION

Reliability analysis covering the broad perspective of the complete combined cycle plant was conducted throughout the course of the Phase I conceptual design studies.

Initially, a reference plant was formulated to represent the current status of a plant designed around an appropriate, existing gas turbine design. The Single Shaft Base Load design concept already available was best suited for the purpose. The sources for the Single Shaft Base Load engine and balance-of-plant reliability background are presented in Section 4. As indicated, the balance-of-plant equipment was predominantly based on the FT4 KECO installation.

Using the reference plant, overall availability was calculated and a breakdown according to major component contributions was determined. These results are covered below, along with additional analysis concerning variations in maintenance practice. In the latter case, valuable insight was provided by using the "Monte Carlo" statistical simulation technique.

Based on the initial Single Shaft Base Load results and further analysis, the plant components were altered to improve overall plant availability. These improvements have been reflected in the final reliability values and their component contributions as calculated for the HRGT-CC. At this stage, reliability projections for the final HRGT gas turbine design were substituted for the values representing the original reference Single Shaft Base Load.

The final calculated reliability results for the HRGT-CC were compared with program goals. Our recommendations concerning the ultimate achievement of these goals are presented at the end of this section.

INITIAL ANALYSIS OF REFERENCE (SSBL) COMBINED CYCLE PLANT

Development Block Diagram Model for Major Components

Initially, failure mode and effect evaluations were conducted for each of the subsystems supporting the SSBL gas turbine and the steam plant. These failure mode and effect evaluations resulted in the elimination of certain auxiliary subsystems not affecting normal running reliability. A component failure mode and effect evaluation was then conducted on each component within the critical systems to determine whether failure of that component would cause plant shutdown during normal operation. On the basis of these failure mode and effect evaluations, a power plant system reliability block diagram was prepared as shown in Figure 6-1. Blocks 1 through 5 are gas turbine related while blocks 6 through 12 are steam turbine related. Block 13 represents the plant output bus.

Failure rates, repair times and scheduled outage requirements for major equipment in the plant with SSBL gas turbines were based on information from major equipment vendors as described in Section 4. Failure rates and repair times for minor components or components where vendor reliability statistical data was not available were based on nuclear plant reliability data system failure rates.

Availability Analysis

Outage time for repair was based on consultation with experienced engineers on the FT50 and the KECO steam plant design including judgements on allowances for plant shutdown, cool down of the particular component, parts lead time, actual repair time, and time to restart the entire power plant. Included in the evaluation and reliability estimates were planned outages for maintenance. The critical item dictating planned outage hours in the steam plant was the steam turbine generator. In the availability analysis, it was assumed that planned maintenance on all of the equipment in the steam plant can be accomplished during the yearly 600 hour planned maintenance on the steam turbine/generator. No additional planned outage time (POH) was included for other equipment in the steam plant. Planned plant outages for inspection, maintenance, and water wash are summarized in Table 6-1.

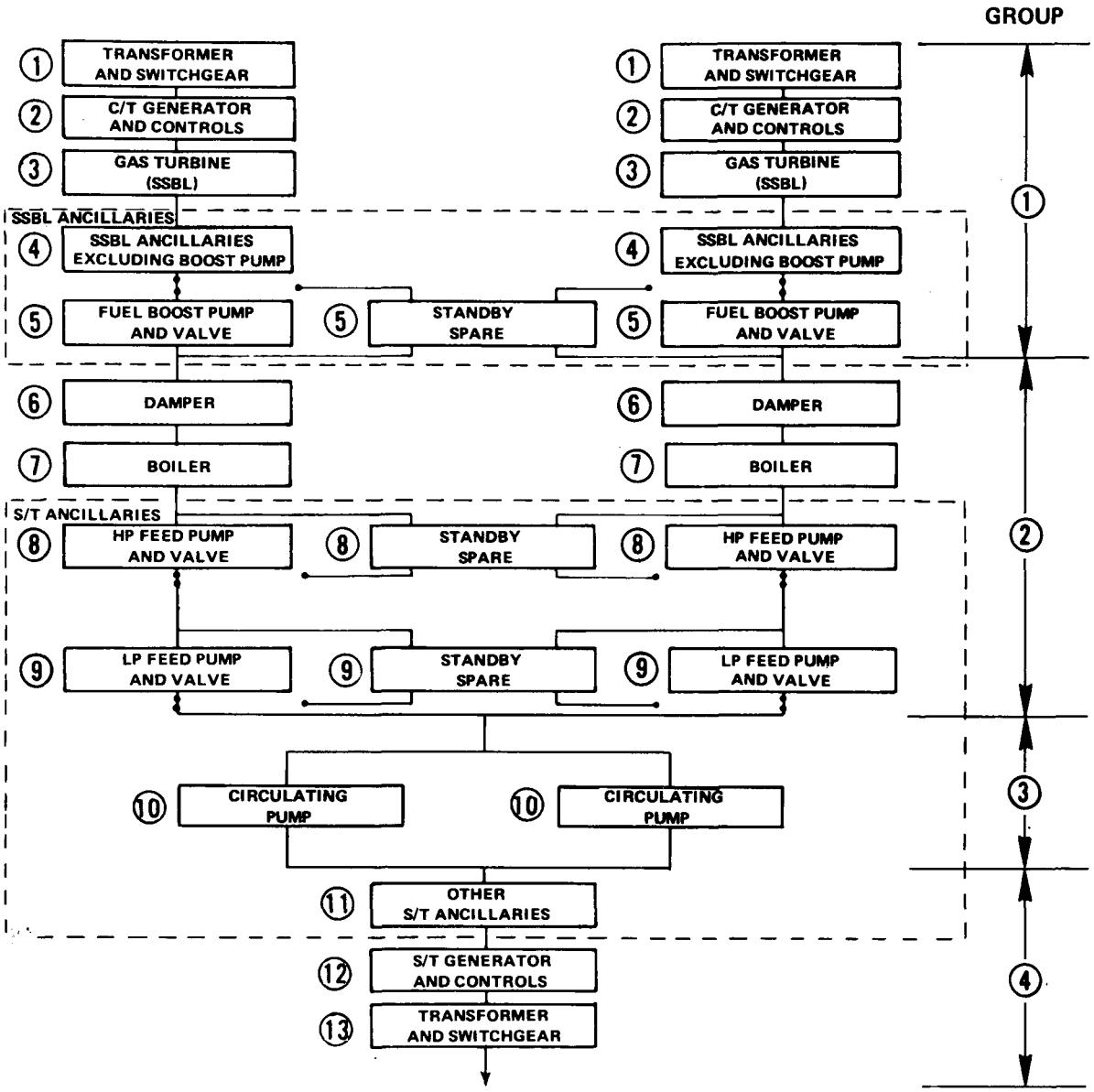


Figure 6-1 SSBL Power Plant System Reliability Block Diagram - A block diagram was used in performing the failure mode and effect analysis.

TABLE 6-1. PLANT SCHEDULED OUTAGES FOR MAINTENANCE

	Basis	Interval	Outage Time
STEAM TURBINE/ GENERATOR	Toshiba International	Yearly	600 Hours
GAS TURBINE			
Water Wash	FT50	600 Hours	13 Hours
Inspection and Maintenance			
Routine Inspection		Bimonthly	12 Hours
Hot Section		Yearly	51 Hours
Major Field		2 Years	69 Hours
	Gas Turbine Total		400 Hours

The calculations to determine equivalent availability and equivalent forced outage rates for the plant employed standard mathematical models as described in Appendix B.

The mean time between failure (MTBF), mean corrective time (MCT), and planned outage hours (POH) for each of the blocks in the reliability block diagram are presented in Table 6-2.

Also included is the calculated number of component failures per year of operation and both the unscheduled and scheduled availability for each of the blocks on the reliability block diagram. Evaluation of each of the components resulted in a breakdown of the power level that could be achieved from the power plant as a function of the section of the power plant subjected to a component failure.

The power levels of the system and the related equipment status which is the primary cause of the condition are shown in Table 6-3. Many fault conditions can exist in the power plant, but the end result is to produce the six reduced power levels shown in the table. This is discussed in detail in Appendix B.

TABLE 6-2. POWER PLANT AVAILABILITY SUMMARY ASSUMING GAS TURBINE REPAIR PARTS LEAD TIME = 0

NO.	BLOCK	MTBF(1)	MTTR(1)	SOH(1)	N(2)	AVAILABILITY UNSCHEDULED	AVAILABILITY SCHEDULED	AVAILABILITY TOTAL
1	G/T TRANSFORMER AND MISC. ELECTR.	487,000.0	2,360.0	0.0	0.0	0.995	1.0	0.995
2	G/T GENERATOR	4,900.0	100.0	0.0	1.8	0.980	1.0	.0
3	SSBL	7,855.0	58.0	400.0	1.1	0.993	0.954	0.947
4	SSBL ANCILLARY, EXCLUSING FUEL BOOST PUMP	1,350.0	32.0	0.0	7.0	0.977	1.0	0.977
5	FUEL BOOST PUMP ASSY	33,900.0	37.0	0.0	0.3	0.999	1.0	0.999
6	DAMPER	6,500.0	20.0	0.0	1.3	0.997	1.0	0.997
7	BOILER, DUCTING AND CONTROLS	3,020.0	19.0	0.0	2.9	0.994	1.0	0.994
8	FEEDWATER PUMP ASSEMBLY-HIGH	14,000.0	222.0	0.0	1.0	0.984	1.0	0.984
9	FEEDWATER PUMP ASSEMBLY-LOW	25,400.0	134.0	0.0	0.3	0.995	1.0	0.995
10	S/T ANCILLARY, EXCLUDING FEED PUMPS	3,480.0	114.0	0.0	7.1	0.979	1.0	0.979
11	STEAM TURBINE/GENERATOR SYSTEM	8,000.0	24.0	600.0	1.1	0.997	0.932	0.929
12	S/T TRANSFORMER AND MISC. ELECTR.	576,000.0	2,800.0	0.0	0.0	0.995	1.0	0.995
13	OUTPUT BUS	0.330E+10	1,090.0	0.0	0.0	1.0	1.0	1.0

(1) HOURS
(2) NUMBER OF COMPONENT FAILURES PER 8760 HOURS OF OPERATION

TABLE 6-3. POWER LEVEL CAPABILITY

POWER LEVEL-MW	EQUIPMENT STATUS
282	ALL EQUIPMENT FUNCTIONING
241	ONE MAIN CIRC PUMP OUT, HALF COOLING TO CONDENSER
238	ONE STEAM GENERATOR OUT, HALF STEAM FLOW TO STEAM TURBINE
200	STEAM TURBINE/GENERATOR OUT
138	ONE GAS TURBINE/GENERATOR OUT
100	ONE GAS TURBINE/GENERATOR PLUS STEAM TURBINE/GENERATOR OUT
0	TWO GAS TURBINE/GENERATORS OUT

From this data, total power plant availability and forced outage rates were calculated using an existing reliability computer program which was modified to describe the SSBL plant. The resulting power profile, individual probabilities at each power level, and cumulative probabilities are shown in Table 6-4. Average power was calculated to be 286 megawatts; the equivalent availability was found to be 87.4 percent. The calculated equivalent forced outage rate is 6.6 percent.

Overall combined cycle plant availability was analyzed to determine the contribution of the individual component elements of the power plant to unavailability and the results are shown in Figure 6-2. The SSBL gas turbines cause 46 percent of the overall plant unavailability. The gas turbine is the largest contributor due primarily to the 400 hours scheduled down time per year, as well as the relatively large block of power (138 MW) that is lost when one gas turbine shuts down. The gas turbine generator and the SSBL ancillary equipment contribute another 27 percent and 19 percent, respectively.

Component failure contribution to unscheduled partial shutdowns is shown in Figure 6-3. The SSBL ancillary equipment is the primary cause of partial plant shutdowns. Its effect on availability is not as great as for the gas turbine, however, because the mean time to repair is quite short.

TABLE 6-4. POWER PROBABILITY AND AVAILABILITY - TOTAL POWER PLANT

Available Power Megawatts	Probability Percent	Cumulative Probability Percent
282	71.5	71.5
241	01.5	73.0
238	01.5	74.4
200	07.0	81.4
138	15.9	97.4
100	01.7	99.0
0	01.0	100.0

Average Power = (282 x 0.715) + (241 x 0.015) + Ect. = 246 MW
Availability = 246/282 = 87.4%

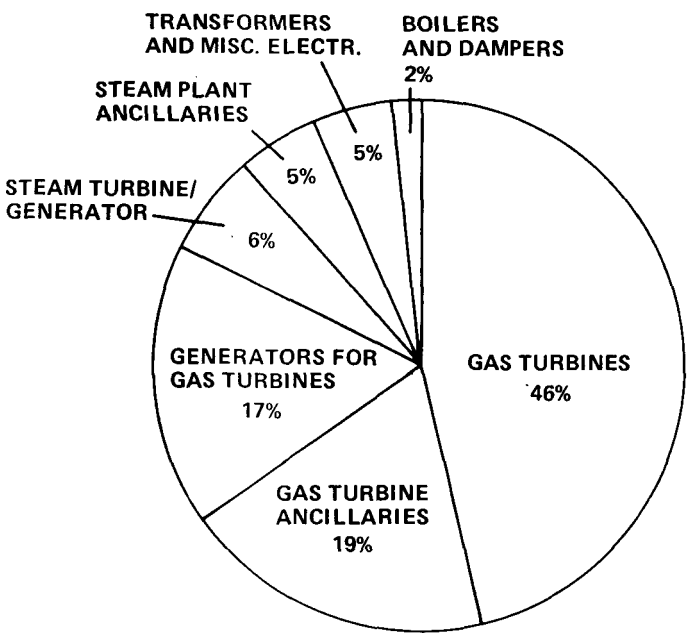


Figure 6-2 Contributions to Unavailability - The gas turbine is the largest contributor, followed by the generators and ancillaries for the gas turbine.

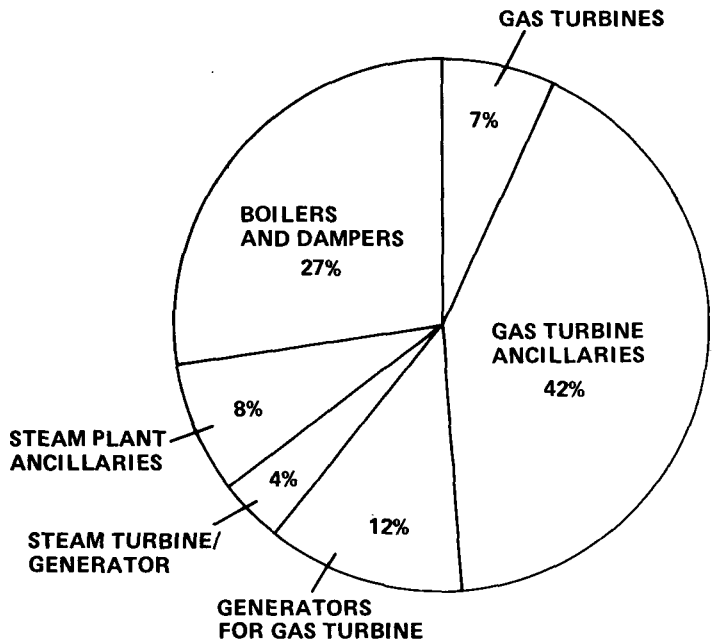


Figure 6-3 Component Contributors to Unscheduled Partial Shutdowns - Gas turbine ancillaries, boilers, and dampers are the major contributors.

Monte Carlo Analysis

As indicated above, the reliability studies of the SSBL gas turbine combined cycle plant yielded a predicted equivalent availability of 87.4 percent. In order to establish an availability distribution and examine the sensitivity of component maintenance philosophy on availability, a Monte Carlo simulation model was developed.

The Monte Carlo simulation model incorporates all of the logic of the reliability model previously described plus the following additions. Maintenance policy decision points are included in the logic to allow scheduled maintenance to be accomplished earlier when the equipment is down for unscheduled maintenance. For the gas turbine, the decision point also applies for late action when the other gas turbine is already down for any reason.

The decision points used for early or late scheduled maintenance actions are as follows:

Gas turbine - 15 percent of scheduled interval = 2 days

Steam turbine - 5 percent of scheduled interval = 1 week.

Gas turbine starting reliability was included in the logic with an assumed starting reliability of 94 percent, based on starting reliability of the FT4's at one installation surveyed. The assumed mean time to restart is one hour and the restart time is assumed to follow an exponential distribution. Additional capability in the simulation model includes an option to handle spares-on-hand or include a lead time for SSBL modules and other components, and an option to simulate various failure characteristics such as random failures only, wearout failures only, or random and wearout failures.

In the Monte Carlo analysis, one-hundred five-year simulations were run to produce a predicted availability distribution as shown in Figure 6-4. The mean value of combined availability is 87 percent, similar to the prediction using the simpler model. The most likely value (mode) of equivalent availability is 90 percent and the range is 76-91 percent. The skewness of the distribution is caused by the infrequent failure of long-leadtime (6-12 months) items.

Sensitivity studies were undertaken to evaluate the effects of various parameters on availability. The influence on availability of stocking spare components is shown in Table 6-5.

Reducing gas turbine cool down time from the baseline value of 10 hours to 6 hours improves availability by 1 percent, as shown in Table 6-6. A more complete discussion of the Monte Carlo analysis and its background is presented in Appendix C.

Effect of Parts Lead Time on Actual Size Availability

The parts lead time for repair of the gas turbine has a significant impact on equivalent availability and equivalent forced outage rate for the overall plant. Mean-corrective-time estimates account for only the actual on-site or module replacement time and do not account for lead time to get replacement modulus or repair parts for in-mount repair. While this is proper in evaluating "inherent reliability", it yields an overall reliability estimate which is more optimistic than actually experienced by utilities which do not have a ready supply of spare parts.

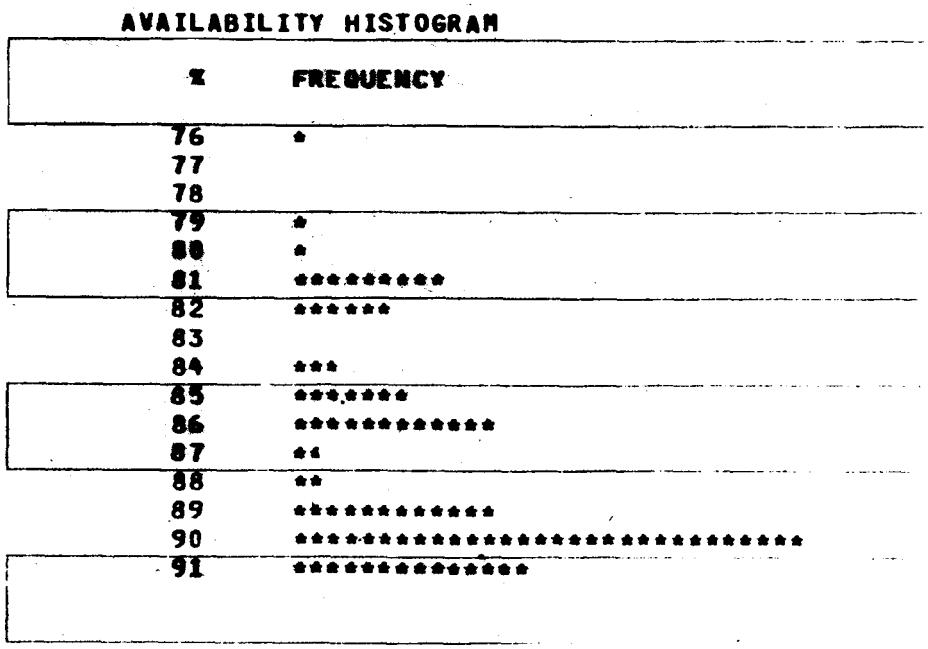


Figure 6-4 Availability Histogram for SSBL Combined Cycle Plant - Analysis of one hundred five-year simulations indicate an availability of 90 percent.

TABLE 6-5. AVAILABILITY IMPROVEMENT BY STOCKING LONG-LEAD-TIME ITEMS

Item	Lead Time	Availability Improvement With Item at Site	ΔMW
G/T Transformer	12 mos.	.37%	1.0
S/T Transformer	12 mos.	.10%	.3
G/T Main Fuel Pump	6 mos.	1.04%	2.9
G/T Generator HEX	6 mos.	.66%	1.9
Instrument Air Compressor	6 mos.	.13%	.4
ALL ABOVE		2.3%	6.5

TABLE 6-6. AVAILABILITY VS. ENGINE COOLDOWN TIME

Cooldown Time	Availability Improvement	ΔMW
10	0	0
8	0.5%	1.4
6	1.0%	2.8

The strong influence of gas turbine repair parts lead time on equivalent availability for the overall combined cycle plant using SSBL gas turbines is illustrated in Figure 6-5. With instantaneous spare part availability "inherent power plant availability" is 87.4 percent. Utilities without spare parts on hand normally can expect delays of 1 to 2 months resulting in real-life equivalent availability levels between 73 and 80 percent. These values are, of course, based on the pessimistic assumption that all of the spare parts associated with gas turbine failures would be subjected to the same repair parts lead time.

The effect of gas turbine repair parts lead time on equivalent forced outage rate for the overall combined cycle plant SSBL gas turbines is shown in Figure 6-6. Inherent forced outage rate for utilities which stock repair parts is 6.6 percent, whereas a utility that does not stock repair parts could have forced outage rates of 14-21 percent.

The influence of gas turbine repair parts lead time on the 30 year levelized power cost is presented in Figure 6-7. The change in power cost for increasing spare parts lead time is shown for three levels of utility system characteristic "m" in the EPRI equation used to calculate cost of electricity, described in Appendix A.

To provide additional perspective, the equivalent availability and equivalent forced outage rate for just a single SSBL gas turbine and its immediate ancillary equipment is presented in Figure 6-8 showing the effect of gas turbine repair parts lead time. With parts on hand, the calculated inherent

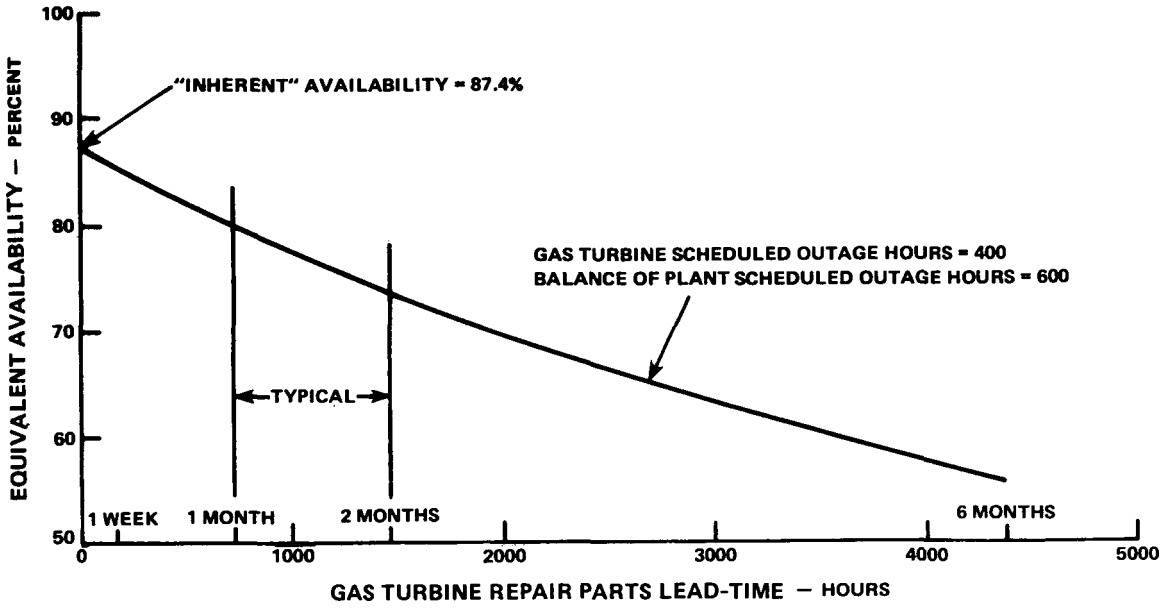


Figure 6-5 SSBL Combined Cycle Plant Availability as a Function of Gas Turbine Repair Parts Lead Time - Parts lead time has a significant effect on plant availability.

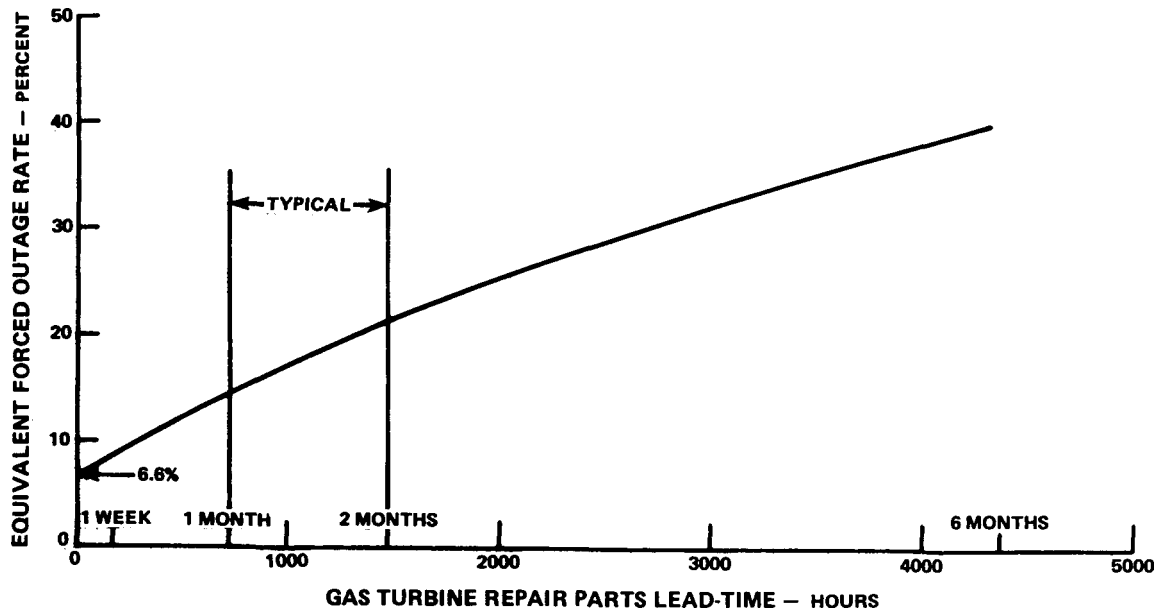


Figure 6-6 SSBL Combined Cycle Plant Equivalent Forced Outage Rate as a Function of Gas Turbine Repair Parts Lead Time

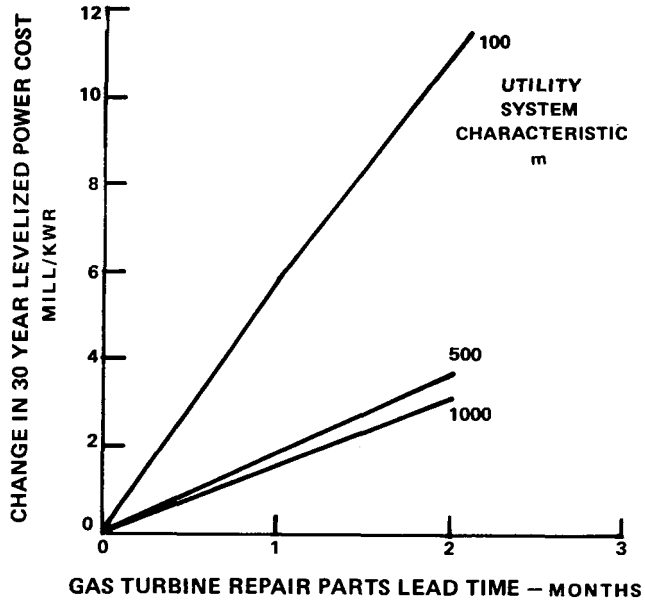


Figure 6-7 Effect of Gas Turbine Repair Parts Lead Time on 30-Year Levelized Power Cost - For low utility system characteristic values, the effect of repair part lead time on power cost can be very large.

availability of an SSBL gas turbine and just its immediate auxiliary equipment is 90.2 percent. With one to two months lead time on repair parts, availability would be expected to drop to the 76 to 83 percent range. Equivalent forced outage rate for just the gas turbine and its auxiliaries is shown in Figure 6-9.

The influence on availability of spares lead time for the gas turbine is shown in Figure 6-10 for the reference design with modular construction and for a non-modular design where an entire engine is replaced. The modular design is considerably better than the non-modular design when the spares are kept on hand, particularly when the lead time is long. Availability drops considerably with increasing lead time if no spares are kept on hand. An availability in the mid-80's can be maintained with the modular design keeping spares on hand, even with lead times as long as 6 months.

Starting Reliability Determination

One of the guidelines in the Phase I Program was a gas turbine starting reliability of 99 percent. Experience on aircraft-derivative gas turbines such as the FT4 as reported in the EEI data shows a starting reliability of 85.3

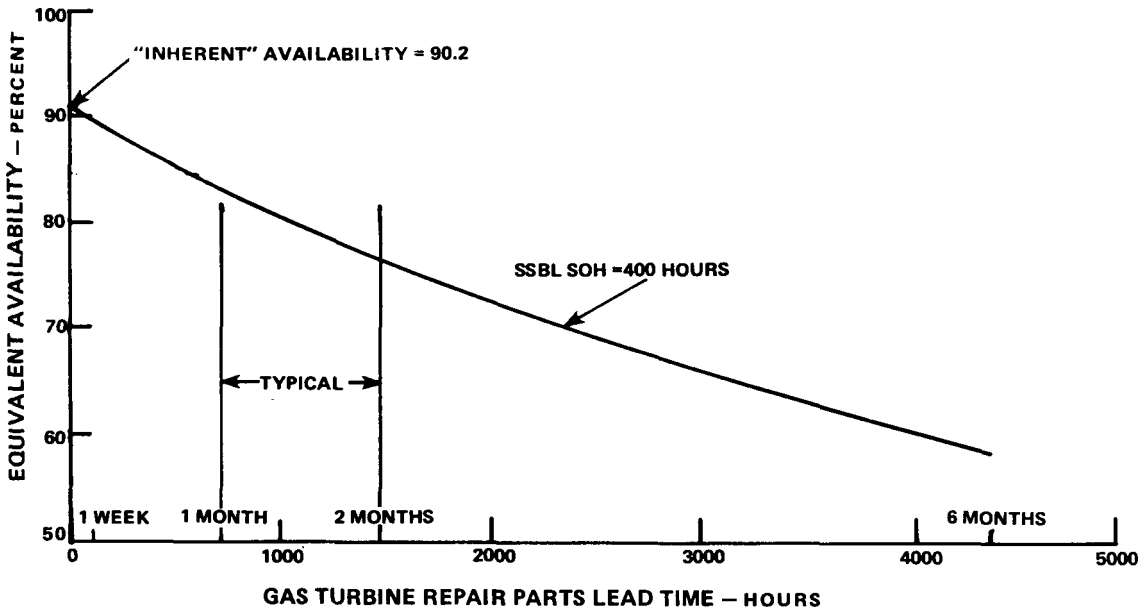


Figure 6-8 SSBL Gas Turbine and Auxiliaries Availability as a Function of Gas Turbine Repair Parts Lead Time - One to two months of lead time on repair parts drops availability from 90.2 percent to the 75 percent range.

percent. The EEI data on aircraft derivative jet engines indicates a MTBF of 799 hours. In order to estimate the starting reliability of the SSBL gas turbine system using the calculated MTBF estimates previously described, a method for estimating startup reliability using normal operating MTBF information was devised by the application of a "start stress factor" to the normal operating failure rate of the equipment. The start stress factor is established by relating starting reliability to start time and operating equipment MTBF for nuclear plants, fossil plants, gas turbines, diesel plants and jet engines from EEI data. The start stress factor is defined as follows:

Start Stress Factor = (1 - R) X (MTBF) X (TS)

where:

R = Starting Reliability = successful starts/attempted starts
TS = Start time in hours
MTBF = Operating mean time between failure

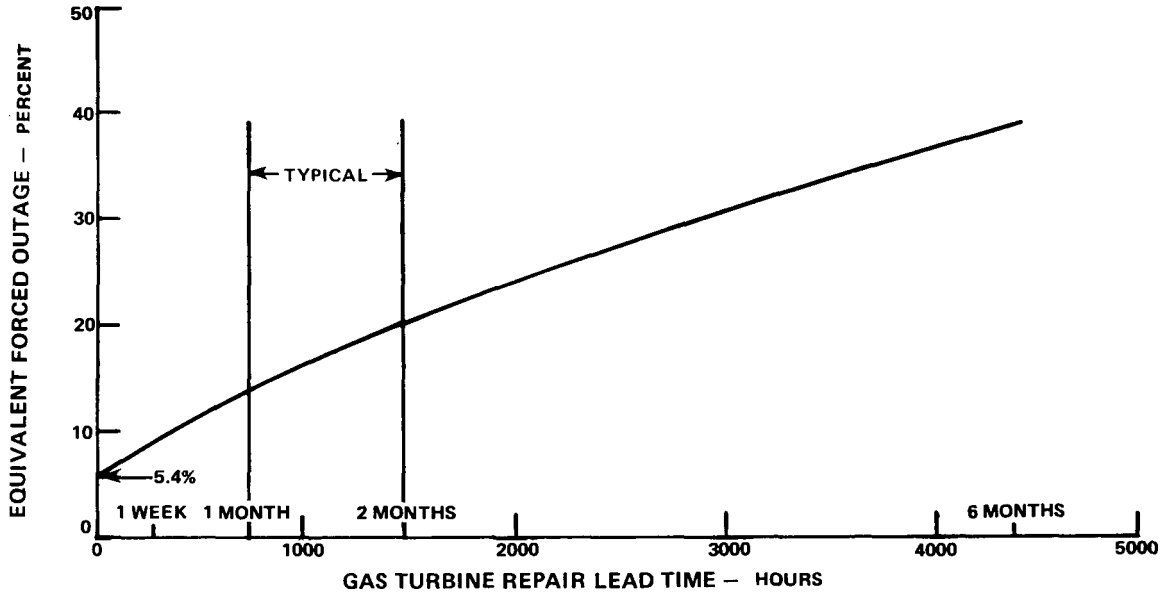


Figure 6-9 SSBL Gas Turbine and Auxiliaries Equivalent Forced Outage Rate as a Function of Gas Turbine Repair Parts Lead Time - A parts lead time of two months can raise the equivalent forced outage rate by a factor of four.

The relation of start stress factor and start time is shown in Figure 6-11. The SSBL gas turbine and its auxiliaries have a start time of 15 minutes, and the expected start stress factor is 400. The operating MTBF for the SSBL and its auxiliaries, fuel, lubrication oil, controls, generator, etc., is 930 hours. On this basis the estimated starting reliability is 89 percent.

FINAL CONCEPTUAL DESIGN (HRGT) PLANT RELIABILITY IMPROVEMENTS

Balance of Plant Improvements

The Balance of Plant system design for the Korean Electric Company plant was used as the original reference. Improvements to this design were limited to a few selected areas as summarized in Table 6-7.

The gas turbine auxiliaries were based on the auxiliaries design for the FT 50 and an assumed spares inventory typical of present practice. On the basis of further study, initial improvements were identified and used to characterize

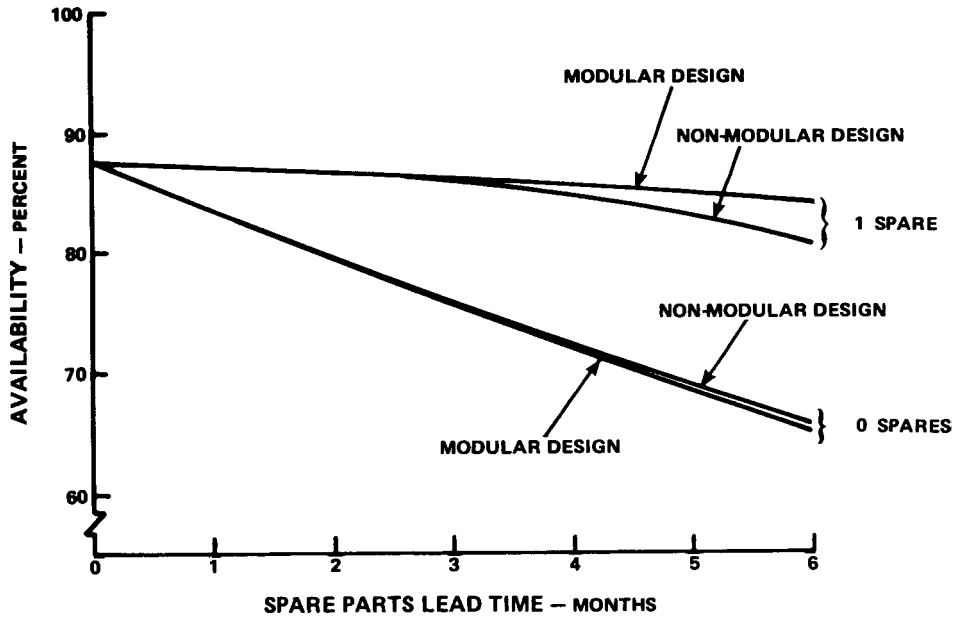


Figure 6-10 Sensitivity of Availability to Spare Parts Lead Time - With a modular design and one spare on site, availability in the mid 80 percent range can be maintained.

auxiliaries for the final HRGT design. The assumed reference SSBL and improvement for the HRGT auxiliaries are presented in Table 6-8.

High Reliability Gas Turbine Plant Reliability Calculation Results

The final High Reliability Gas Turbine Combined Cycle Power Plant conceptual design is calculated to have an estimated equivalent availability of 92 percent, and an operating equivalent reliability of 96 percent. The final result shows an availability improvement of some 5 percent (from 87 percent to 92 percent) over the initial SSBL plan design. Both of these values exceed the initial program goals of 90 percent and 95 percent respectively. Starting reliability for the gas turbines is estimated to be 91 percent.

The power plant reliability and availability are derived from the individual reliabilities of all of the power plant elements. These elements are grouped into the same thirteen categories according to their function as developed for the SSBL and shown in Figure 6-1.

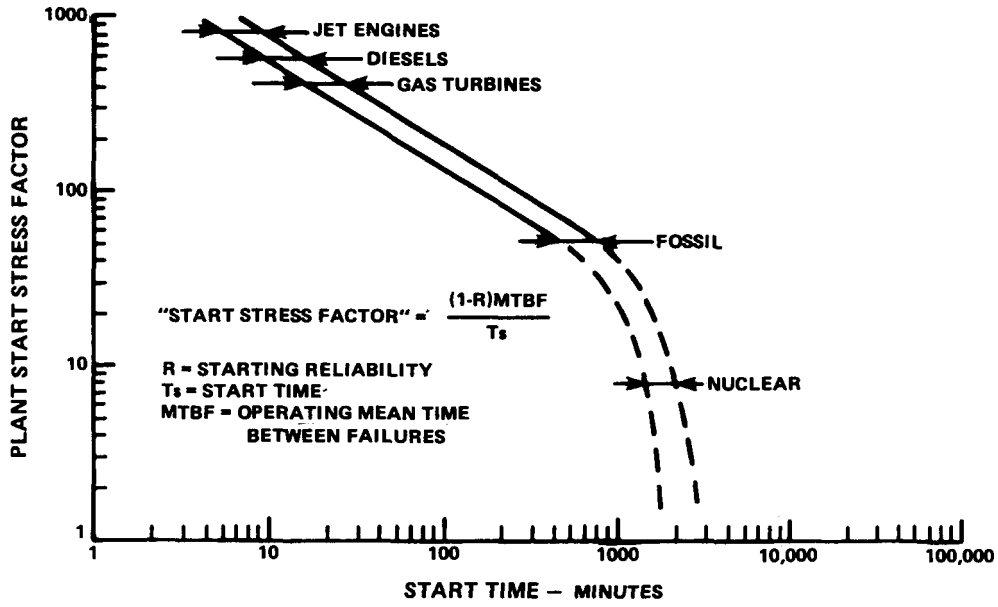


Figure 6-11 Effect of Start Time on Plant Start Stress Factor - The start time for the SSBL gas turbine and its auxiliaries is fifteen minutes with a stress factor of 400.

The breakdown of block reliabilities is presented in Table 6-9 using a format and definitions consistent with definitions outlined by EPRI. Also shown in the table are the block mean time between failure (MTBF), mean corrective time (MCT), planned outage hours (POH), and forced outage hours (FOH). The details of component failure rates and MCT for each component within the block is presented in Appendix D. The mathematical model is presented in Appendix B. Combining the block reliabilities and availabilities, the power profiles were developed as presented at the bottom of Table 6-9. Also shown are the resulting values of total power plant equivalent availability and reliability.

A power plant failure analysis showed that the plant power is available in six discrete levels from full power to zero power, hence Table 6-9 shows six power probabilities. There is an 81 percent probability that the plant will deliver full power, or conversely the plant is expected to deliver full power 81 percent of the time, including a scheduled maintenance shutdowns.

The availability characteristic estimated for the plant is shown in Figure 6-12. This chart shows the percentage of plant nameplate power available as a

TABLE 6-7. BALANCE OF PLANT RELIABILITY IMPROVEMENTS FOR HRGT-CC

	Reference	HRGT-CC
Main Circ. Water Pumps	2-50%	3-50%
Steam Turbine Generator POH	600 Hrs/Yr.	Reduced to 400 Hrs/Yr. (with condition monitoring)
Reference HRSG Controls		Improved HRSC Controls (per Table 5-3)
Instrument Air Compressor	2-100%	3-100%

percentage of calendar time per year. The integrated average availability -- the equivalent availability — is seen to be 92 percent, slightly above the EPRI Phase I goal of 90 percent. Details on the analyses used to predict availability and methods are presented in Appendices B and D.

The levels of availability and reliability are the result of improvements incorporated in the gas turbine and the balance of plant design during the Phase I studies. The balance of plant improvements come primarily from the application of redundancy, to such components as pumps, filters, and instrumentation. In addition, we have specified solid state controls and included a power plant control integration concept. An on-line condition monitoring system has been included to minimize dependence on scheduled maintenance and to permit maintenance "on condition" or as required.

TABLE 6-8. AUXILIARIES IMPROVEMENTS FOR HRGT

Reference	HRGT
Lube Oil Cooler - Air Cooled	Cooled by Plant Aux Cooling Water
Fuel Control (Solid State) Sequencer (Relay)	Integrated Controller (Solid State with Manual Takeover)
Instrumentation - Single Sensor	Three Temp Sensor plus median select
Main Fuel Pump - No Spare	Spare on Site
Flow Divider - No Spare	Spare on Site
Output Transformer 3 Per Total Plant	One Spare per Site or Local System
Auxiliary Transformers 3 Per Total Plant	One Spare per Site or Local System

Starting System Reliability Evaluation

A more detailed study of the starting system reliability was conducted subsequent to the initial starting system trade-off study described in Section 5. Manufacturers of start drive system equipment lack statistical information on starting reliability.

Review of information from EEI data and other sources suggests that starting reliability is influenced by the reliability of the system controls, start sequencer, and instrumentation more than the reliability of the main drive components and their mechanical accessories. For example, EEI data (EEI 77-64 Report on Equipment Availability for the Ten Year Period 1967-1976 and EEI 77-64A Equipment Availability Component Cause Code Summary Report for the Ten Year Period 1967-1976) shows that large gas turbines, which use electric motor drive for start, have a starting reliability of 82.7 percent. Aircraft derivative gas turbines which use compressed air start motors, have a starting reliability of 85.3 percent. However, a review of statistical data on gas turbine electric start motors and jet engine start systems shows a rate of less than one unsuccessful start associated with the start drive per 300 start attempts, as summarized in Table 6-10.

TABLE 6-9. POWER PLANT RELIABILITY AND AVAILABILITY (PHASE I RESULTS)

POWERPLANT RELIABILITY & AVAILABILITY (PHASE I RESULTS)

* * * * * * * * * * * * * * * INPUT * * * * * * * * * * * * * * *

BLOCK		MTBF	MCT	POH	FOH	REL. FACTOR	AVAIL. FACTOR
1	C/T TRANSFORMER	487000.	203.	0.	4.	0.999583	0.999583
2	C/T GENERATOR	7080.	76.	0.	93.	0.989380	0.989380
3	COMBUSTION TURBINE	8264.	88.	230.	90.	0.989740	0.963485
4	C/T ANCILLARY, EXCL. BOOST PUMP	1550.	32.	0.	177.	0.979772	0.979772
5	BOOST PUMP ASSY	33900.	37.	0.	10.	0.998910	0.998910
6	DAMPER	6500.	20.	0.	27.	0.996933	0.996933
7	BOILER	4220.	16.	0.	33.	0.996223	0.996223
8	FEEDWATER PUMP ASSY - HIGH	14000.	222.	0.	137.	0.984390	0.984390
9	FEEDWATER PUMP ASSY - LOW	25400.	134.	0.	46.	0.994752	0.994752
10	S/T ANCILLARY, EXCL. FEED PUMPS	5345.	31.	0.	51.	0.994234	0.994234
11	STEAM TURBINE/GENERATOR SYSTEM	8000.	24.	400.	25.	0.997146	0.951483
12	S/T TRANSFORMER	576000.	221.	0.	3.	0.999617	0.999617
13	OUTPUT BUS (2 IN //)	3300000000.	1090.	0.	0.	1.000000	1.000000

Stal Laval's GT200 operation has undergone 89 starts to date with only one unsuccessful attempt related to the start system. Evidently electric start motors and compressed air start systems are in themselves highly reliable. Start failures for gas turbines are primarily caused by problems in other equipment such as sensors and controls.

As further indication, a reliability study was conducted by the Army and reported in Power Magazine in January, 1970 ("How Reliable are Today's Prime Movers?" By R. W. Parisian, Bechtel Corp.), on diesels and small gas turbines for both operation and starting. In that study, starting reliability of 70 diesel generators in the 1500 kilowatt size, was analyzed. Most of the failures were attributed to the governor, voltage regulator and frequency controls. Probability of failure to start was calculated to be about 1 percent. Small industrial gas turbines had a failure to start probability of about 6 percent. Both of these starting reliabilities are relatively good and indicate an acceptable inherent reliability for starting.

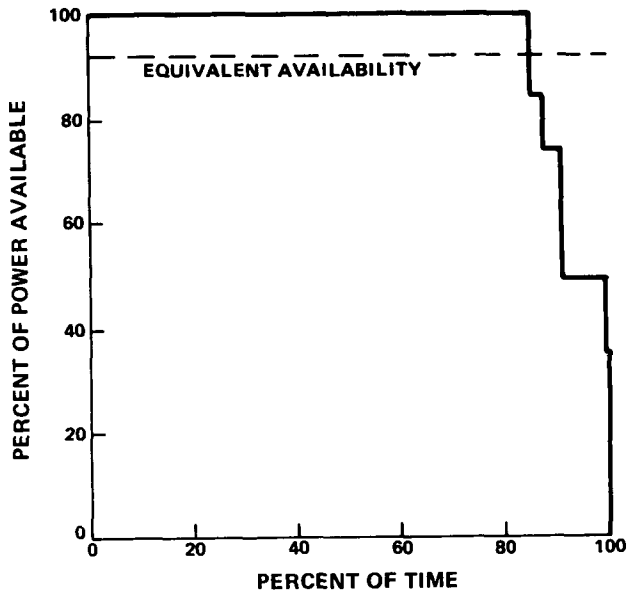


Figure 6-12 Availability Characteristic for High Reliability Gas Turbine Combined Cycle Power Plant - The equivalent availability is 92 percent.

TABLE 6-10. SUMMARY OF EEI DATA ON START DRIVES

	Avg. Number Outages per Unit Year	Attempted Starts Per Unit Year	Starts per Start Drive Failure	Starting Reliability
Gas Turbine				
Start Motor (electric) (Code 322)	0.5	179	358	99.72%
Jet Engine				
Start Motor (air) (Code 367)	0.58	183	316	99.68%

On the basis of this information, it appears that the inherent starting reliability is high for all of the alternatives, given proper attention to instrumentation and controls and the plant start sequencer.

Recommendations Related to Program Goals

Near the conclusion of the Phase I program, the EPRI suggested goals for the Phase II program. In response to these suggested Phase II goals, a brief review was made of the elements of the power plant that have relatively low individual availability. The required MCT's and POH's were also examined.

From this review, new values of these parameters were developed assuming various improvements which could be made to the power plant. The result is presented in Table 6-11. MTBF's for gas turbine and ancillaries were increased and POH's were reduced for the gas turbine and steam turbine. This increased the full power probability to 86 percent and equivalent availability to 94 percent. In the Phase II program, additional reliability improvements to both the gas turbine and the balance of plant will be implemented. Additional system redundancy can be incorporated in the gas turbine auxiliary systems and the balance of plant systems to improve MTBF.

Based on the improvements shown in Table 6-11, preliminary projections for Phase II have been established and are shown in Table 6-12. The table also shows United Technologies Corporation's Phase I final results and EPRI's suggested goals for comparison. The table shows that an equivalent availability in Phase II of 95 percent is projected. This is equal to EPRI's suggested goal. Rated power availability is also expected to reach the 90 percent suggested goal.

MTBF for the gas turbine system and total power plant are also shown in Table 6-12. These MTBF projections are consistent with the reliability and availability projections. MTBF on these elements will be improved by adding redundancy as required to meet the power plant availability goals. An important point to note is that the mean corrective times for auxiliary and BOP items are generally very low and failures do not have a strong effect on power plant equivalent availability. This is particularly pertinent in the case of a gas turbine combined cycle power plant where partial outages (especially in the steam system) have only small to moderate effects on power output. In this power plant, for example, a gas turbine system MTBF of 4,000 hours is sufficient to

TABLE 6-11. POWER PLANT RELIABILITY AND AVAILABILITY IMPROVEMENT ANALYSIS

* * * * * * * * * * * * * * * * INPUT * * * * * * * * * * * * * * * *

BLOCK		MTBF	MCT	POH	FOH	REL. FACTOR	AVAIL. FACTOR
1	C/T TRANSFORMER	487000.	203.	0.	4.	0.999583	0.999583
2	C/T GENERATOR	7080.	76.	0.	93.	0.989380	0.989380
3	COMBUSTION TURBINE	9000.	59.	100.	55.	0.993670	0.982254
4	C/T ANCILLARY, EXCL. BOOST PUMP	1640.	33.	0.	173.	0.980275	0.980275
5	BOOST PUMP ASSY	33900.	37.	0.	10.	0.998910	0.998910
6	DAMPER	6500.	20.	0.	27.	0.996933	0.996933
7	BOILER	4220.	16.	0.	33.	0.996223	0.996223
8	FEEDWATER PUMP ASSY - HIGH	14000.	222.	0.	137.	0.984390	0.984390
9	FEEDWATER PUMP ASSY - LOW	25400.	134.	0.	46.	0.994752	0.994752
10	S/T ANCILLARY, EXCL. FEED PUMPS	5345.	31.	0.	51.	0.994234	0.994234
11	STEAM TURBINE/GENERATOR SYSTEM	8000.	24.	300.	25.	0.997111	0.962865
12	S/T TRANSFORMER	576000.	221.	0.	3.	0.999617	0.999617
13	OUTPUT BUS (2 IN //)	3300000000.	1090.	0.	0.	1.000000	1.000000

* *

POWER PROFILE & AVAILABILITY

AVAILABLE POWER (%)	PROBABILITY	CUM PROBABILITY
100.0	0.855464	0.855464
84.4	0.012239	0.867702
70.9	0.059087	0.926790
48.9	0.066397	0.993187
35.5	0.004533	0.997720
0.0	0.002280	1.000000

EQUIV. AVAILABILITY FACTOR = 0.9374 93.7%

POWER PROFILE & RELIABILITY

AVAILABLE POWER (%)	PROBABILITY	CUM PROBABILITY
100.0	0.906601	0.906601
84.4	0.012970	0.919571
70.9	0.028418	0.927989
48.9	0.069530	0.997519
35.5	0.001135	0.998654
0.0	0.001346	1.000000

EQUIV. RELIABILITY FACTOR = 0.9579 95.8%

meet the projected CT system availability of 95 percent and reliability of 97 percent. Likewise for the total power plant, an MTBF of 1,000 hours is sufficient to meet the plant full availability and reliability of 90 and 95 percent, respectively, In this discussion, MTBF defined by EPRI is any failure resulting in a 2 percent or greater loss in power.

To put the gas turbine combined cycle reliability in perspective, a comparison is made with the fossil steam power plant in Table 6-13. The combined cycle power plant full forced outage MTBF is seen to be over ten times longer than that of the steam plant. This is because of the extremely low probability that both gas turbines will experience an outage at the same time.

Another approach to meeting the EPRI rated power availability goal and reducing the sensitivity to scheduled maintenance is the use of an installed standby gas turbine system as defined by Block 1 through 9 on the reliability block diagram. This approach would increase the capital cost about 22 percent, but would not

TABLE 6-12. GAS TURBINE COMBINED CYCLE POWER PLANT GOALS

Item	UTC Phase I Results	UTC Phase II Projection	EPRI Phase II Goals (Suggested)
Power Plant			
Availability, Full-%	81	90	90
Equivalent-%	92	95	95
Op. Reliability Full-%	90	95	95
Equivalent-%	95	98	90
Start Reliability-%	--	90	90
MTBF - Hrs.			
Comb. Turbine - Hr.	8364	9000	9000
CT System - Hrs.	1100	4000	6000
Power Plant - Hrs.	350	1000	3000
Gas Turbine System			
Availability, Full-%	94	95	95
Op. Reliability, Full-%	96	97	97
Start Reliability-%	91	95	95
Major Maintenance Period-Hr.	18,000	18,000	18,000
Hot Sec. Parts Life-Hrs.	50,000	50,000	50,000
Cold Sec. Parts Life-Hr.	100,000	100,000	100,000

raise the 30-year levelized power cost by more than 1-2 percent because of the dramatic reduction in unscheduled outage rate from about 5 percent down to less than 1 percent. The approach of one or more standby spare gas turbine systems in a larger installation, such as a 1,000 MW power plant also should be considered.

The MTBF goal of 9,000 hours for the gas turbine will serve as a basis for engine preliminary design to be conducted in Phase II. The gas turbine starting reliability is projected to be 95 percent. Existing units at well maintained sites such as LILCO's Holbrook peaking station have a starting reliability of 94 percent and the HRGT will have greater control redundancy which will improve the starting reliability. A 90 percent plant starting reliability is expected based on the use of an integrated plant control system.

TABLE 6-13. HRGT-CC VS. FOSSIL PLANTS

	Phase I HRGTPlant	Fossil Plant	
Size, MW	300	200-389	200-299
Source	Phase I Analysis	EEI Data 65-74	EEI Data 67-76
MCT, Hrs - GT System	47	62	65
Power Plant MTBF, Hrs	343	646	673
CT System, MTBF, Hrs	1,100		
Full Forced Outages MTBF, Hrs (1)	12,872(1)	1,126	1,105

(1) The mean-time between full-forced outages for the HRGT conceptual plant is:

$$\text{MTBF (Full Forced Outages)} = \frac{(\text{GT SYSTEM MTBF})^2}{2 \times \text{MCT}} = \frac{(1100)^2}{2 \times 47} = 12{,}872$$

This reflects the probability of both combustion turbine systems failing at the same time causing a full outage.

(2) Based on Meeting the EPRI GT System MTBF goal of 6,000 hours and a 47 hour MCT.

SECTION 7.0

GUIDANCE FOR PHASE II PRELIMINARY DESIGN

INTRODUCTION

During the Phase I conceptual design study, we completed several subsidiary tasks whose major purpose was to lay the groundwork for continuing into subsequent program phases. The results of these tasks are summarized in this section. Some of the material generated for subsequent phases took the form of stand-alone documents. These documents are briefly reviewed at the end of this section. The actual documents are included in Section 9 of this report.

Subsidiary tasks included:

- An assessment of the new technology associated with the new centerline conceptual design. This includes an outline of required R&D program plans, and an evaluation of that technology which might be applied by retrofit to the current FT4 engine.

- A survey of fuel flexibility background to establish combustor design requirements. This included separate consideration of both coal-derived liquid and gaseous fuels.

- An in-depth study of the gas turbine control concept. Specific functions and requirements are laid out in sufficient detail to serve as a basis for the control manufacturer to initiate preliminary design.

- Consideration of alternate (peaking) service. An evaluation was made of features which might be incorporated into the design that would accommodate peaking service with an engine primarly designed for base load service.

- Condition monitoring. Design features and an overall scheme were defined which provide the capability for maintenance to be performed based on the continuous monitoring of the engine's health status.

- The following three guidance documents: Gas Turbine Design Criteria; Ancillary Equipment Design Criteria; and Reliability Program Plan.

TECHNOLOGY ASSESSMENT

Key Technology Research and Development Programs

The components for the engine were selected based on the use of existing technology to ensure that the reliability goals can be achieved and the engine will be available by 1986. The only component where the use of existing technology was not adequate was the combustor. Advanced technology is required to satisfy the emission requirements when the engine is operating on residual oils, or coal derived liquids or gases. An advanced combustor concept has demonstrated the potential of achieving the emission requirements with the desired fuels. Further R&D is required to meet the emission requirements, combustion stability, and liner durability over the full range of power, and to determine the allowable fuel property limits.

A second area where some additional development would be desirable is the compressor. A low aspect ratio compressor was selected for the powerplant to improve reliability, durability, and reduce cost. The concept has been demonstrated in compressor rigs which confirmed the analytical models. However, tests with a subscale compressor rig of the HRGT design to investigate and confirm flow dynamics during start-up and over the full power range and to verify performance estimates would be useful and cost effective. The combustor and compressor R&D program plans will be described in the following sections.

Improvements in materials, coatings, turbine cooling, and manufacturing methods are expected from other programs that would improve the reliability, and/or reduce the cost of the HRGT powerplant. Advances in the state of the art in these areas are not required for the engine design selected. The improvements will be incorporated into growth versions of the engine. Since R&D programs in these areas are not essential for the engine, the plans were not defined at this time. It is anticipated that some programs to obtain specific design data or design limits in specific areas will be required. The detailed design studies in Phase II will point out the areas where this additional design data is required. The programs required to obtain this data will be defined in Phase II of the program.

Combustor R&D Program

Research programs sponsored by EPA and EPRI have been in progress for some time to develop a dry combustion system to meet the NO_x emission requirements. Over 15 concepts have been evaluated and the rich burn/quick quench (RB/QQ) concept has demonstrated the ability to meet the NO_x emission requirements not only from thermal NO_x but also from fuel bound nitrogen. This concept was selected to provide the wide fuel flexibility desired. The powerplant can burn distillate or residual petroleum and coal-derived fuel oils as well as natural gas and low or medium Btu gasified coal by replacing fuel injectors or by replacing both the injectors and the burner cans.

The research programs have shown the potential of the RB/QQ combustor concept to meet the emission and fuel flexibility requirements. Additional research and development will be required to achieve the reliability, durability, and efficient, stable combustion over the full power range required by a powerplant. The combustor R&D program, shown in Figure 7-1, is divided into three tasks; combustor research, subscale development, and engine development.

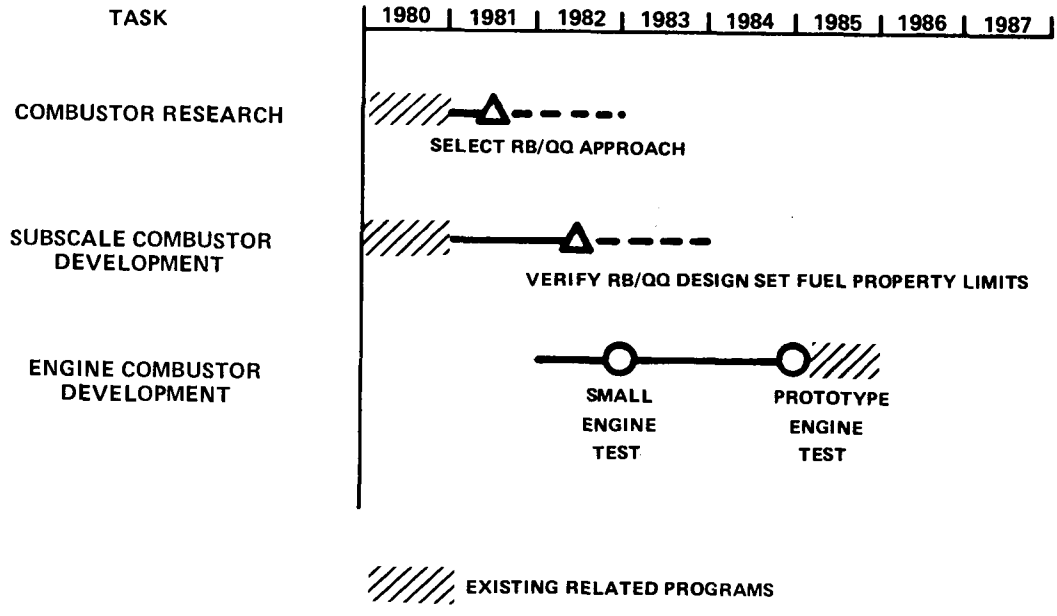


Figure 7-1 Combustor Development Program Plan

The purpose of the combustion research program is to provide fundamental data for the models and combustor design studies, screen and evaluate design concepts to be tested in subscale combustors, and test alternate petroleum and coal-derived fuels to further document the effect of fuel properties on emissions, combustor cooling load, and combustor efficiency. Previous tests have shown that the primary fuel air ratio must be controlled to meet the emission requirements at all power levels. Thus, an early task will be to evaluate simple methods of either controlling primary airflow or staging injectors. Other areas of investigation will be improved atomization and vaporization of heavy fuels, and methods of reducing the carbon formation and thus the liner cooling load with heavy coal-derived fuels. This task will provide information and data to the Phase II combustor design study and the subscale combustor development program. The priorities of this task will be based on the input from these tasks. The result of this task will be the selection of the specific design approach for the RB/QQ combustor.

The research combustor rigs are designed to easily vary operating conditions and design parameters. As such, they do not fully represent a combustor. The subscale combustor program will be used to verify and develop the design concepts selected in the combustor research and design programs. Various coal-derived liquid and gaseous fuels as well as residual oil will be tested to demonstrate the fuel flexibility capability and that not only the NO_x but also the CO, unburned hydrocarbons, and particulates meet the emission requirements. The result of these two tasks and the combustor design study of Phase II will be the verification of the specific design details of the HRGT combustor, and selection of the fuel property limits for the powerplant.

The final task will be to retrofit a FT4 engine with the RB/QQ combustor concept selected for the HRGT powerplant. The primary objective is to demonstrate the combustor reliability and durability under actual engine operating conditions. The program will be completed with the testing of a prototype powerplant.

Much of the first-year combustor R&D program is covered under existing programs or a program about to be started. A program sponsored by EPA is in progress to test the RB/QQ burner of the FT4 size on a coal-derived liquid fuel, a shale oil, and a residual oil. This will provide data on the impact of fuel properties on combustor performance and emissions. A program sponsored by DOE/NASA is about to start that will investigate improved fuel preparation, staging, and screen

concepts to meet emission requirements over the power range. A program sponsored by EPRI is continuing to develop burner concepts for low and medium Btu coal gasified by the Texaco process.

Compressor R&D Program

Low aspect ratio compressors have been investigated analytically and experimentally for several years. The concept was first demonstrated by tests on three and four stage rigs representative of compressor inlet and exit conditions. In 1977, a seven stage rig of 7.5:1 compressor ratio demonstrated high efficiency and good surge margin. A rig of 18:1 compression ratio consisting of six axial and one centrifugal stage has been designed, fabricated, and assembled under a DOE program. This test should verify the design procedure concerning performance, surge, start-up dynamics, and structural integrity.

The compressor development program for the HRGT engine is shown in Figure 7-2. Most of the required program is contained in the related programs. While the amount of additional development required will depend upon the results of the tests of the 18:1 compression ratio rig, it is anticipated that the only additional effort required will be the design, fabrication, and test of a subscale compressor based on the detailed compressor design from the Phase II program. This program may not be a necessity, but it is probably a cost effective technique to ensure that the last few performance points are achieved, and to determine the specific details of the variable geometry guide vane and compressor bleed valves for reliable, surge-free start-up. The program will be completed in time so the results can be used to update and refine the compressor design for the prototype engine.

RETROFIT ANALYSIS

Gas Turbine

Potentially, new technology from the high reliability development program could be used to advantage in our current product, the FT4. Some consideration was made of what specific technologies might be applicable.

The FT4 is an engine originally derived from an aircraft antecedent (JT4) which has seen many years of service in both utility peaking and industrial service.

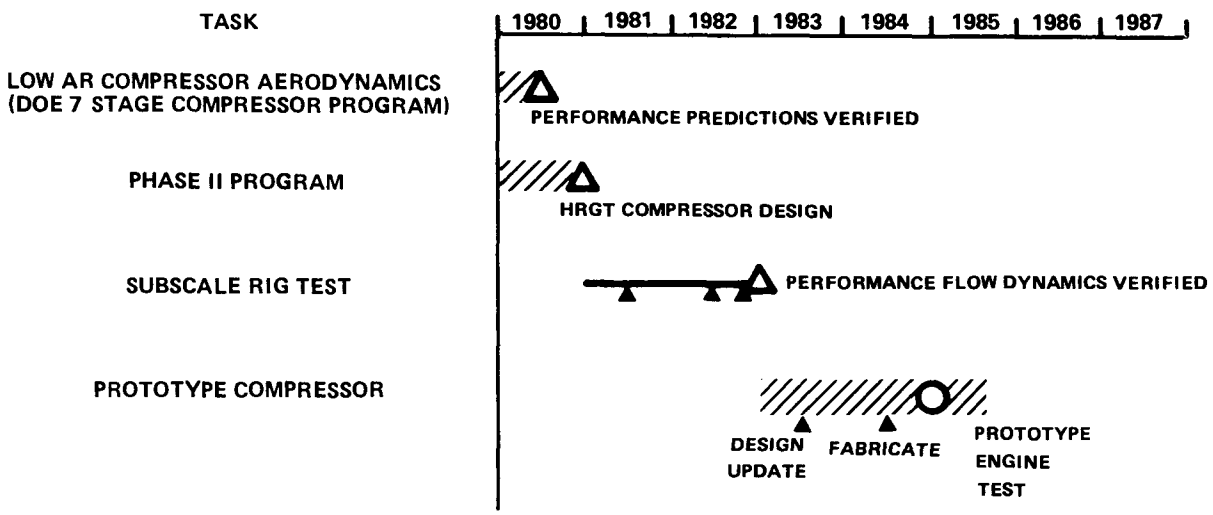


Figure 7-2 Compressor Development Program Plan

As such, the size-class (30 mw) and basic configuration turn out to be significantly different from the final (107 mw) HRGT design as evolved during Phase I.

Nevertheless, future FT4 models could conceivably benefit from two general technology areas deemed essential for meeting HRGT goals: (1) Rich-Burn-Quick-Quench Combustion; and (2), all-convective cooled advanced turbine airfoils.

Indeed, the FT4 has already served as the test vehicle for the rich burn quick quench technology. A series of tests of a full-scale rich burn quick quench combustor was run by Pratt & Whitney Aircraft in an FT4 engine in West Palm Beach, Florida during 1979. These tests were highly successful in proving that NO_x emissions requirements could be met even with heavy, nitrogen-laden fuels. Further concept proof requires continuous operation to develop hardware durability. Such testing, using the FT4, is envisioned as part of the R&D development program, as is recommended in the foregoing sub-section. Once perfected as part of such a program, rich burn quick quench is automatically retrofitable into the FT4, by virtue of that test program.

As regards new turbine airfoil cooling technology, as the two-piece construction and advanced cooling capability becomes developed, it would have application to the FT4 for the same basic reason. The need for such technology comes from the requirement to keep metal temperatures low to avoid sulfidation corrosion, while keeping firing temperature sufficiently high to result in competitive performance. This is just as true for the FT4 as it is for the HRGT. However, FT4 vanes and blades are inherently smaller than the 100 MW class HRGT, while the cooling air supply system would be different as well. Consequently, the technology would require undertaking a special R&D effort before it could be successfully retrofit into a new FT4 model.

Gas Turbine Auxiliaries and Balance of Plant

As a result of the reliability analysis and site survey conducted in the Phase I program, gas turbine auxiliary and balance of plant components and innovations have been identified as being potentially beneficial for reliabiliy, availability, and cost if electricity, as follows:

- Review and revise spare component and parts inventories considering cost, replacement lead time, failure rates, and impact on unscheduled outage rate to minimize cost of electricity for specific sites. A computer program could be prepared to establish component and parts inventories at levels to minimize the cost of electricity.

- Upgrade the reliability of the gas turbine electronic fuel control with additional sensor and circuit redundancy, rate of change lock out, and self test features, to reduce the number of nuisance shutdowns. Because the mean corrective time for fuel controls is only two to four hours, the availability and forced outage time are not significantly affected. Therefore, the cost of electricity would not be noticeably affected.

- Upgrade general system instrumentation such as temperature, speed, and vibration sensors, and circuits with redundancy and median select features to reduce nuisance shutdowns. The cost of electricity would not be decreased because outages for instrumentation failures are relatively short. An increase in the cost of electricity may result, but should be relatively minor.

- Upgrade inlet air filtering to increase the intervals between compressor wash for performance recovery, and explore methods for on-line cleaning. The impact on availability for these improvements can be significant, and, therefore, the cost of electricity should be reduced since availability has a strong influence on the cost of electricity.

- Install duplex oil and fuel filters with automatic changeover provisions and isolation provisions for on-line maintenance of the by-passed filter. This will reduce occasional nuisance shutdowns but will not significantly improve availability or impact the cost of electricity.

- Install an on-line condition monitoring system with diagnostic and prognostic capabilities to reduce periodic scheduled outages for inspection. Reduction in the scheduled outage time significantly improves availability. An analysis of cost and realistic availability improvement is needed because the cost of the condition monitoring system must be outwieghed by the cost of electricity benefits through improved availability.

ALTERNATE FUELS

Modifications for Coal Derived Liquids and Residual Oils

Identification of Fuel Properties

A design goal is to operate the HRGT power plant on residual oils and coal derived liquids to improve fuel availability and/or reduce the cost of electricity. However, the properties of these fuels can have a significant impact on the power plant emissions and durability. Previous experience and information in the literature have been reviewed to identify the fuel properties that affect component design, and to determine the properties of the fuels of interest. This review will be summarized in the following paragraphs.

Alternate fuels primarily affect the combustion process and emissions. Over the years various fuel properties and tests have been used to characterize fuels. These properties and tests have been correlated with operating experience and trends. However, there is little fundamental data. The trends observed are often

very sensitive to the combustor configuration. Many tests were developed for a specific type of combustor and often are not applicable in general.

The fuel properties (and tests) which are considered important for the HRGT application are listed in Table 7-1. Other parameters which are often measured and could have been included are smoke number, luminosity, pour point, aniline point, and bromine number. In general these parameters provide the same information as one or more of the properties listed. Which is preferable depends upon the specific use and personal preference. The three most important paremeters are $\%N_2$, specific gravity and $\%H_2$. The fuel bound nitrogen ($\%N_2$) has a major effect on the NOx produced and thus the difficulty of meeting the emission requirements which the power plant must satisfy. The specific gravity and $\%H_2$ are important since with a knowledge of the fuel source, i.e., petroleum or coal, the other parameters can be roughly correlated with these two parameters. The difficulty of burning the fuel cleanly and reliably increases as the specific gravity increases and the $\%H_2$ decreases.

TABLE 7-1

FUEL PROPERTIES TO CHARACTERIZE FUELS FOR HRGT APPLICATION

Specific Gravity
$\%N_2$
$\%H_2$ (Aromatics)
Volatility (IBP, FBP)
Viscosity
Metals
Ash
Carbon Residue
Thermal Stability
Sulfur

Coal can be liquified at high pressures and moderate temperatures in the presence of H_2. Three liquification processes are being developed which differ by the method by which the hydrogen is made available to the coal and the catalysts used. The three processes are the EDS (Exxon Donor Solvent) process of Exxon, the SRC (Solvent Refined Coal) process of Gulf Mineral resources, and the H-Coal process of Hydrocarbon Research. Each process produces a variety of products ranging from fuel gases and light liquids to the heavy bottoms of residual fuels. Some information is available on the properties of these coal derived liquids. This data shows that the fuel characteristics will not only depend upon the process selected, but will also depend upon the fuel cut selected, the liquification process conditions, and to some extent the fuel source.

The coal liquids can be upgraded to the quality required by further refining and hydrogenating.

As the fuels become available a product slate and specifications will evolve. The fuel properties and specifications for each product will depend upon the demand and availability of each product, and an economic balance between the cost of liquifying and upgrading the coal liquids versus the cost of utilizing heavier fuels in each application.

Since there are no specifications for coal derived liquids at the present time, the properties of the various coal liquids were reviewed and properties selected that would be representative of coal derived liquids equivalent to #2 oil and a heavy oil. These properties together with typical properties of petroleum #2 and #5 oils are listed in Table 7-2. The petroleum #5 oil is a residual oil blended with #2 oil to reduce the sulfur level to the value required to satisfy the emission requirements. The properties listed are not the worst that might occur. They were the values used to assess the impact of coal derived liquid fuels and residual oils on the HRGT power plant.

Impact of Coal Derived Liquids and Residual Oils

The components most affected by the fuel properties are the combustor configuration, the combustor cooling, fuel injectors, turbine, and the fuel pretreatment system. The fuel properties that are important to each component or design consideration are shown in Table 7-3. Further tests with the specific combustor configuration will be necssary to quantify the effect of these properties and select the limits for the fuel specifications.

The power plant must satisfy the emission requirements. The sulfur dioxide emission limits are achieved by limiting the sulfur level in the fuel. The NOx emission limits are achieved by use of the Rich Burn/Quick Quench (RB/QQ) combustion concept that was selected for the HRGT. Combustor tests with #2 oil altered to contain 0.5% fuel bound nitrogen demonstrated only a slight increase in the NOx over that with #2 oil. This test indicates a potential for achieving the NOx requirements even with the high nitrogen content of the coal derived #5 oil.

TABLE 7-2
REPRESENTATIVE FUEL PROPERTIES

	Petroleum		Coal Derived	
		Blended Resid.		
Fuel	#2	#5	#2	#5
Gravity (°API)	33.5	19	14	1
Volatility				
I.B.P. (°F)	350	350	370	545
E.B.P. (°F)	650	>1000	545	935
H_2 (Wt %)	12.8	10.7	9.1	7.5
N_2 (Wt %)	.08	.25	.8	1.3
S (Wt %)	.09	.5	.2	.4
Ash (Wt %)	<.003	.03	.06	.1
Viscosity (C.S. @ 120%)	2.6	40	3	70
Carbon Residue or 10% bottoms	.14	4*	1	1.5**
Vanadium (ppm)	<.1	35	<.1	1
Sodium (ppm)	<.1	10	<.1	5

*Dial Sample
**SRC II

TABLE 7-3

FUEL PROPERTIES SIGNIFICANT TO HOT COMPONENTS

Design Consideration	Fuel Property
Emissions	
NOx	$\%N_2$
SO_2	Sulfur
Smoke, combined hydrocarbons, CO	Volatility, viscosity $\%H_2$, carbon residue, thermal stability
Fuel Injections	
Spray/atomization quality durability, plugging	Viscosity, volatility thermal stability
Combustion Durability	
Liner heat load	$\%H_2$, carbon residue, volatility
Hot spots	Thermal stability
Turbine Durability	
Temperature pattern/hot spots	Carbon residue thermal stability
Erosion	Ash, carbon residue
Corrosion	Metals
Fuel Pretreatment	
Additives	Metals
Preheat	Viscosity
Atomization air	Viscosity

The hydrogen content of the coal derived fuels is lower than #2 oil indicating that these fuels contain a high percentage of unsaturated, aromatic compounds. In general aromatics in the fuel increase the amount of carbon in the flame, increasing the radiation and heat duty on the combustor liner and increasing the smoke emissions. This could impact the durability of the rich burn section of the RB/QQ combustor where the amount of cooling available is limited. A review of the liter-

ature revealed that NASA has demonstrated that it is possible to design a combustor where the liner temperature factor (heat duty) does not increase with decreasing hydrogen in the fuel, as shown in Figure 7-3. The data of Wright on the oxygen/carbon ratio at which carbon formation occurs with premixed fuel and air in a well stirred reactor was reviewed. The results, shown in Figure 7-4, indicate that the type of compound (%H_2) or the number of carbon atoms in the molecule have no effect on the O/C required to prevent carbon. The data also shows that the O/C in the primary section of the RB/QQ combustor is outside that carbon limit. It can be concluded from this data that decreasing the hydrogen content of fuel increases the carbon formation in the burners where carbon formation occurs, but does not cause carbon formation in carbon free burners. It also shows that with proper fuel injection and mixing, carbon free operation is potentially possible in the RB/QQ combustor even with coal derived fuels.

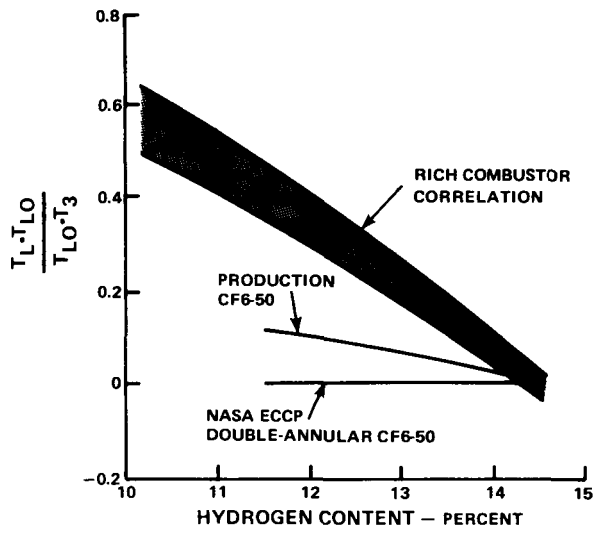


Figure 7-3. Comparison of Modern and Advanced Designs with Rich Combustor Correlation (CF6-50 and NASA ECCP Data From Reference (1))

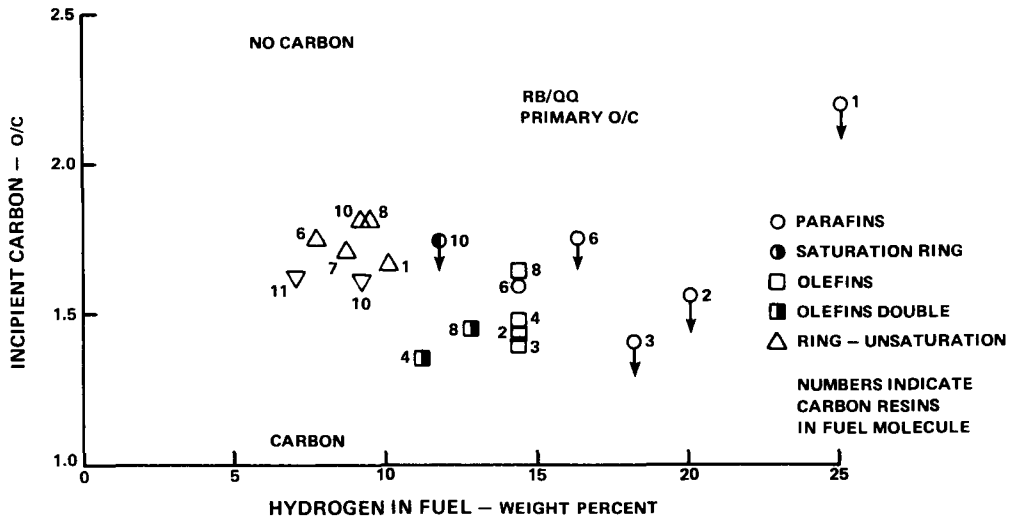


Figure 7-4 Impact of Fuels on Incipient Carbon in a Well Stirred Reactor (Wright)

The well stirred reactor tests were conducted using fuels which could be fully vaporized and premixed at a preheat temperature of 340 to 475°F. There is a concern that the trends would be different with heavy, low volatility fuels which could not be fully vaporized before combustion, such as the blended residual oil. A FT4 engine has run on a blended residual oil for over 750 hours. An air atomizing nozzle was used to atomize the fuel. While the combustion was practically smokeless, there is a need to increase the cooling of the combustor dome to offset the effect of the flame being closer to the dome with the air atomizing nozzle. This test indicates that with very good fuel atomization and mixing, the small droplets of unvaporized fuel act the same as vaporized fuel. While the ash content of the blended residual oil in this test is only 0.016%, the preliminary results indicate that the fine ash particles from air atomized fuel causes very little erosion of the turbine.

The RB/QQ combustor selected for the HRGT engine will be designed and developed to utilize coal derived liquid fuels. Thus, for a #2 type coal derived fuel no modifications or design changes will be required and there will be no impact on the reliability or availability of the power plant. The only impact on the cost of electricity will be the cost difference of the coal derived fuel.

The impact of burning residual oils was investigated as part of Phase I trade-off studies. The study concluded that with the RB/QQ burner concept to satisfy the NOx emission requirements, operation on residual oils was possible with the following modifications.

- Addition of a fuel treatment system to reduce the sodium and potasium, and adding neutralizing agents to the fuel to reduce the effects of vanadium and silicon,
- Periodic water wash of the fuel injectors and turbine to remove deposits,
- Steam heating the fuel to reduce the viscosity to ease fuel pumping and atomization,
- Addition of a compressor to air atomize the fuel, and
- Start-up on a light, clean fuel.

Unless the sulfur level of the residual oil is low enough to satisfy the emission requirements, the residual oil would be blended with #2 oil to lower the sulfur level to an acceptable level. The reliability analysis indicated that the use of the blended residual oil would have no effect on reliability provided the hot sections were inspected more frequently. The more frequent inspections would reduce the equivalent availability 1%. The lower cost of the blended residual oil more than offset the slightly higher plant cost. The net result was a 2.6 mill/kWH reduction in the cost of electricity. The results of the study are summarized in Table 7-4.

The properties of the heavy coal derived liquid fuels are similar to that of the blended residual oil except for the lower H_2 content of the fuel. The use of coal derived liquid fuels will require no further modifications than that for the residual oil. Assuming that the R&D program confirms the analysis that with good atomization and mixing the liner heat duty is not increased with the lower $\%H_2$, the power plant reliability and availability will be the same as with residual oils. The cost of electricity will also be the same except for the effect of the fuel cost. The liquification of coal is a very capital intensive process. Only about 25% of the fuel cost is due to the cost of coal. Thus, the cost of the coal derived fuel will depend not only upon the process which has still to be selected, but will be very dependent upon government action relative to the financing of the plant and/or subsidizing of the fuel cost. The projected cost ranges from being equal to two times the cost of #2 oil.

TABLE 7-4
FUEL FLEXIBILITY STUDY RESULTS

Fuel	Reference	Residual Oil Alternate "A"	Residual Oil Alternate "B"	Residual Oil Blend
Type	No. 2	No. 6	No. 6	No. 4
Sulfur		<.5%	1%	~0.5
Vanadium, PPM		25	70	35
Fuel Cost, $/10^6 BTU	4	3.82	3.58	3.79
Performance				
Heat Rate, HHV, ISO BTU/KWHr	7615	7700	7700	7658
Reliability				
Gas Turbine, MTBF, Hrs	7855	7855	7855	7855
Gas Tubine, SOH, Hrs	400	528	528	450
Overall Plant EFOR, percent				
Repair Parts Lead Time				
0	6.6	7.0	7.0	7.0
1 month	14.51	14.9	14.9	14.9
2 months	21.18	21.5	21.5	21.5
Equivalent Availability, percent				
Repair Parts Lead Time				
0	87.4	85.6	85.6	86.4
1 month	79.9	78.3	78.3	79.1
2 months	73.3	72.3	72.3	73.0
Relative Plant Capital Cost	1.0	1.03	1.03	1.02
Cost of Electricity Change, Mills/KWH				
Repair Parts Lead Time / Utility System Characteristic				
0 m = 100	0	-1.26	-4.89	-2.63
m = 500	0	-1.43	-5.06	-2.75
m = 1000	0	-1.44	-5.06	-2.75
1 month m = 100	0	-0.8	-4.43	-2.26
m = 500	0	-1.24	-4.87	-2.61
m = 1000	0	-1.28	-4.91	-2.63
2 months m = 100	0	-0.59	-4.22	-2.14
m = 500	0	-1.14	-4.77	-2.55
m = 1000	0	-1.19	-4.82	-2.58

Modifications for Coal Derived Gaseous Fuels

Coal will be the primary fuel for many of the future power generators. Studies have shown that a combined cycle system integrated with a coal gasifier is an efficient, economical method of generating electricity from coal and satisfying the emission requirements. Thus, the ability to modify the HRGT engine to utilize gasified coal is an important design consideration. The information on the characteristics of coal gasified by various processes, and the results of combustor tests with gasified coal have been reviewed. A brief study was made to determine

the modifications required to utilize gasified coal and to estimate the impact on the reliability of the HRGT engine. For this phase of the program, the study was limited to the gas turbine and ancillaries and did not consider the gasifier. The results of these studies are summarized in the following sections.

Gasified Coal Characteristics

Numerous processes have been or are being developed to gasify coal. The information available in the literature on the composition of the gas produced by the various gasifiers has been reviewed. This review shows that the composition of the gasified coal depends upon the process, the process operating conditions, the specific coal being gasified, the oxygen concentration, and the type of gas clean-up and processing after the gasifier reactor. The potential range of gas compositions for gasified coal is shown in Table 7-5. The gasified coal is generally classified in three categories depending upon the heating value of the gas, i.e. low, medium or high BTU. The low BTU gas is coal gasified with air and the fuel gas is diluted with the nitrogen from the air. The medium BTU gas is coal that has been gasified with oxygen. The high BTU gas is coal gasified with oxygen which has been further processed to remove CO_2 and convert the hydrogen and carbon monoxide to methane by methanation.

Combustor tests have shown that all of these fuels can be burned in a conventional combustor with a natural gas fuel nozzle. The wide range of fuel composition has only a slight impact on the combustion characteristics. The fuel gas composition does have a significant effect on the NOx produced by the combustor. The measured amount of NOx produced by the combustion of low BTU gas, medium BTU gas, and natural gas, is shown in Figure 7-5. The NOx produced increases as the maximum flame temperature increases. The measurements were taken while the combustor was operating at 1 atmosphere pressure. At HRGT operating pressures, the NOx produced would be higher and would exceed the allowable NOx emission limits with medium BTU and natural gas. Thus, a low NOx combustor is required to meet the emission requirements with medium BTU or high BTU gasified coal but is not necessary for low BTU gas.

The sulfur level of coal exceeds the allowable emission levels for SO_2. To satisfy the SO_2 emission requirements, all of the coal gasifier systems include sulfur removal and recovery equipment downstream of the gasification reactor. Sufficient sulfur to meet the emission requirements is removed from the gas by cooling the gas

and passing it over sulfur scrubbing material at relatively low temperatures. The tars, heavy hydrocarbons, phenols, ammonia, ash, particulates, etc. that may be present in the effluent from the gasifier reactor are reduced to very low levels as the gas is cooled to meet the requirements of the sulfur removal system. Thus, the coal gas delivered to the gas turbine is essentially free of all heavy hydrocarbons, particulates, ash, and metals.

TABLE 7-5

PRIMARY CONSTITUENTS OF GASIFIED COAL

Type of Gas	Low BTU	Medium BTU	High BTU (SNG)
Heating Value (BTU/SCF)	100-250	300-800	900-1000
Composition (Vol % Dry)			
H_2	13-22	10-55	0-1
CO	9-22	4-56	0-.5
CH_4	0.5-10	0-75	90-99
N_2	35-60	0-1	0-2
CO_2	7-17	0-30	0-3

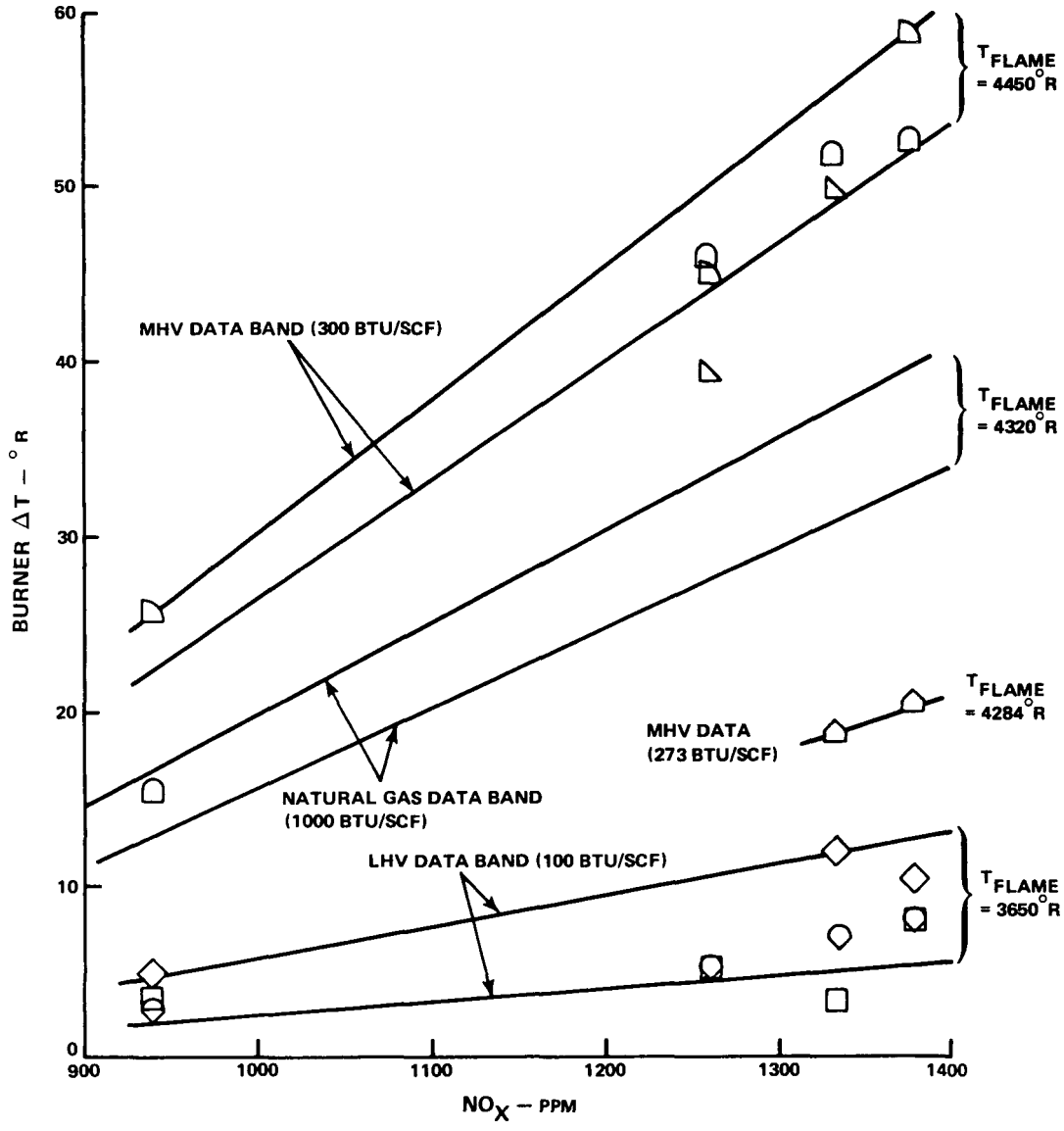


Figure 7-5. Impact of Gas Composition of NO_x

Modifications Required for Gasified Coal

A specific type of gasifier has not been selected for the HRGT power plant. The entrained flow type of gasifier is a prime candidate since it is at an advanced stage of development, produces few byproducts or waste materials, and has been studied extensively in related programs. The compositions of medium and low BTU gasified coal as produced by the Texaco entrained flow gasifier were selected as representative compositions of gasified coal. These compositions (Table 7-6) were used to estimate the modifications required and the impact on the gas turbine

reliability, availability, and cost of electricity. The change in gas composition produced by other gasification processes are not expected to have a significant effect.

TABLE 7-6

COMPOSITION OF MEDIUM AND LOW BTU GASIFIED COAL
REFERENCE AF 753

Fuel	Texaco Gasified Coal Medium BTU	Texaco Gasified Coal Low BTU
Gas Composition (Vol % Dry)		
H_2	35.7	11.7
CO	52.5	19.6
CO_2	10.8	7.7
N_2+A_R	1.0	60.9
CH_4	0.1	0.1
Gross Heating Value (BTU/SCF)	286	102

Operation on medium BTU gasified coal will require new low NOx combustor designs to satisfy emission requirements. EPRI program RP 1040 and RP 985 have shown that NOx levels can be reduced to the required levels by controlled mixing of the fuel gas and air to reduce the maximum flame temperature. This has been demonstrated on the pre-mix, lean burn type of combustor, shown in Figure 7-6. With this type of burner, the measured NOx (Figure 7-7) was lower than the design goal using medium BTU fuel gas supplied by a Texaco gasifier. The rich burn/ quick quench combustor configuration selected for the HRGT engine may also be applicable to low and medium BTU fuel gas. This configuration is optimum for minimizing both thermal and chemi-

cal NOx when operating on liquid fuels which may contain fuel bound nitrogen. Further analysis and tests are required to determine if the RB/QQ type of combustor would be preferable for coal gases. This combustor's unique capability for low NOx operation with nitrogen containing fuels may not be required in coal gas service because most nitrogen compounds are removed by the gasifier's clean-up system. Nonetheless, the fact that an RB/QQ combustor would be available because of its need for liquid fuel applications may mean that it would be the best design to modify for coal gas fuel applications. It reduces the NOx to a low level with liquid fuels and should do the same with gaseous fuels. The only modifications required to operate on medium BTU gasified coal will be to replace the liquid fuel injectors with a simple gaseous fuel nozzle, and enlarge the fuel manifold. Since this manifold is external to the engine case, the change is very simple and causes no structural problems. If there are sufficient installations to warrant the change, the RB/QQ burner could be simplified and converted to a pre-mixed/lean burn combustor by fabricating the combustor can without the rich burn section. If dual liquid and gaseous fuel capability is desired, it could probably be accomplished by injecting the gaseous fuel through both the fuel and air side of the air atomizing liquid fuel injector.

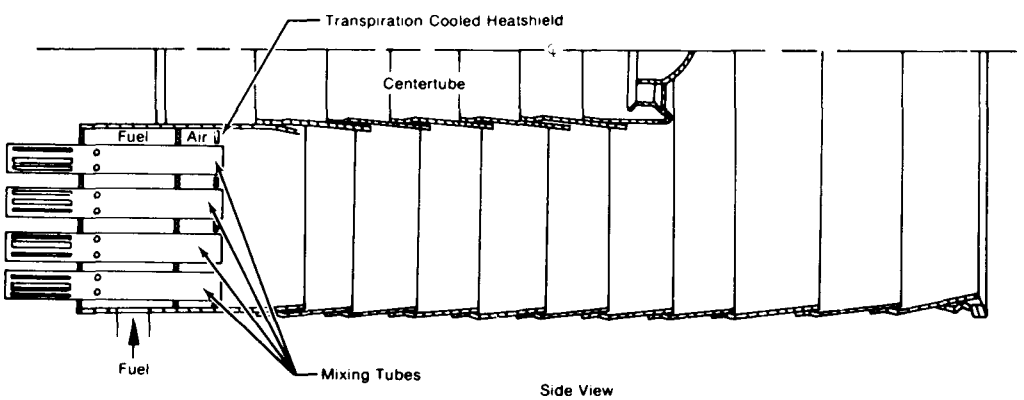


Figure 7-6. Premixed/Lean Burn Combustor

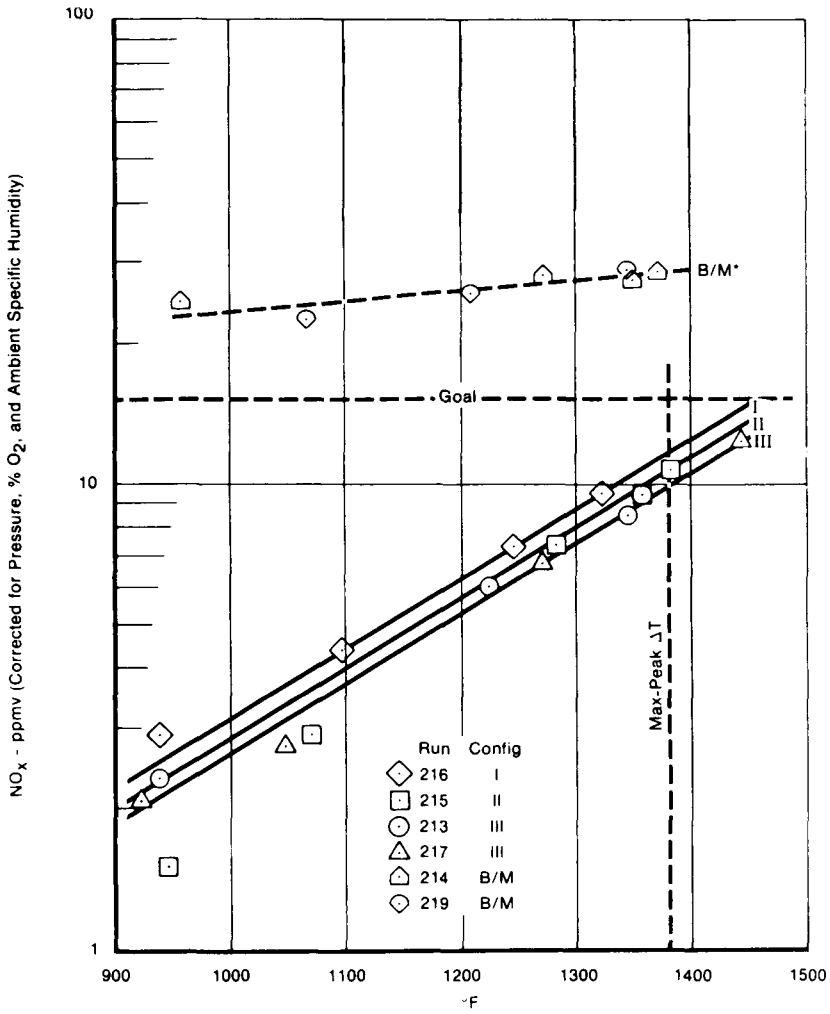


Figure 7-7. NOx Reduction with Pre-mix/Lean Burn Combustor

For both low and medium BTU gasified coal, the turbine ancillaries can be simplified by the elimination of the following components:

- Main Fuel Pump
- Safety Relief Valve
- Hydraulic Ball Valve
- Gear Pump
- Vacuum Pump
- Fuel Pretreatment and Additive Streams

Impact of Coal Derived Gaseous Fuels on Reliability

The review of the gasified coal characteristics indicates that the gas delivered to the gas turbine will be clean gas essentially free of heavy hydrocarbons, particulates, ash, and metals. The impact of gasified coal should be similar to that of natural gas. The use of gaseous fuel eliminates cooling problems in the fuel nozzles, reducing streaking and hot spots on the combustor liners and turbines. The erosion and corrosion of the turbines are virtually eliminated. Thus the use of gasified coal reduces the failure rate of the combustors and turbines.

The Reliability Prediction Model was used to estimate the impact of medium and low BTU gasified coal on the reliability and availability of the HRGT power plant. The reduced combustor and turbine failure rate increases the estimated MTBF of the HRGT engine from 8300 hours with the reference liquid fuel to 9800 hours with medium BTU gasified coal. With the already high reliability level of the combustor and turbine designs selected for the HRGT engine, the improvement with gaseous fuel is less than that experienced with natural gas in the present day gas turbines.

The lower maximum flame temperature of low BTU gasified coal further reduces the impact of hot spots and temperature distribution on the combustor and turbine. The reduced failure rate of the combustor and turbine increases the estimated MTBF of the engine to 10,900 hours.

For low BTU gasified coal a new burner designed to minimize NOx is not required to satisfy the emission requirements. Critical design requirements are combustion stability and turn-down capability when using the low energy fuel. It may be possible to use the RB/QQ combustor can and avoid the need for alternate combustors. This would have to be confirmed by analyses and test.

It is anticipated that the gasified coal will be supplied to the gas turbine at a sufficient pressure to eliminate the need for a compressor. The elimination of the liquid fuel pumps and associated pretreatment systems simplifies the fuel forwarding system and improves the reliability of the ancillary system. Thus, use of gasified coal improves the equivalent reliability and availability of the HRGT power plant, as shown in Table 7-7. The rates are virtually the same for low and medium BTU gasified coal. With gasified coal, the planned outage hours could be reduced by increasing the inspection interval. Since the inspection interval for the reference liquid fuel is one year, it was assumed that inspections would be kept on a yearly basis for gaseous fuels even though this interval could be increased.

TABLE 7-7

IMPACT OF COAL DERIVED GASEOUS FUELS ON HRGT RELIABILITY

	Reference	Gasified Coal	
Fuel	Liquid Fuel	Medium BTU	Low BTU
Gas Turbine			
MTBF (Hrs)	8300	9800	10,900
MCT (Hrs)	58	58	58
POH (Hrs)	230	230	230
Equiv. Reliability (%)	91.9	92.5	92.6
Equiv. Availability (%)	95.7	96.3	96.4

Studies have been made to estimate the cost of gasified coal for use as petrochemical feed stocks and as SNG. For electric power generation, the gasifier is integrated with the gas turbine and the cost of the gas is not estimated as a separate item. Since the coal gas cost depends upon the type of gasifier which has not been selected yet, it was beyond the scope of this study to estimate a projected cost of gasified coal corrected for integration credits. The cost of electricity from gasified coal was estimated parametrically as a function of fuel cost, as shown in Figure 7-8. For this estimate it was assumed that the gasified coal is supplied to the gas turbine at pressure. For medium BTU gas the turbine preformance is the same as with the reference liquid fuel. With low BTU gas, it was assumed that the air for the coal gasifier would be bled from the compressor of the HRGT engine. This offsets the effect of the higher mass flow through the turbine and the performance is the same as with liquid fuel. Based on these assumptions, at equal fuel cost ($/MM BTU) the cost of electricity is the same for low and medium BTU gasified coal. Studies including the gasifier, such as reported in reference (3), are required to establish the relative cost difference with coal derived liquid, medium BTU and low BTU gaseous fuels. Since the gasified coal

reduces the effects of erosion, corrosion and hot spots on the turbine and combustors, it would probably be cost effective to increase the turbine inlet temperature and reduce the gas turbine heat rate when operating on low or medium BTU gasified coal.

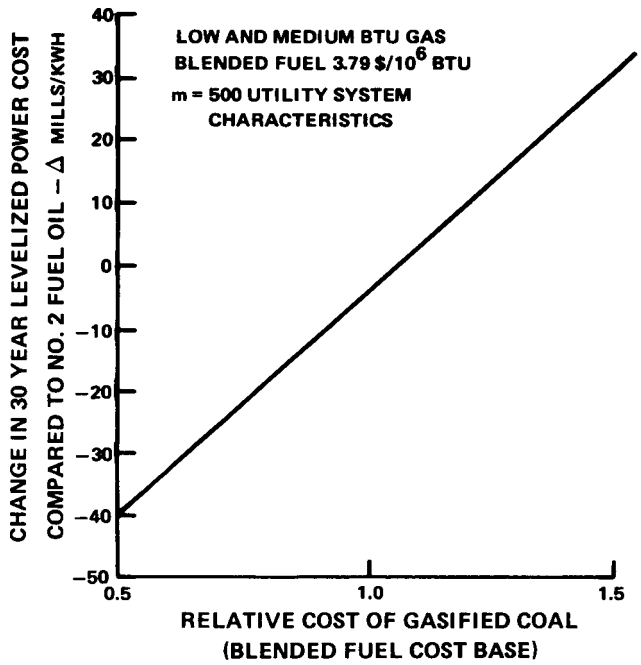


Figure 7-8. Impact of Coal Derived Gaseous Fuel on Plant Levelized Power Cost

REFERENCES

7-1. Gleason, C. C. and Bahr, P. W., "Experimental Clean Combustor Program Alternate Fuels Addendum Phase II Final Report," NASA-CR-134972, January 1976.

7-2. Chandra, K., et al., "Economic Studies of Coal Gasification Combined Cycle Systems for Electric Power Generation Final Report," EPRI AF-642, January 1978.

7-3. McElmurry, B. and Smelser, S., "Economic Studies of Coal Gasification Combined Cycle Systems for Electric Power Generation Final Report," EPRI AF-753, April 1978.

INTEGRATED ENGINE CONTROL

Introduction

The conceptual design for the HRGT engine control is an integrated assembly of devices such as controllers, monitors, synchronizers and sensors. Central to this design is a programmable, digital, solid state controller. Since the introduction of microprocessors to the commercial market, some 6 years ago, it has been shown that microprocessor based control systems can offer significant performance and integrity improvements over conventional approaches. The controller should be an "off-the-shelf", reliability-proven item with sufficient process and batch programming flexibility to cope with the engine control requirements. Since these systems can handle both sequence and analog control tasks, the sequencer and fuel governor are not treated seperately. This document describes the conceptual design of the Integrated Engine Control (IEC).

Focus on Reliability

The following summarizes the ideas and concepts in this control system design that focus on reliability.

Engine Controller

Present power station installations use an electronic controller for the fuel control, and relay or electronic equipment for the control of the station sequencing. A programmable, digital, controller replaces much of the mechanical or electronic hardware with software and eliminates communication between the two units. This results in less system hardware complexity, less noise susceptibility, and no mechanically or electrically driven parts. These features improve control reliability and reduce cost.

Inputs

The control system inputs include redundant sensors, except for functions which only activate an alarm. Control inputs are provided with two levels of alarm prior to shutdown. All analog control inputs have range checks with automatic switchover to the redundant input if an "out of range" or "rate of change limit" occurs. Other inputs alarm and hold at last valid value. Critical thermocouple inputs will

be in triplicate with a median select feature. Process control blocks can be set to manual for direct operator control of analog outputs.

Outputs

All digital outputs have a three position switch with an automatic, on, and off position for direct manual control of the output. This feature makes it possible to control the sequence in a manual mode. Switches in the control room panel are behind a cabinet door to prevent inadvertent actuation.

Processing Unit

There are two completely redundant Central Processing Units (CPU's). The CPU checks its operation periodically (arithmetic check) and transfers control to the redundant unit if any problem is detected. This condition is alarmed so the module can be repaired during normal operation.

Engine Protectives

The control system executes all engine protective parameters including alarms and system shutdown as necessary. However, there is a redundant and completely independent protective system that cannot be disabled. (Independent sensors, monitors, and fuel shutoff solenoid). This feature enables the operator to exercise manual control through the computer variables without danger of catastrophic engine failure.

Vibration Sensors and Monitors

Both vibration sensors and monitors are redundant at each location. An alarm and absolute high limit are designated for each. A shutdown condition results only if both monitors exceed the high limit. If only one monitor shows excess vibration a warning is flashed requiring immediate attention. If a monitor is suspected of false readings, it can be deactivated and the other monitor becomes a shutdown device. This approach prevents shutdowns caused by faulty vibration instruments and sensors.

Synchronization

Generator to bus frequency synchronization can be achieved manually or automatically. The synchronization module has all sensors and instrumentation required for manual operation.

Performance Display

The control system is responsible to measure input parameters and calculate standardized performance numbers that can be compared to initial performance. This will enable the operator to reach quick decisions about the condition of the engine. Performance parameters are to be displayed at all times, in digital format, and updated at least once per second.

Operating Display

All input and output parameter values are available to the operator, in engineering units at all times. Desired engine conditions are displayed in analog form on dials and gauges. All others are available at the system terminal in digital form.

Scope

The conceptual design approach defines the hardware, software and logic required for the control of the HRGT engine and associated ancillaries. The engineering and manufacturing requirements of a microprocessor based, solid state, programmable, sequence and process control device are described as well as the associated sensors, synchronizer and monitors. The primary control elements are:

- Engine Fuel Control — Liquid Fuel, Gasous Fuel, Dual Fuel
- Engine Bleed Control
- Compressor Inlet Guide Vane Control
- Sequence Control for Engine and Ancillaries
- Process Control for Engine and Ancillaries
- Performance and Status Display
- Alarms and Annunciators
- Engine Permissives
- Engine Protectives
- Line Frequency Synchronizer

The control of the generators, boilers, steam turbine and auxiliaries, or the electrical systems is not included in the Integrated Engine Control (IEC).

Control System Description

The IEC is an integrated assembly of devices required for the orderly and safe startup, stable operation, shutdown or trip of the HRGT. The system monitors and alarms all required engine parameters and display pertinent engine data.

The controller is a solid state, programmable, digital device, assembled of standard rack mounted cages and modular printed circuit boards. The IEC consists of all necessary connectors, input and output modules, memory and logic systems compatible with all other HRGT control equipment. All of the devices constituting the IEC are electrically interconnected to provide the most efficient integration of control equipment. Figure 7-9 illustrates this concept.

- Controller
  - Input/Output
  - Sequencer
  - Fuel Governor
  - Alarms and Annunciators
- Synchronizer
- Vibration Monitors
- Combustible Gas Monitor
- Independent Protective Control
- Station Instruments
- Control Keyboard
- Condition Monitor Panel

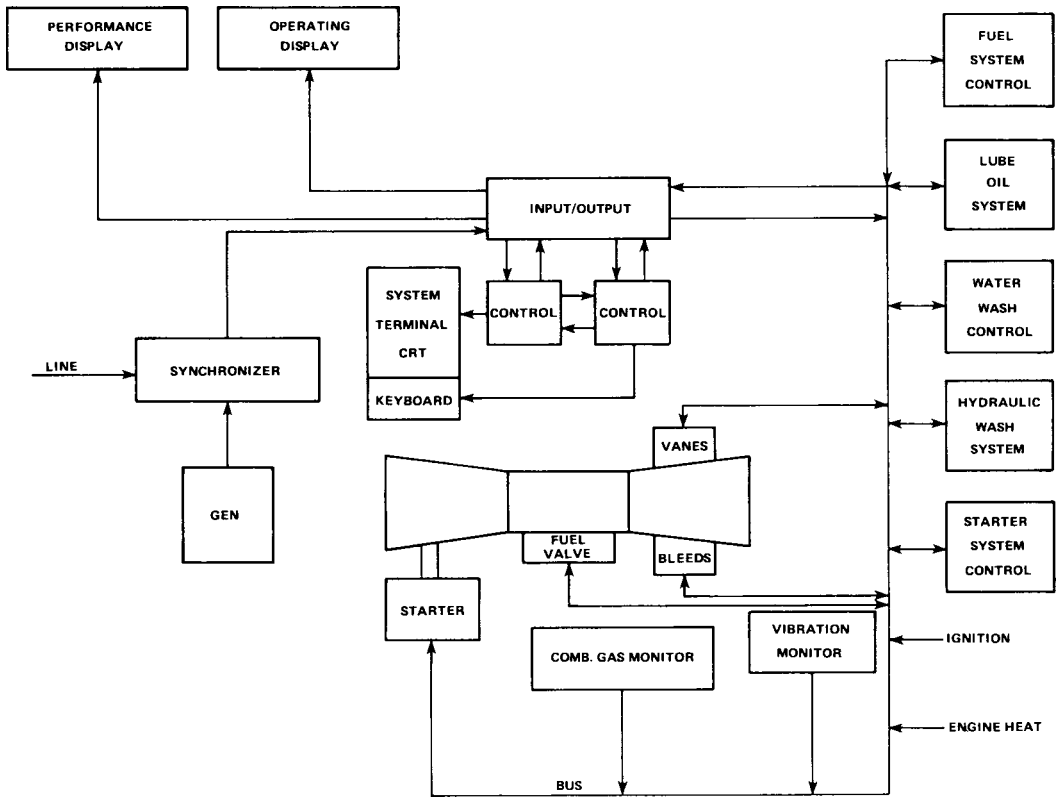


Figure 7-9. IEC Flow Diagram

Applicable Publications

The documents which are applicable to the Integrated Engine Control design include the following:

- JIC — Joint Industry Conference
- NEC — National Electric Code
- EIA — Electronics Industries Association
- USASI — USA Standards Institute
- IEEE — Institute of Electrical and Electronic Engineers
- NEMA — National Electrical Manufacturers Association
- UL — Underwriters Laboratories

General Design Approach

Except where otherwise specified the design, construction and performance of all equipment and software shall conform to the latest version of the applicable standards.

General Design

Materials and Processes

Materials and processes used in the assembly of the control system shall be of high industrial quality, shall not contribute to personnel hazard, and shall not be in conflict with the described use.

Protective Treatments and Coatings

Conformal coatings shall be applied on electrical circuits, where applicable, in accordance with generally accepted procedures. Parts shall be made of materials resistant to corrosion from salt, humidity, and airborne contaminants commonly encountered in electric generating utility plants, or suitable protective treatments shall be applied.

Environmental Conditions

The IEC need not be subject to environmental tests, such as vibration, fungus, sand and dust. However, all practical precautions against these conditions shall be implemented.

Ambient Temperature
(Except for outside equipment)

The IEC shall operate satisfactorily when placed in still surrounding air at temperatures ranging from -20°F to 120°F.

Application Software

The gas turbine design engineer most familiar with the process shall be responsible to learn the control system application software language and write the process

code. Therefore, no attempt to translate the process control to the control system manufacturer is required. This will avoid translation errors.

Identification

All major sub-assemblies shall be suitably identified with permanently attached name plates. All modules shall be permanently marked to enable identification for replacement, service and warranty.

Warmup Time

The IEC warmup time shall not be the limiting factor for station start-up.

Service Life

All equipment shall be designed for a 30 year service life when used under normal conditions.

Reliability

The IEC shall be designed so as to provide the highest possible degree of reliability. All critical sensors, instrumentation, and control circuits shall be fail-safe by redundancy. Manual control modes shall be available where possible.

Electrical Design

Power

The primary input power to the control system components shall be 120 VAC. In case of power failure, the system will continue operation on 125 VDC emergency power the system design shall be such that ±10% variation in input voltage will not adversely affect the process.

Load

The power dissipated by the IEC shall not exceed 2 kW.

Noise Immunity

Proven grounding and shielding methods shall be used to prevent internal inductive and capacitive coupling as well as to provide high common mode rejection throughout the control system. Internal circuitry shall be designed to be immune from all undersirable transients and noise. The system shall distinguish between a correct input signal and a spurious transient signal such as generated by reactive loads. All input and output signals shall be transformer isolated to meet factory mutual requirements for intrinsic safety and protect internal logic level devices from high voltage. (LED's are permissible only if above requirements is met). A common mode rejection ratio of 100 db at 60 Hz shall be provided. The IEC shall operate satisfactorily in the electromagnetic noise environment found in a power control house with electric generating equipment.

Connections and Terminations

All connectors to external wiring shall have a locking mechanism to maintain engagement of mating pieces. Connectors other than modular card must be keyed and marked to prevent mismatched connections.

To provide for easy rack removal, service and replacement, the modular cages shall be interconnected by connectors and flexible cables. The "wire wrap" method of making connections is not acceptable.

All input power terminations shall be made to circuit breakers within the IEC. The breakers shall be properly sized and UL approved.

Cooling

If it is necessary to use cooling devices such as fans or blowers, the devices shall operate on 120 VAC.

Wiring

All unit wiring shall be stranded and flexible. The wire type, size and insulation thickness shall be as defined by the applicable publications.

Self Testing

Self test features shall be incorporated and shall operate whenever the IEC is powered.

- Each analog input signal shall be tested by a range check for each program cycle.
- Analog input interface hardware shall be checked by a test signal for each program cycle.
- Each central processor unit shall be checked by an arithmetic exercise periodically.
- A means shall be provided to isolate bad memory cells.
- Each output register shall be checked by feedback to the related processor each time it is used.
- The total time for each processor to complete a program cycle shall be checked by independent means.
- The IEC shall be designed such that all circuits are failsafe, any fault shall switch the logic to a redundant circuit or put the output drivers in a state that prevents damage to the engine.

Instrument Range and Accuracy

The required range and accuracy for each display instrument is TBD.

Mechanical Design

Overall Dimension

No special restrictions apply to the overall dimensions of the control system.

Cages

Standard cages shall be used and designed for 19 inch rack mounting. The cages shall be removable from the IEC enclosure from the front. The cages shall be designed such that they can be moved up or down in the enclosure to facillitate additional equipment.

Modular Cards

Printed circuit card sizes shall be standardized. A minimum number of different kinds of cards shall be employed, each card shall be independently supported in the cage and shall be locked in place by convenient, manual release hardware.

Specific Design

Input/Output and Sensory

The primary control elements and input signals to the control system will be from the engine and ancillary equipment. The control system shall be compatible with these input signals and control requirements. Table 7-8 is a preliminary estimate and definition of the process I/D count. The control system shall have the capability to increase the number of each type of I/D signal by 50 percent.

TABLE 7-8
INPUT/OUTPUT LIST

Analog Inputs Include:

- Rotor Speed
- Ambient Temp
- EGT
- Liquid Fuel Flow
- Gas Fuel Flow
- Grid Frequency
- Gen. Frequency
- Output Power
- Desired Power
- Rotor Speed
- Sync. Signals

Analog Outputs Include:

- Liquid Fuel Valve Position
- Gas Fuel Valve Position
- Inlet Guide Vane Position
- Calc. Performance
- Operating Condition

Discrete Inputs Include:

- Control Signals
- Engine Stop
- Automatic
- Manual
- Engine Test
- Fuel (Gas/Liquid)
- Breaker (Closed/Open)
- Water Wash
- Ancillary Status

Protective Signals: (Analog or Discrete)

- Low Start Speed
- No Light Off
- Low Idle Speed
- Breather Pressure High
- Engine Lube Pressure Low
- Generator Temperature High
- General Lube Pressure Low
- General Bearing Temperature High
- Lube Filter Diff. Press. High
- Lube Oil Level Low
- Lube Oil Temp High
- AC Lube Pump
- DC Lube Pump
- DC Motor On
- DC Lube Pump Overload
- Vibration Monitor
- Chip Detector
- Loss Of Battery Charge
- Fire
- Fuel Pressure Low
- Fuel Filter Diff. Press. High
- Transformer Fault
- Emergency Lock Out
- Inlet Filter Problem

Discrete Outputs Include: (On/Off)

- Fuel Shutoff Valves
- Liquid Fuel Staging Valve
- Flow Divider Bypass Valve
- Station Bleeds
- Indicator Lights

Annunciators Include:

- Engine Incomplete Sequence Low Start Speed
- Engine Incomplete Sequence No Light Off
- Engine Incomplete Sequence Low Idle Speed
- Starter Duct Pressure High
- Engine Breather Pressure High
- Engine Lube Pressure Low
- Engine Enclosure Temp. High or Fan Off

Engine Heat Valve Open or Heat Off
Generator Incomplete Sequence
Rotor Ground or Exiter Overcurrent
Generator Exhaust or Exiter Air Temp. High
Generator Lube Pressure Low
Generator Bearing Temp. High
Lube Filter Diff. Press. High (Engine)
Lube Oil Level Low
Lube Oil Temp High
AC Lube Pump Trouble
Lube Pump Backup Timer Trouble
DC Lube Pump Power Off or Vapor Exit Off
DC Motor Running
DC Lube Pump Overloaded
Diode Failure
Voltage Regulator in Manual Position
Vibration Monitor
Chip Detector
Battery Ground or Voltage Low
Loss of Battery Charger AC or Control AC
Fire
Fire Valve Closed
Fuel Pressure Low
Fuel Filter Diff. Press. High
Lube Filter Diff. Press. High (Generator)
Transformer Or External Fault
Emergency Lock Out
Inlet Filter Trouble

Temperature Measurements

The engine and ancillaries will furnish shielded cables from ISA Type "K" thermocouples to the control system scanner input connectors. Overall accuracy of conversion from millivolts to digital value including cold junction compensation shall be ±2°F from 32°F to 530°F and ±00.5% of range from 530°F to 2300°F over the ambient temperature range from 32°F to 122°F. Positive open thermocouple detection is required.

Pulse Train Inputs

The control system will receive pulse train input signals for determining the engine rotor and generator speed. These pulse trains will be voltage signals of zero or 15 volts. The pulse rate will range from 0 to 70 Hz. Accuracy of the calculated speed shall be ±0.5% of full scale.

The engine rotor speed shall be determined at least 5 times per second for adequate overspeed detection in drop load condition.

Pressure Measurements

All pressure measurements will be provided as 4 - 20 ma signals. The control system shall have the capability to convert differential pressures from orifice or venturi meters into flow rates. Absolute and differential pressure signals shall be coverted directly into engineering units. Analog to digital conversion shall be at least 10 bit resolution.

Analog Outputs

All analog control output signals shall be 4 - 20 ma signals capable of driving into a 500Ω load. Resolution shall be at least 10 bits.

Digital Inputs

All digital inputs will be contact closures. The control system will furnish the power source and interface circuits required to sense the contact closure. To minimize radiated noise, the exitation voltage shall be less than 15 volts.

Digital Outputs

All digital outputs will be used to drive primary elements directly.

Type I	120 VAC 20 WATT MAX 41 VA HOLDING 78 VA INRUSH
Type II	120 VAC 7.5 WATT MAX 15 VA HOLDING 50 VA INRUSH

Output Protection

A fault occuring on any output shall not result in the loss of any other output of the control system. All digital outputs shall remain latched in their command position if the output driver power fails. Three position switches shall be provided for each digital output to enable manual operation.

Input Scan Rates

Input scan rates shall be selectable from 0.1 second to 1 minute.

Process Computer

Memory

The process computer memory shall be a solid state storage device with high reliability, fast access times and low power requirements. Both single bit error correction and multiple bit error detection techniques shall be employed. Single bit memory read errors shall be corrected, and multi bit errors shall cause a processor halt and transfer to redundant process computer.

Processor

The process computer CPU shall be a 16 bit, microprogrammed digital device which performs arithmetic and logical operations. The instruction set shall be stored in Read Only Memory (ROM). The basic execution cycle of the processor shall be not more than 0.1 sec.

System Security

The process computer shall include hardware and software to protect the system from:

Process Disturbances
Interface Equipment Failure
Operator Errors
Computer Malfunctions

System Software

The software is to consist of a computer process control system including both Direct Digital Control (DDC) and sequencer control. Supporting software shall be supplied for building the control data base, the sequencer and continuous control logic and the CRT displays. Diagnostics must be included for determing the cause of malfunctions.

Direct Digital Control (DDC)

Must implement control strategy of Figures 7-10 and 7-11. The Direct Digital Control system will consist, as a minimum, of the following control algorithm modules:

PID
Lead/Lag
Feed Forward
Ratio/Bias
Multiply/Divide
Add/Subtract

The output of any module must be usable as the input to any other module for forming control cascades.

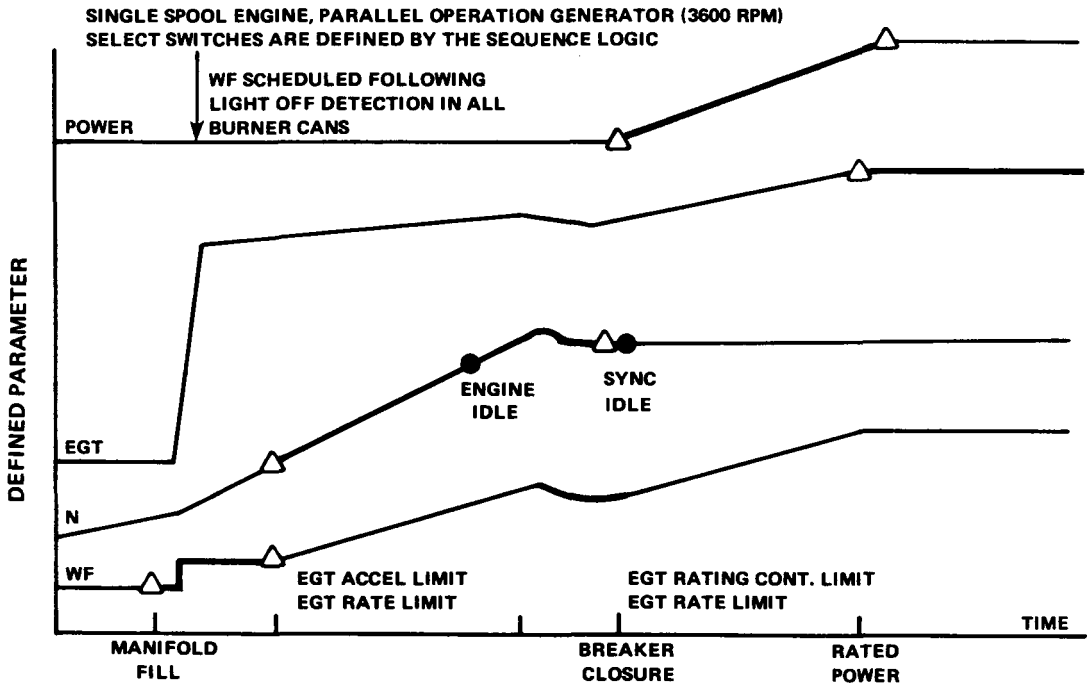


Figure 7-10. Direct Digital Control Defined Parameter vs. Time

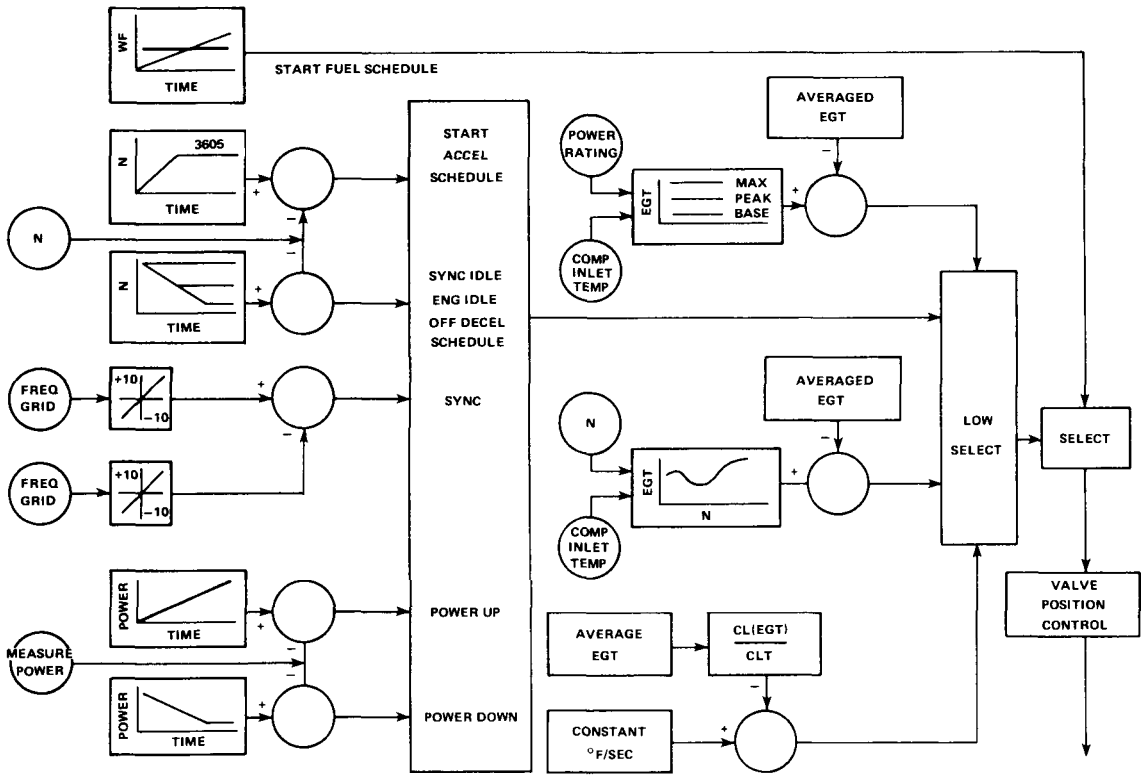


Figure 7-11. DDC Strategy

Alarm Limits

All analog inputs, whether used for control or only for process monitoring, have available four limits -- two high limits, two low limits and/or a deviation limit. All limits shall be entered in the data base in engineering units. The deviation limit is applied to the setpoint or stated desired value of the analog input. If no limit is entered at program generation, default values of zero and 100% shall be assigned.

Whenever any of the limits is exceeded, an alarm message activating the audible alarm is sent to the operator's console and to the logging device and the audible alarm is activated. Similarly, whenever an out-of-limits measurement returns within limits, a message shall be displayed on the operator's console and logged on the logging device, and the audible alarm silenced. Each alarm limit has an associated deadband to avoid repeated messages caused by a noisy signal from a measurement near one of the limits. Each alarm limit is capable of being inhibited in designated system states by the sequence control system.

Every alarm sent to the operator's CRT continues to blink until it is acknowledged by the operator. After acknowledgement, an alarm is displayed without blinking and the audible alarm is silenced. The operator is able to see all points currently in alarm by displaying a fixed format summary on the CRT. The system must accommodate up to 50 simultaneous points in alarm. The first page of alarm data must be available through a single function key on the CRT. Successive pages of alarm data must be available with a simple procedure to roll-up or roll-down the display pages.

The purpose of the second higher and lower limits is to provide sequence logic branching in addition to annunciation.

The engineer's CRT has the capability to display and then to modify any alarm limit any time the system is operating. Similarly, the engineer has the capability to display and then to modify any setpoint at any time. It will be the engineer's responsibility not to modify setpoints if the sequential control software is in control of a setpoint.

Data Base Generation

The preferred method for generating the DDC data base is an interactive system which prompts the control engineer for all data required to define each DDC control block. Initial data base generation may alternatively be done from a source provided on DDC. Each variable in every data block is assigned default values, unless defined by the process engineer.

The DDC software package contains a utility program which will display, modify and delete control algorithm blocks in an existing DDC data base. This utility is usable from a CRT and displays all entries for a control block in engineering units. It also generates hardcopy reports of each control block in English and in engineering units on the system printer. The user of this utility is able to obtain single reports or to specify a series of control blocks to be printed.

Analog Input Control

DDC package software processes the output of the analog-to-digital converter, applying any corrections required to account for amplifier gain or bias variations.

The analog signal shall be checked for validity, and if it is found to be invalid, e.g., an open thermocouple, an alarm shall be displayed or a logic branch taken at the control of the systems programmer.

All valid measurements will be checked for high, low and deviation alarms, and the appropriate flags will be set or cleared in the control data base for each point.

Update Frequency

Each analog input and each control output shall be scanned or updated at its own assigned frequency. Analog input and control output for some control loops shall be updated at least five times every second. This rate shall be capable of being reduced to a slower fixed period under software control. All other analog I/O points in the system shall have available scan periods ranging from 0.2 seconds to one minute. It will be UTC's responsibility to assign scan rates consistent with the throughput capability of the analog multiplexers.

Sequential Control

A sequential control package shall be provided which will allow UTC to create software tasks which will startup, shutdown and monitor normal operations of the Demonstrator. The sequential control package shall provide a higher level language with the verbs of the language being easily related to the corresponding function.

As a minimum, the following functions shall be provided:

- Place final control outputs on or off control
- Open or close a cascade control loop
- Issue setpoint to a control loop or to a control output
- Obtain current analog input value and check its validity
- Obtain current digital input status
- Add, subtract, multiply, divide. If done with fixed point arithmetic, an overflow or underflow flag shall be provided.
- Test the state of a bit or contact input
- Set or clear a digital output, (i.e., start/stop or open/close a binary control device).

- Logical functions - AND, OR, NOT
- Comparison - arithmetic and logical
- Time delays
- GO TO statement to control logical execution software

Each sequential control program shall have a COMMON Block by which it can communicate information to other control programs. It shall also have a RECIPE area in which mathematical parameters can be stored. The process engineer shall be able to redefine and load the recipe for any control program without having to reassemble or recompile the program.

Displays

A number of standard displays are required which can be used by the operator or process engineer to observe and/or control real-time values in the system. While the format of each display is fixed, the user shall be able to select and change the specified process variables assigned to a particular display at any time. The data on any page being displayed shall be updated at the same rate it is updated for the control system or once per second whichever is slower.

Process Operator Annunciator Display

This display shall list each annunciator point which is in alarm in the order in which the alarms occur. The list will contain the annunciator point name and the time at which the alarm occurred. Each annunciator alarm will blink until it is acknowledged by the operator. The display will show the current time of day updated at least every five seconds.

There shall be at least 75 annunciator points assigned to analog and digital inputs. If more than one page is required to display the alarms, each page used shall indicate whether or not there are additional alarms displayed on a succeeding page.

Process Functional Requirement

The following requirements must be satisfied by the application software. The system software must provide the control blocks and batch process tools to implement the process functions.

Sequence Control

The sequencer must provide for an orderly, stable progression of ancillary operation and control loop activation in the fuel governor during startup, normal operation, shutdown or emergency shutdown, a normal sequence is as follows:

- Start ancillaries
- Check permissives for startup
- Engine startup
- Constant fuel for ignition
- Ignition
- Closed loop acceleration on rotor speed (speed vs time schedule)
- Closed loop on rate of change of rotor speed
- Closed loop on power for loading and/or unloading
- Closed loop on EGT for rated power
- Closed loop on rotor speed for cooldown
- Closed loop on min fuel flow to prevent flame out during drop load conditions.

Fuel Governor

The fuel governor provides the logic to control the fuel valve under all conditions. It assures constant fuel for ignition, controls start acceleration, synchronization, loading, unloading, cooldown and shutdown.

The sequential logic activates and deactivates the following control loops in the fuel governor:

Light-Off Flow

This is a fixed fuel flow, determined by a mechanical adjustment of the metering valve. After sensing flame in all burners, the light-off sensing circuit will allow the engine to accelerate.

Maximum Fuel Flow Limiter

The maximum fuel flow schedule controls fuel flow if the scheduled limit is reached at any operating state. The schedule is variable and a function of rotor speed and inlet air temperature. It performs fuel controlling function by acting as a variable under limit on the governor output.

EGT Accelerated Limit Control Loop

This loop limits the EGT to protect the hot section.

Power Control Loop

The power control loop operates on the error between desired power and output power.

Synchronization

During acceleration to synchronous idle, the rotor speed (generator frequency) is compared to the grid frequency. Close to synchronous speed, a synchronizing signal will match speed and phase to breaker closure.

Manual Mode

The manual control mode provides manual control of rotor speed up to engine trip. The limiting loops will override the manual control.

EGT Rating Limit Control Loop

This loop protects engine life and controls max power by setting a safe limit on the EGT during power operation.

Engine Loading Rate Limit

The engine loading rate limiter prevents excessive loading rates.

Minimum Fuel Flow Rate

The minimum fuel flow limit determines safe limits during deceleration to protect engine from flame out.

Dual Fuel Control

The fuel governor, when directed from the keyboard, at any time the engine is above idle speed, provides stable transitions from one type fuel to another or simultaneous burning of both, without fluctuations in output power or speed.

Station Bleed Control

The fuel governor will provide an on/off signal to the bleed valve actuator.

Inlet Guide Vane Control

The fuel governor provides an analog output signal for the proportional control of the inlet guide vane position.

Engine Protection

The control system assures engine protection by a permissive and two independent protective controls. If the permissives for a sequential step cannot be satisfied, an "Incomplete Sequence" shut down results or the condition is annunciated until corrected. The controller protectives are normally armed but can be bypassed by operator action. The independent protectives will lead to engine trip if predetermined setpoints are exceeded.

Engine Permissives

The engine permissives are operational in the sequential control. If a permissive is not met, the sequence will stop and a shutdown is initiated. The exact details of the permissive structures must be worked out when the characteristics of the engine are available.

Engine Protectives

The protection circuits receive analog signals directly from primary sensors. These circuits are primarily digital in operation and alarm and trip levels are adjustable from the control panel. Each fuel shutdown valve can be manually tested. The engine protective circuits will provide the following trip functions:

Bearing Temperature

Three temperature sensors at each bearing are required. Temperatures must exceed a preset limit for engine trip, one high indication will be annunciated.

Engine Exhaust Gas Temperature

Three temperature sensors for each burner can are required. The high and low T/C will be neglected. The T/C's are individually alarmed. If the median signal at any sensor location reads a prescribed setting above the alarm point, a shutdown is initiated after a set time delay.

Burner Can Temperature

Three temperature sensors on each can are required. The burner can T/C's are used to verify ignition and constitute the permissive signal to the fuel governor to start fuel flow. If ignition is not confirmed within a predetermined time, an incomplete sequence signal will cause a shutdown. Subsequently, burner can temperatures are compared. A "flame out" condition is annunciated and an engine trip scheduled.

Rotor Speed

Any rotor overspeed causes an engine trip.

Vibrations

All vibration sensors and monitors are redundant. The input to the control system from the vibration monitors are digital in nature. If both monitors from one location indicate excess vibration, the engine trips. One normal and one high input is an alarm condition only. Failure of a vibration sensor initiates an alarm. Failure of the monitor initiates a shutdown.

Lube Oil

The lube oil pressure is monitored at the engine interface with a pressure switch, a loss of oil pressure causes engine trip, also, the lube oil temperature at each bearing exit is measured, alarmed and causes engine trip if an excess condition exists.

Turbine Cooling Air

The loss of turbine cooling air will cause an alarm. This alarm must be acknowledged and loss of cooling air trip disabled within a predetermined time or an engine trip will occur.

Generator Temperature

Both generator bearings and stator temperature are monitored, annunciated and can lead to engine trip.

Independent Protective System

The independent protective system requires a controller, sensors and redundant fuel shutoff solenoids. The IEC does not limit or control this system in any way. All trip points are set to values higher than for the IEC protectives. This system must be manually reset if tripped by and of the following functions:

- Loss of Flame
- Rotor Overspeed
- Excess Vibration
- Turbine Overtemperature
- Loss of Lube Oil
- Loss of Turbine Cooling Air
- Bearing Overtemperature of Engine
- Generator Bearing Overtemperature
- Generator Stator Overtemperature

Instruments

All input signals to the IEC are available at the system terminal in engineering units and digital format.

Displayed Information

Important engine operating parameters will also be displayed at the instrument panel for manual operation and condition monitoring. The update frequency of the instruments will be not more than once per second. The following instruments are required.

- Rotor Speed
- Engine Temperatues
- Generator Temperatures
- Engine Pressures
- Fuel Flow
- Output Power
- Vars
- Current
- Frequency
- No, or Trips
- Running Time
- Clock
- Engine Status
    - Lock Out
    - Start on
    - Shut Down
- Sychonizing Scope Lights and Switches
- Number of Starts

Engine Performance Monitor

Engine performance data will be calculated by the control system from existing control inputs. This standardized data will be displayed in digital form.

Control Mode Switches

These switches provide direct input to the control system for the selection of various operating modes. The switches are front panel mounted, momentary contact devices with associated indicator for status confirmation. The following switches are required.

Automatic Operation

Allows automatic operation from start to load limit.

Manual Operation

This mode allows engine to accelerate to synchronous speed after start is initiated. Synchronization and loading are under the control of the operator.

Engine Test

Allows the engine to sequence to engine idle after start is initiated. Further operation is under operator control on rotor speed only.

Parallel Operation

This is the normal mode of operation and requires a live bus before breaker closing.

Gas/Liquid

This switch selects the gaseous or liquid fuel option and can be operated during or before engine operation.

Start Lockout

This is a two position key switch and acts as a permissive to an engine start. This permissive is operative prior to start only.

Engine Start/Stop

This pair of logically exclusive switches initiates a engine start if the unit is shut down or a normal shutdown, if the unit is in operation.

Engine Trip

This large switch initiates an immediate, emergency shutdown. A restart is prevented while the engine is in the tripped condition.

Synchronizer

The synchronizer will contain the generator and bus voltage matching and checking circuitry as well as the phase and slip frequency checking circuits. The synchronizer can be both automatically or manually operated it will receive an activation signal from the sequencer and will send appropriate signals to the fuel governor. When synchronization is achieved, a signal activates breaker closure.

Monitors

Vibration Monitor

The vibration monitor will recieve signals directly from four accelerometer pickups mounted on the engine and two accelerometer pickups mounted on the generator. The monitor will supply an identifying alarm signal to the annunciator if the vibration level from any pickup exceeds a preset limit. It will supply a discrete trip input to the sequencer if a higher, preset limit is exceeded.

Combustible Vapor Monitor

The combustible vapor monitor will receive signals directly from sensing heads mounted in the engine enclosure. The monitor will supply a discrete input to the annunciator if the concentration of combustible vapor detected by any sensing head exceeds a preset limit. It will supply a discrete trip input to the sequencer if a higher preset limit is exceeded. The concentrations, expressed as a percentage of the lower flammable limit, will be indicated on a channel select panel meter on the monitor.

ALTERNATE SERVICE

The HRGT design is intended principally for base/intermediate load service in a combined cycle plant. However, in order to provide more universal application for general use as an electric utility prime mover, the engine could be employed for peaking applications as well.

As part of the Phase I conceptual design work, two possible avenues were explored for adapting the engine to peaking service. One of these was the use of firing temperature re-rating, and the other was the addition of a supercharging compressor stage. It was concluded that with either modification. the HRGT design could be well suited for peaking service as a simple cycle alternative to combined cycle base load service.

Firing Temperature Re-rating

Peaking service is usually defined as utilization of 1500 hours or less per year. By contrast, a baseload machine can be expected to operate approximately 6000 hours per year, with an average utilization of 70 percent. With this type of expected operation as a baseload machine, the HRGT design was predicated on ensuring that no maintenance action would be required before 9000 hours accumulated operation. In other words, a planned outage would occur only about once a year.

If the principal of annual maintenance intervals is preserved, then an engine in peaking service can endure more demanding duty while in operation. Specifically, the limiting factor for the engine design is the coating life of the turbine airfoils. The time required to recoat and refurbish blades and vanes, in turn, establishes the rated firing temperature. In the case of increased firing temperature, which reduces coating life, the elapsed calendar time before incipient parts failure can be held constant by an equivalent reduction in service hours, which would be the case for peaking service.

An example of the effects of re-rating is shown in Figure 7-12. The figure shows that power output increasees and simple cycle heat rate improves as firing temperature is allowed to increase with fixed annual life for the exposed turbine parts. For example, at about 2000 hours, power output could increase to around 140 MW.

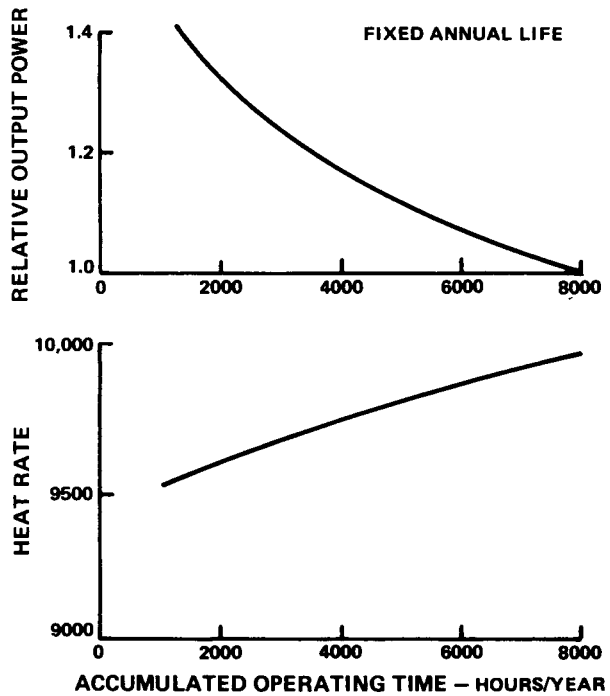


Figure 7-12 Effect of Re-rating on Combined Cycle Performance

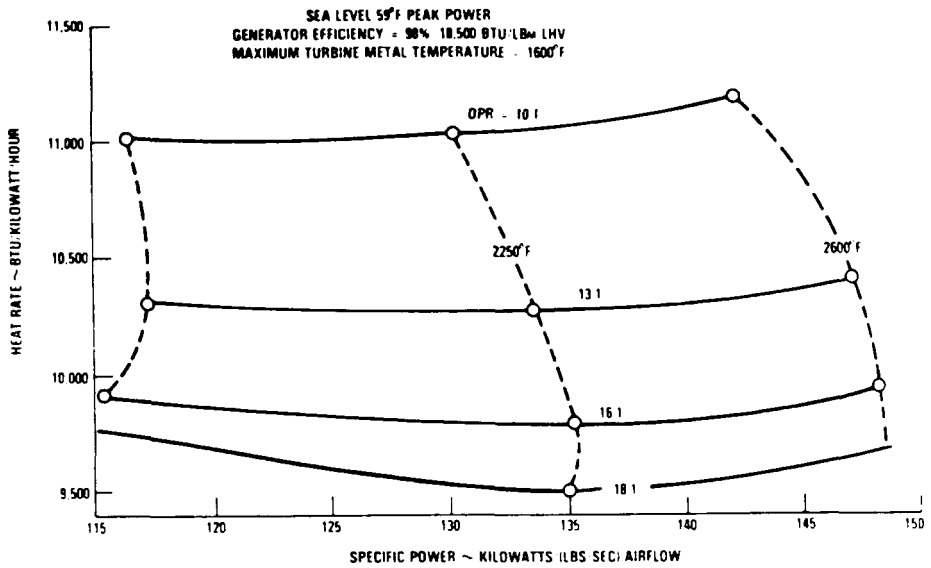
Compressor Supercharging

In the combined cycle powerplant with waste heat recovery, there tends to be an optimum pressure ratio, as shown in Figure 7-13. However, when the plant is operated as a simple cycle, the performance tends to improve as the pressure ratio is increased. Consequently, the addition of a supercharging stage to the front of the compressor could be beneficial if the gas turbine is to be used for peaking service. Supercharging has the additional benefit of increasing the engine flow which, in turn, increases the power turbine output.

The effect of supercharging on overall operating characteristics has been estimated, and the results are shown in Table 7-9.

The conceptual design studies completed to date indicate that the addition of the extra compressor stage is mechanically feasible. There appears to be adequate rotor shaft critical speed margin to absorb the extra length between bearings, and adequate surge margin for starting and part power operation can be provided using variable inlet guide vanes. The turbine and combustor designs

appear sufficiently durable to withstanding the added internal pressure and flow rates. However, more detailed analysis would be required to verify these conclusions.



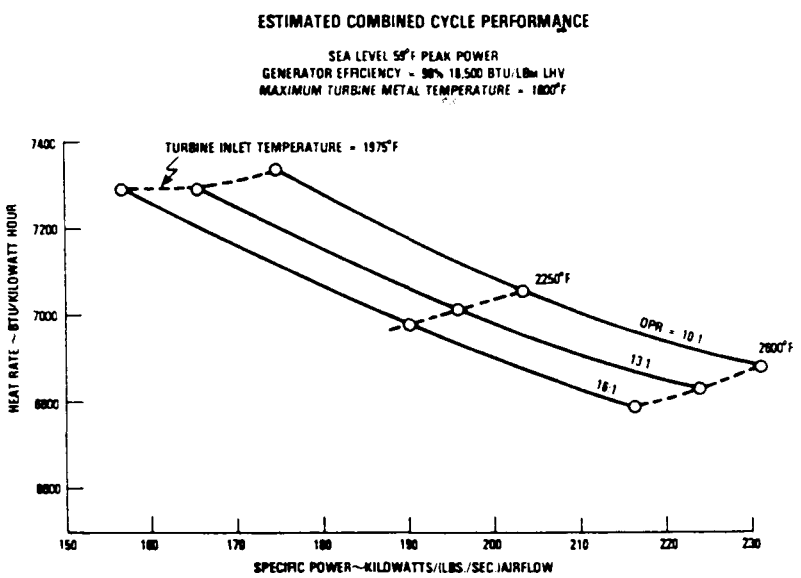


Figure 7-13 Combined and Simple Cycle Performance

TABLE 7-9. EFFECT OF SUPERCHARGING ON PEAK RATING PERFORMANCE

Application	Peaking	Base
Firing Temp. - (comb. exit)	2,250°F	2,150°F
Pressure Ratio	16:1	14:1
Airflow - (LB/sec)	925	840
Heat Rate - (BTU/KW-HR-LHV)	9,520	9,960
Power Output	132 MW	108 MW

(ISO conditions, installed)

CONDITION MONITORING SYSTEM

An important design feature in the HRGT-CC is the use of an on-line condition monitoring and diagnostic system. This system is independent of the control system, but provides necessary information about the performance of each major component in the plant. The objective is to improve plant availability by reducing routine scheduled maintenance and performing maintenance only on an as-required basis using condition monitoring as the tool to diagnose when maintenance is required.

In addition to diagnosing problems and predicting when maintenance is needed, the condition monitoring system also provides the following:

- Status printout of current plant performance and conditions when demanded by the operator.
- Instantaneous notification when any alarm limit is exceeded.
- History file storage of selected critical parameters used to generate diagnostic and prognostic maintenance messages.

General Description

In the Phase I study, Hamilton Test Systems Division, in conjunction with PSD, conceptualized a version of the Hamilton Test System TRENDS® system for the overall combined cycle plant, as shown in Figure 7-14.

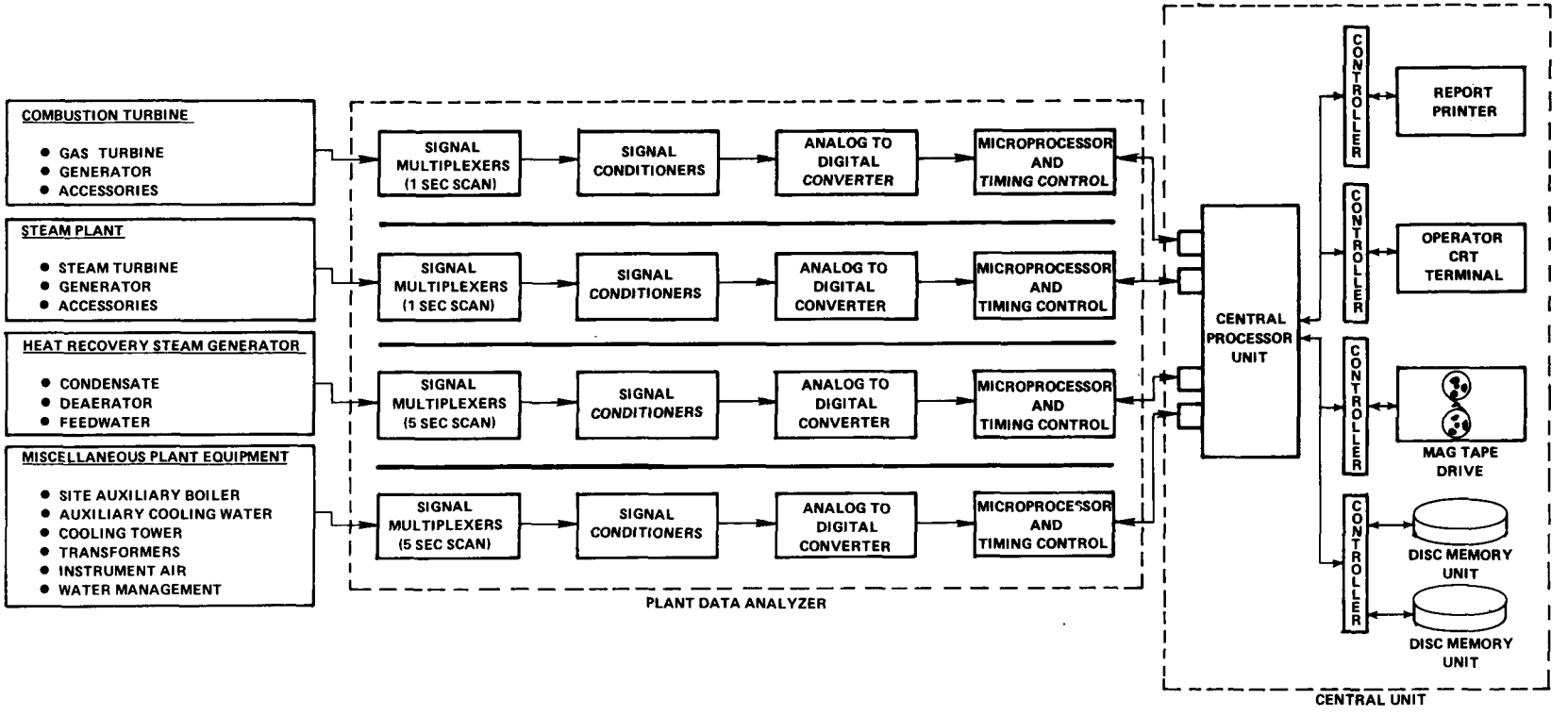


Figure 7-14. Combined Cycle Power Plant Condition Monitoring System Diagram

The TRENDS® system accepts sensor inputs from the following plant areas:

- Combustion Turbine and Auxiliaries
- Steam Plant
- Heat Recovery Steam Generator
- Miscellaneous Plant Equipment

These inputs are passed into the Plant Data Analyzer. The Plant Data Analyzer consists of four independent sections, one for each of the four areas of the plant. Each section consists of individual modules designed such that they can be easily interchanged or replaced as required. These modules consist of signal multiplexers, signal conditioners, analog to digital converters, and microprocessor and timing control units. Each independent section of the data analyzer has an estimated MTBF of 25,000 hours.

The data sampling or scan rate for the gas turbine and steam plant is typically set at 1 second. The scan rate for the heat recovery steam generator and other miscellaneous plant equipment is typically set at 5 seconds. These rates were selected based on Hamilton Test Systems' experience in the field of diagnostic data systems, but can be adjusted.

The microprocessor and timing control unit determine the mode of operation (start-up, alarm, etc.) averages data, checks alarm limits and transmits data to the Central Processor Unit when required. Typical turbine modes checked are startup, shutdown, stable running condition, alarm and trip.

In addition, the Plant Data Analyzer contains extensive self health signals. This data is transmitted to the Central Processor Unit. The Central Unit checks the validity of these signals; and either accepts the data if the signal is correct, or if the signal is incorrect prints out the appropriate diagnostic message pinpointing the faulty module or sensor/cable and does not allow its use in any diagnostic logic.

The Central Unit (CU) polls the four sections of the Plant Data Analyzer. It obtains data from these units, and processes this data using a priority system. The Central Unit converts the applicable data received from the Plant Data Analyzer into engineering units, processes through diagnostic routines, and stores pertinent data in the history files on the discs and the backup magnetic tape.

Formatted messages are then displayed to the operator via the CRT and Line Printer. In addition, the CU then transmits appropriate reports to the Line Printer.

These reports will provide the operator with an accurate detailed picture of his overall plant performance and reliability along with all the pertinent data of the plant subsystems.

This system will monitor and diagnose not only operation of the gas turbines, but the gas turbine auxiliaries, the steam turbine, each of the generators, each of the heat recovery steam generators, boiler feed pumps, condensers, transformers, cooling towers and important auxiliaries in the steam plant. In the conceptualized system, over 600 parameters are either sensed or calculated for monitoring diagnostics and prognostics.

Prognostic Techniques

The prognostic techniques, using the processing power and memory of the Central Unit to compile trend history files of parameter deviations from baselines to determine equipment degradations and associated rates of degradation, can accurately predict when in the future a particular unit should be taken out of service for maintenance to avoid serious problems or failures as shown in Figure 7-15. Some of the areas of evaluation are:

- Gas Turbine and Generator
    - Compressor/Turbine Efficiency
    - Starting Reliability
    - Operating Reliability
    - Availability
    - Blocked/Streaky Nozzles or Damaged Combustor
    - Fouled Turbines or Worn Seals
    - Compressor Blades
    - Turbine Blades
    - Bearings
    - Simple Cycle Heat Rate
    - Generator Overheating
    - Lube Oil Cooler Performance
- Steam Turbine and Generator
    - Plant Efficiency
    - Availability
    - Turbine Efficiency
    - HRSG Performance
    - Gland Seal Condenser Duty

- Turbine Vibration
- Condenser Performance
- Generator Overheating
- Lube Oil Cooler Performance
- Cooling Tower Performance
- Feedwater Pump Efficiency
- Deaerator Duty

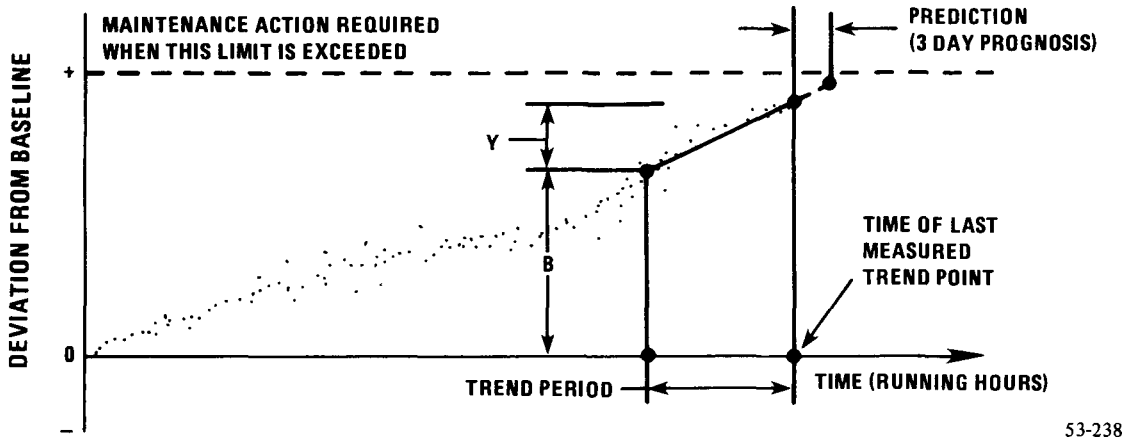


Figure 7-15. Condition Monitoring and Diagnostic System Predicts Need for Maintenance Action

Prognostics on the gas turbine will be based on Gas Path Analysis developed at Hamilton Test Systems.

Gas Path Analysis (GPA) is a patented analytic technique which takes measured gas turbine parameters (speed, temperature, pressure) and accurately calculates important degradation in non-measurable gas turbine performance parameters (compressor and turbine efficiency, turbine inlet temperature, nozzle area change, etc.).

The degradation in performance parameters is trended over a long term, and a prognosis made based upon the latest rate-of-change of the performance parameters. A prognostic message then advises the operator of the "time-to-go" before the parameter is expected to go "out of tolerance."

The parameters to be used as a basis for judgment of the relative gas path health are the measurable parameters of corrected rotor speed, fuel flow, output power, compressor pressure ratio and compressor and turbine discharge temperatures.

Differences in these parameters from their expected values can be used to determine which elements of the gas path have undergone change, or departed from their initial or expected condition. It must be emphasized that any measurable parameter in itself is not necessarily a direct indicator of faults in any particular gas path element. For example, at any given megawatt output, a change in compressor discharge pressure does not necessarily mean that there is a compressor fault. The change may be due to a combined compressor and turbine fault, or due to a turbine fault alone.

The primary independent, non-measurable performance parameters of the gas path are the compressor air pumping capacity and efficiency, (including the effects of bleed and IGV), the burner performance, the turbine stator nozzle areas, and rotor expansion efficiency.

Fundamentally, the parameters being measured are dependent variables whose levels depend on the absolute levels of all primary independent (performance) engine variables; changes in these primary but unmeasurable independent variables result in changes in the measurable dependent parameters. Therefore, the diagnostic logic interrelates changes in the measured dependent variables to changes in the independent unmeasurable performance parameters, whether they occur singly or in any random simultaneous multiple fault combination.

The prognostic outputs are produced by analysis of previous engine trend data. The central unit performs the prognosis by regressing the trend (or history) file. The regression of data is accomplished by a linear least squares - fit through 12 data points representing the most recent 72 hours of steady-state engine operation. Each of the data points, representing 12 steady-state half hours, is smoothed and filtered (conditioned) prior to history file entry. The straight line fit is extrapolated to predict a point value 72 engine operation hours in the future. When the extrapolated line exceeds a limit value for the specific parameter, a pertinent message is printed out. The selected number of data points and the time span covered are based on Hamilton Test Systems extensive past experience in the field of diagnostic data systems, but can be readily adjusted. HTS experience indicates that, except for slow degradation and wear of engine components, the latest 72 hours of steady-state operating data are most meaningful for diagnostic and prognostic purposes, and that older data becomes less meaningful.

The history file is therefore structured to place emphasis on the last 24 to 36 points of operation, older data being successively compressed and stored in the history file. The original base value of the new engine is always maintained in the file and slow degradation is detected by comparison.

The history file contains up to 54 points which corresponds to over 2000 hours of steady-state operation and is available as a printout upon operator request. The total file contains a structure of this type for selected measured parameters and the independent gas path parameter values.

The techniques are easily extended to the steam portion of the power plant and other miscellaneous plant equipment. The system constantly evaluates the performance of the steam plant through the process of automatically monitoring and analyzing the input data.

A maintenance advisory is generated when a degradation in performance is detected and diagnosed. Warning messages are automatically generated when an alarm limit is exceeded. Appropriate history files for steam plant data are maintained and available as a printout upon operator request.

Prognostic outputs are generated by analysis of steam plant trend data. These outputs will advise the operator of the "time to go" before the parameter is expected to go "out of tolerance."

In summary, the system is an automatic real-time condition monitoring diagnostic system that serves as a maintenance tool for the combined cycle power plant. It enables the user to detect an impending loss of performance or malfunction, perform a diagnosis, provide a prognosis and recommend corrective action. The system does not depend on, interfere with, or affect the operation of the Combined Cycle System.

The estimated MTBF of the Plant Data analyzer is 25,000 hours exclusive of sensors or the Central Unit Computer. The configuration and power requirements for the various system equipment is presented in Figure 7-16.

EQUIPMENT	INPUT POWER (WATTS)	WIDTH (INCHES)	DEPTH (INCHES)	HEIGHT (INCHES)	WEIGHT (POUNDS)
PLANT DATA ANALYZER	2000	21	31	72	300
CENTRAL CABINET 1	656	21	31	72	376
CENTRAL CABINET 2	1135	21	31	72	425
OPERATOR CRT TERMINAL	300	18	14.25	14.5	41
REPORT PRINTER	337	33	26	44	340

Figure 7-16. Condition Monitoring System Equipment Characteristics

SECTION 8.0

PRIOR DESIGN BACKGROUND (SINGLE SHAFT BASE LOAD)

INTRODUCTION

United Technologies had completed the conceptual design designated Single Shaft Base Load prior to undertaking the High Reliability Gas Turbine (HRGT) Phase I program. Much of the Single Shaft Base Load design background was considered relevant to the HRGT program. Consequently, this background was contributed as a cost-sharing element to the program and was used extensively during the course of the Phase I effort. Indeed, some of the basic elements of the Single Shaft Base Load are reflected in the final HRGT design.

As indicated, especially in Section 4 of this report, the Single Shaft Base Load served as the basis for the original reference design used in the early trade-off studies. Additionally, the comprehensive Single Shaft Base Load design background served to guide the mechanical layout and structural analysis conducted during the latter course of Phase I as the final HRGT design evolved. Thus, it is appropriate that a summary of the previous Single Shaft Base Load design history be included in this Phase I report in the interest of completeness. The Single Shaft Base Load historical background comprises the material presented in this section.

SUMMARY

The Single Shaft Base Load gas turbine (SSBL) conceptual design is the culmination of a series of evolutionary single shaft design studies. These studies funded by United Technologies were carried out over a number of years dating back to the early 1970's. A number of summary reports were written in 1972 and 1974 regarding these single shaft design studies. These reports provide the technical basis supporting the various mechanical and performance features of the SSBL conceptual design. Pertinent sections of these reports were condensed and are contained in the References at the end of this section. These condensed reports are referenced where applicable in the discussion of the SSBL.

The intent of undertaking conceptual design studies is to explore several design options in order to choose the best design approach compared to various evaluation criteria. The design criteria and objectives changed over the extended time period devoted to the studies. Initially, in 1972, the single shaft direct drive (SSDD) engine was considered as an alternative to the FT50 dual spool free turbine engine. The simple cycle efficiency of the SSDD, however, was about 2% poorer than that of the FT50 due to lower compressor pressure ratio. This was considered a serious liability for the SSDD because the engine was viewed primarily for simple cycle peaking applications.

The supercharged compressor studied in 1974 offered higher pressure ratio from a single shaft than previously available (up to 18:1 vs. 14:1) and eliminated the performance advantage of the dual spool for simple cycle applications. In this time period, however, it had become apparent that petroleum fuels would eventually become unavailable for base load power generation. The inherent efficiency of gas turbine combined cycles using coal derived fuels would, in time, result in their consideration as prime candidates for base load power generation.

The 1974 conceptual design was, therefore, held down to a pressure ratio of 15.5:1 by removing rear compressor stages to increase the temperature into the heat recovery boiler. Combined cycle efficiency could have been maintained at still lower pressure ratio, but it was felt that to assume performance would not suffer if more stages were removed too risky.

A design review of the 1974 conceptual design led to the conclusion that nearly all of the performance and mechanical design criteria could be met. Questions arose

concerning whether the horizontally split combustor case and turbine case was desirable and whether the lack of modularity was acceptable.

During 1975 the FT50 program was suspended due to a lack of a market for peaking gas turbines. Late in 1975 and early in 1976, new conceptual design studies were undertaken to arrive at an advanced base load gas turbine. The objectives of this design included substantial cost reduction and improved reliability by minimizing the number and complexity of the parts and assemblies. Also, it was desired to study a design maximizing the maintainability of the gas turbine by using a modular design.

This design study of what is now referred to as the Single Shaft Base Load (SSBL) engine continued through 1976 into mid-1977. The major features of the SSBL include:

- Single spool 10 stage compressor with a welded rotor
- Three bearing rotor
- Three stage turbine
- Elimination of the compressor and turbine intermediate cases and free turbine case
- Minimized number of airfoils
- Simple low loss diffuser case
- Annular combustor (later changed to can-annular)
- Split compressor case allowing on-site airfoil replacement
- Single module turbine rotor and front end generator drive, allowing access to annular combustor by turbine removal
- A significant reduction in manufacturing cost.

The SSBL conceptual design was laid out to examine a particular set of design options. These options would not necessarily be those chosen for a preliminary design, or if examined in preliminary design, might be eliminated before reaching the final design stage.

DISCUSSION

The following sections describe the pertinent features of the Single Shaft Base Load (SSBL) gas turbine shown in Figure 8-1. The technical background developed to support the SSBL design is also discussed in the following sections and in the attached reports. The intent of the studies leading to the SSBL was to use more advanced technology and to simpilfy the mechanical design and reduce the number of parts while maintaining a high performance level. The goals were greater reliability, reduced cost, weight, and maintenance cost. The SSBL conceptual design is based on the latest and most appropriate technology within the state-of-the-art available to the design team.

The simple, straightforward single shaft approach using appropriate levels of technology permits a relatively short engine with high airflow. Moderately high wheel speeds and low aspect ratio blading minimize the number of stages and airfoils. The low aspect ratio blades, while few in number, are physically large and rugged. The simple design permits relatively easy analysis of all temperatures, pressures, loads, and stresses both steady state and transient. Design analysis can then be used to ensure that each engine part and assembly will function in accordance with the design criteria including performance, durability, reliability, maintainability, and cost. The major design features are summarized in Table 8-1:

TABLE 8-1. MAJOR DESIGN FEATURES

Performance	
Combustor Exit Temperature	2300°F
Compressor Pressure Ratio	14:1
Airflow	840 #/sec
Power Output	112 MW
Heat Rate ISO/LHV)	9950 Btu/kWhr
Mechanical	
Front Drive	
3 Bearing Rotor	
10 Stage Compressor	
Can-Annular Combustor	
3 Stage Turbine	
Compressor Case: Horizontal Split	
Burner & Turbine Cases: Vertical Split	

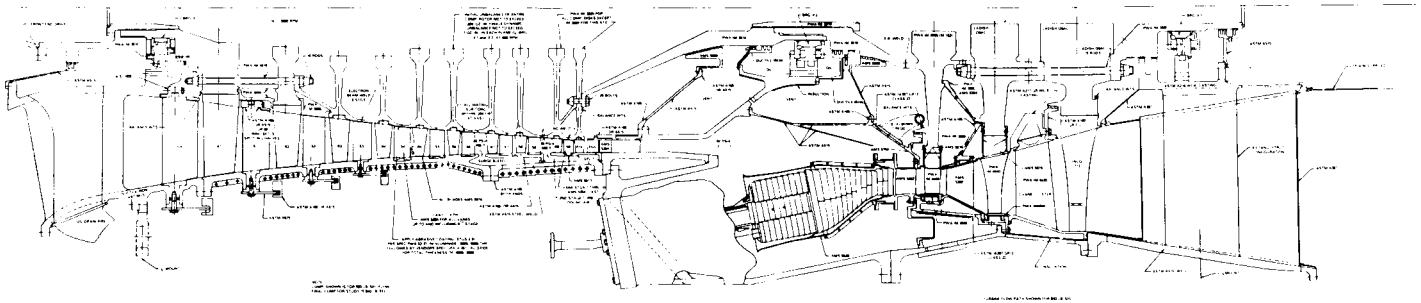


Figure 8-1. Single Shaft Base Load Gas Turbine Mechanical Arrangement

PERFORMANCE

The SSBL engine ISO rating is 112 MW at a 2300°F combustor exit temperature and an airflow of 840 lbs/second. At these conditions, the engine has an overall pressure ratio of 14:1 and a simple cycle heat rate of 9950 BTU/kW-Hr based on the lower heating value of the fuel. The variation of power and heat rate as a function of inlet airflow is shown in Figure 8-2. The component performance summary is shown in Table 8-2. Higher airflow would produce higher power, with slightly higher heat rate due to higher losses in the compressor, turbine and exit duct. The heat rate improves slightly with lower airflow, but power is lost in direct proportion to the airflow. The performance studies of 1972 and 1974 are shown in References 8-1 and 8-2, respectively.

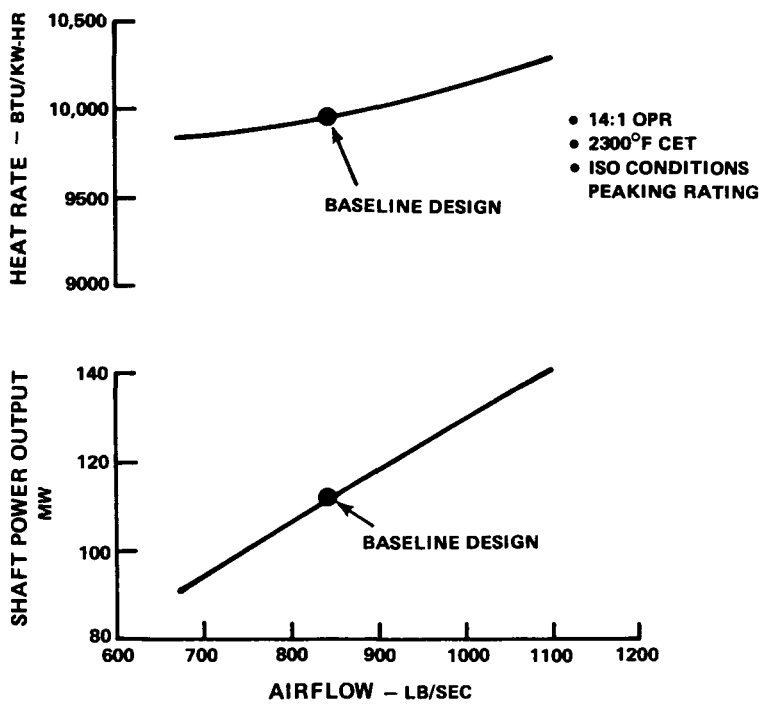


Figure 8-2. Power Output and Heat Rate vs. Inlet Airflow

TABLE 8-2. SINGLE SHAFT BASE LOAD GAS TURBINE PERFORMANCE SUMMARY

	Perf. Tables
Cycle Pressure Ratio	14.0
Combustor Exit Temp, °F	2300
Compressor Efficiency, %	87.8
Burner $\Delta P/P_B$	0.062
Turbine Efficiency, %	90.8
Turbine Cooling Air & Secondary Flow, %	22.8
Exhaust $\Delta P/P_T$	0.040
Bearing Losses, HP	650

The SSBL engine airflow of 840 lbs/sec was chosen to hold the last stage turbine blade stress to reasonable levels and lower exit Mach number, relative to the 1974 study of the supercharged compressor where 1028 lbs/sec was used. The lower exit Mach number would reduce the exit losses, but the reduced flow also would reduce the engine output and raise the specific cost ($/kW). The alternative of increasing the exit area to reduce the exit losses was considered, but rejected because of the increased turbine stresses.

Since the compressor would have to be extensively modified to reduce the flow, it was considered no additional risk to change the design pressure ratio to 14:1 from the 15.5 assumed in the 1974 study.

The combustor exit temperature of 2300°F selected for the SSBL is about 300° above the temperatures of currently operating peaking gas turbines using clean fuel. With advanced air cooling methods metal temperatures of 1500°F are achieveable at inlet temperature levels of 2300°F. This temperature, then, represents an upper limit for design optimization assuming evolutionary increases in turbine temperatures on light fuels.

Heavier fuels which have higher carbon to hydrogen ratios have higher flame luminosity. As a result, more heat radiates to the combustor liners and turbine vanes. The flame length may also increase and emissions will tend to increase. The heavy

fuels may also be dirtier than light distillates or natural gas. This will also increase emissions and corrosion rates. All of these factors, when considered at a given level of technology, will lower the permissible combustor exit temperature for heavy or dirty fuels consistent with a durable, reliable engine design.

The cycle pressure ratio selected is 14:1. This is a compromise between power and efficiency in both simple and combined cycle modes for light fuels up to 2300°F and heavier fuels at over 2000°F. The thermal efficiency for simple cycle and combined cycle operation over the temperature range of interest and for several pressure ratios is shown in Figure 8-3. For maximum power per pound of airflow at 2000°F combustor exit temperature, the pressure ratio should be 12.5 and for 2300°F, the pressure ratio should be 13. These pressure ratios are only slighly less than the 14:1 of the baseline engine and would give slightly higher simple cycle efficiency with a slight loss in power. To obtain maximum simple cycle efficiency, a pressure ratio of over 20 is necessary. This higher pressure ratio would introduce higher risk at a given technology level while decreasing reliability due to the increased number of parts. The cost of the higher pressure ratio engine would also increase, and there would be a reduction in specific power as shown in Figure 8-4. At 2300°F, for instance, a given airflow size engine would lose 15% in power output by increasing the pressure ratio from 13 to 22 to increase the efficiency from 33 to 35%.

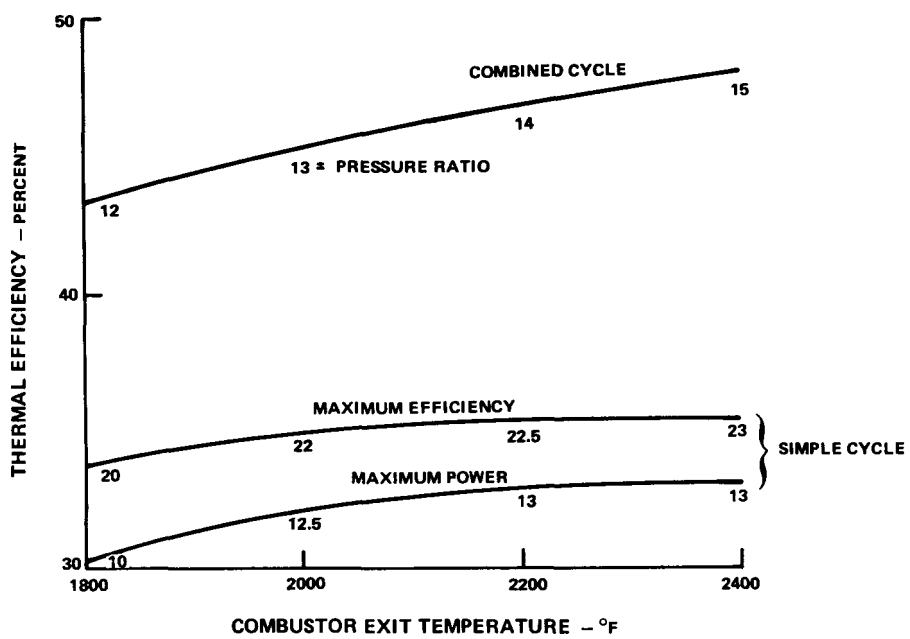


Figure 8-3. Comparison of Gas Turbine Cycles

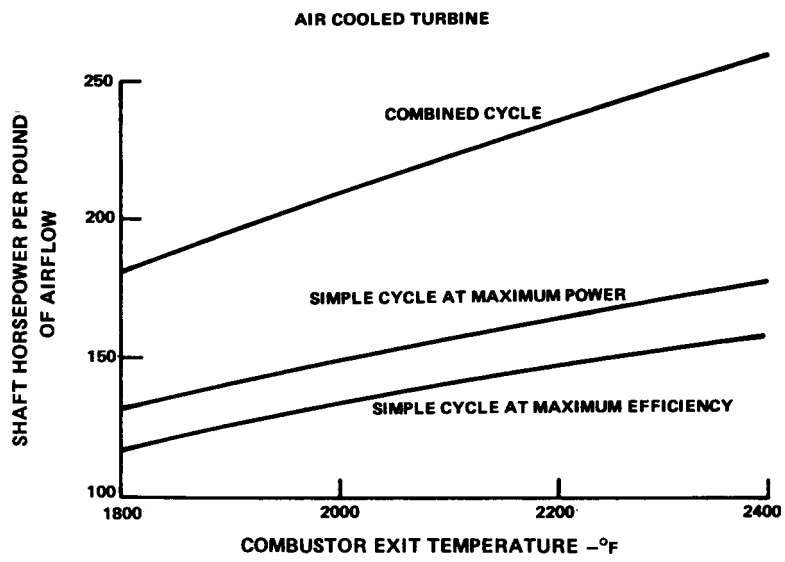


Figure 8-4. Specific Power of Various Cycles

Combined cycle efficiency is less sensitive to cycle pressure ratio. The increase in simple cycle efficiency of higher pressure ratio results in a lower turbine exhaust temperature. As shown in Figure 8-5, the steam cycle bottoming efficiency decreases with a decrease in turbine exhaust temperature offsetting the gain in simple cycle performance. Changes in overall reliability of the system related to gas turbine pressure ratio are small, but the gas turbine at a pressure ratio of 14:1 has an inherent advantage over one of higher pressure ratio and the exhaust temperature may permit a single pressure steam system which is simpler and slightly more reliable than a two pressure system characteristic of exhaust temperatures in the 800-900°F range.

MECHANICAL ENGINE DESCRIPTION

Overall Description

As shown in Figure 8-1, the rotor consists of a 10 stage compressor and a 3 stage turbine supported on three bearings. The power delivered by the engine is taken off from the front or compressor end. The static structure supports the bearings and rotor using three main strut-ring assemblies. The compressor case is split along the horizontal centerline for ease of maintenance. The compressor static structure locates the cantilevered compressor stators including a variable inlet guide vane and three additional variable stages.

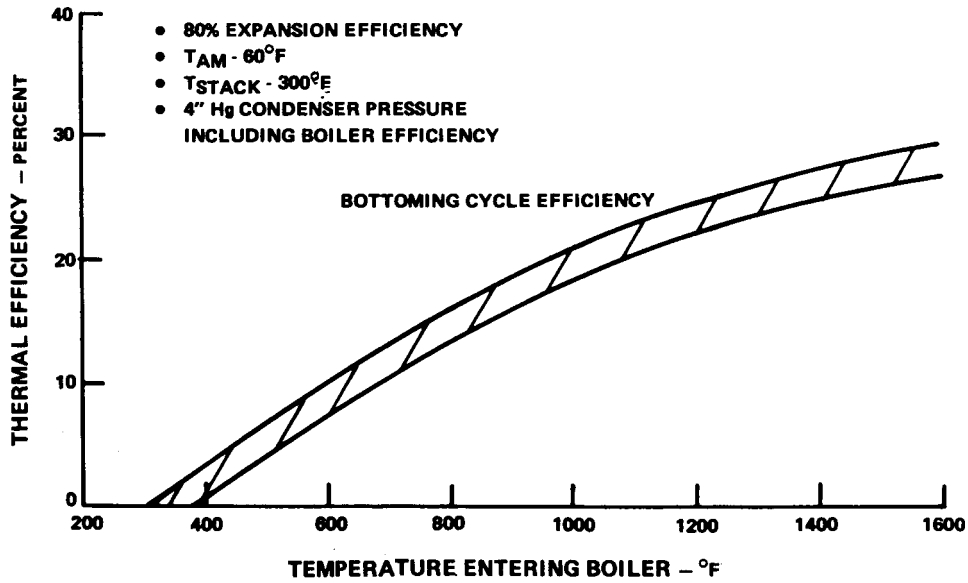


Figure 8-5. Approximate Steam Bottoming Cycle Efficiency

The compressor diffuser section and the turbine cases of the SSBL are not split along the horizontal centerline. These circular cases are split vertically only and permit modular removal of the turbine and combustor. The combustor cans can also be repaired and removed as necessary by translating the combustor case rearward over the turbine. If necessary, the first stage turbine nozzles can be inspected, removed, and replaced without further engine disassembly.

Compressor

The compressor of the SSBL is shown with 10 stages incorporating disks welded together into a monolithic, stiff drum rotor. The blades are loaded into circumferential slots and the configuration uses cantilevered stators with small tip clearances relative to the drum rotor. This construction departs from the mechanical construction discussed in the 1972 and 1974 reports of References 8-3 and 8-4, respectively. The welded rotor is stiffer and can transmit torque more readily than a tie bolt configuration and is, therefore, more applicable to the front drive arrangement.

Simple cast ductile iron compressor cases are used to support the cantilevered stators and provide for compressor bleed ports. The variable geometry stators are located by machined bosses in the case and actuated by linkages supported by the compressor case.

Diffuser - Combustor

The compressor diffuser and combustor are similar in concept. The mid bearing is located by the diffuser combustor assembly. Buffered seal oil compartments are used. The combustor cans are offset and shrouded. Extra space is provided for accommodating different combustor designs in anticipation of future requirements. The outer case can translate rearward to provide access for maintenance and for removal of the turbine as a module. The entire module can be removed rearward after removing a portion of the exhaust ductwork with the shaft being piloted by the mid bearing.

Turbine

A three stage turbine is used to reduce the number of parts. It is larger in diameter than the four stage turbines recommended in the 1972 and 1974 studies. It is of conventional construction with tie bolts and is similar in design to the earlier studies. The last stage steady state blade and disk stresses are somewhat higher than a free turbine design. This is compensated for by lower overspeed requirements. The nozzle vanes are shrouded at the inner diameter. Conventional labyrinth seals are used.

DESIGN ANALYSIS

Rotor Dynamics

The single spool design analysis study of Reference 8-5 was done in 1972 and concluded that the design criteria of 50% critical speed margin on stiff bearings and less than 25% rotor strain energy for soft bearings could be met with the three bearing design. The rotor construction of the SSBL engine is stiffer than that of the 1972 design and no critical speed problems would be expected. It is probable that only two bearings would be necessary if a welded or other stiff rotor configuration was used for the entire rotor.

Compressor End Generator Drive

The front end or compressor end drive has several advantages for direct drive gas turbines and was chosen for the SSBL. The drive shaft from the engine to the generator is shorter because the volume flow of the cold inlet air is less than the

hot exhaust flow. A shorter drive shaft can be used with improved vibrational characteristics and less displacement of the generator rotor due to thermal growth. In addition, the exhaust diffuser can be designed to be more efficient by the elimination of the drive shaft to give a lower heat rate.

Alternatively, the more efficient exhaust diffuser can be used to permit a higher exhaust Mach number (and flow) through a given flow annulus without a heat rate change relative to a rear drive. This is especially important in large engines where the airflow is limited by last stage turbine stresses. The turbine stress level is proportional to the exit annulus area and the rotational speed squared. Since the rotational speed is fixed by the generator, the higher flow for equivalent losses will yield higher power for the front drive than a rear drive configuration.

Bearings

The SSBL thrust bearing is located at the front of the compressor. This location was chosen as a result of a structural thermal analysis discussed in Reference 8-5 using expected temperatures and thermal response rates. It was concluded that the minimum last stage tip clearance would result from the front bearing location.

Tilting pad radial bearings and a combined thrust-radial front bearing were chosen for the SSBL based on previous studies. Unit loadings and surface speeds are well within the state-of-the-art and no problems are anticipated.

THERMAL ANALYSIS, MATERIALS AND SECONDARY FLOW

These topics are discussed in the Reference 8-5 excerpts from the 1972 Report and no further analysis for the SSBL was felt necessary.

COMPONENT SELECTION

Compressor

At the time of the 1972 and 1974 studies, Reference 8-6 and 8-7, respectively, achieving the design performance from high pressure ratio advanced technology compressors was judged to have excessive program risks for an electric utility gas turbine. Intensive development of compressors by P&WA for advanced aircraft engines

has minimized the risk of meeting the design performance levels. It is generally recognized that the most significant risk factor in the development of a new gas turbine is in meeting the compressor design objectives. To minimize risks, ground based gas turbine compressor aerodynamics have nearly always been scaled from aircraft engine compressors or based on the aerodynamic design information obtained from aircraft engine compressors.

The SSBL compressor, therefore, was initially scaled directly from an aircraft engine compressor design. The resulting 1028 pounds/second flow, however, is greater than desired and it was assumed that the flow could be reduced to 840 pounds/second by reducing the annulus slightly, and retaining the aerodynamic design. Other options would be explored in a preliminary design phase.

The SSBL compressor performance parameters are summarized in Figure 8-3. The airflow of the compressor is 10% to 40% higher than current large 3600 RPM industrial gas turbines. The tip speed at 1250 fps is within the state-of-the-art for aircraft high compressors, and about 100 to 150 fps higher than current industrial gas turbines. The reduced number of airfoils and stages and relatively high pressure ratio result from the somewhat more aggressive performance parameters. The performance goals for this compressor have been substantially met by testing.

TABLE 8-3. COMPRESSOR PERFORMANCE PARAMETERS

COMPRESSOR SUMMARY	
Number of Stages	10
Corr. Airflow, LB/SEC	840
Corr. Tip Speed, FT/SEC	1250
Inlet Specific Flow lb/sec/Ft^2	38.0
Inlet Hub/Tip Ratio	0.60
Avg. Aspect Ratio	1.5
Number Variable Geom. Stages	IGV + 3
Surge Margin	>15%
Exit Hub/Tip Ratio	0.918
Exit Mach No.	0.30

The study completed in 1972 (Reference 8-6) compared the performance of the Advanced Aircraft Compressor (AAC) to the FT50. It was concluded that there was a 0.9 percentage points lower efficiency for the AAC. The 1974 study (Reference 8-7) updated the earlier one, and considered the added supercharging stage. It was concluded that the new compressor had adiabatic efficiency equal to the best demonstrated in current industrial gas turbines.

Combustor

A can-annular combustor system similar to current industrial designs was chosen for the SSBL conceptual design. This arrangement enables convenient maintenance of individual cans by sliding the combustor case rearward. The design parameters of length, diameter, flow velocities, heat release rates, etc. were chosen consistent with current design practice. These parameters are summarized in Figure 8-4. The cans are shrouded to promote uniform flow and minimize temperature maldistribution. The estimated pressure loss is 6.2%.

Design studies were also conducted of annular combustors which have a low ratio of volume to surface area. The annular combustor could be partially maintained by sliding the combustor case rearward, but major maintenance would require removal of the turbine module.

The offset combustion system is also adaptable to modifications to enable complete removal of all the compressor air to a remote combustor such as a fluidized bed and return of the heated air to the turbine. Studies of the offset combustion systems have shown that they are adaptable to a wide variety of fuels and configurations without affecting the basic engine design and rotor dynamics.

The can-annular combustor studies are discussed further in the summaries of the 1972 and 1974 reports of Reference 8-8 and 8-9, respectively.

Turbine Section

The SSBL conceptual design has a three stage high work design to reduce the number of parts and to rapidly reduce the gas temperature to simplify cooling. The three stage turbine at an overall engine pressure ratio of 14:1 would be more lightly loaded than a two stage turbine at an overall pressure ratio of 9:1. Compared to a four stage turbine, the velocity ratio of the three stage turbine is lower and the

exit swirl is higher; the efficiency loss has been estimated to be less than 1%. Part of this efficiency loss is recovered by requiring less cooling air and part is recovered in the steam system of a combined cycle powerplant.

TABLE 8-4. SUMMARY OF SSBL DESIGN PARAMETERS

	Peaking Power - 59° Day		
	Twin Spool Free Turbine	SSDD	SSFT
Wa - lbs/sec	695	710	780
Mn4	0.26	0.30	0.30
Diffuser Length	18"	21"	22"
Length Comp. Exit to TIGV	44"	54"	56"
ΔP/PT4 Burner Section	.065	.065	.065
Required Burner L/D	3.4	3.4	3.4
Burner Diameter	14.3"	14.8"	15.65"

The pressure ratio of 14:1 is partially set by the desire for no more than three stages of turbine. The efficiency loss and risk due to extending the state-of-the-art of going to a much higher pressure ratio, say of 18:1, with a three stage turbine would be prohibitive. A four stage turbine would then be required. An advantage of a four stage turbine is that it is conveniently convertible from a direct drive engine to one that incorporates a free turbine. The gas generator design could then be directly scaleable to other sizes.

The engine airflow is set by the allowable stress in the last stage turbine blade. In the baseline engine, the flow is calculated to be 840 pounds/second. There is a trade between cost, power, efficiency, exit Mach number and risk that must be continuously assessed before this vital parameter is finally selected.

The total work is 295 BTU/pound yielding an expansion ratio of 12.4 for an average pressure ratio of 2.3 per stage. This level of work is higher than that of currently operating industrial designs, but well within available technology. The mean velocity ratio is nearly 0.6 for good efficiency.

The 1972 study in Reference 8-10 and the 1974 study in Reference 8-11 considered both three and four stage turbines and recommended that the four stage be used due to the higher efficiency.

RELIABILITY

Two fundamental paths were pursued to improve the reliability of the SSBL beyond current and other advanced engines. The first path, undertaken in the conceptual design, was to reduce the number of parts and stages and to design functionally simpler parts. The second path, which is more appropriate to the preliminary design phase, is a thorough design analysis of each part and function to ensure that design life and other criteria are met.

Engines of current design that are simple in design and operate at constant speed are not necessarily reliable. Part of this reliability problem has been due to an inability to compute transient stresses and LCF life. To achieve high inherent reliability additional analysis using advanced computer techniques is required.

Reliability can also be improved using data obtained during the early development of the engine. It is important that the test data be compared to the predicted values and the resulting effect on stresses and lives be recomputed with changes in design made where needed.

The 1972 study in Reference 8-12 concluded that the single shaft designs had a higher potential reliability than the dual spool design. The free turbine configuration was initially favored due to lower starting torque requirements and the higher power assumed in the free turbine design. The later supercharged version eliminates the power advantage of the free turbine since the turbine exit area sets the maximum flow, not the compressor.

MAINTAINABILITY

Currently available gas turbines are of two types: aircraft derivative designs and industrial designs such as the SSBL. The aircraft derivative gas generators are maintained in place if the maintenance time is reasonably short, but are exchanged for replacement gas generators if extensive maintenance is required. Industrial type engines are presently maintained as installed for both light and heavy maintenance. The quick changeout feature of the aircraft derivative engine reduces down time and increases availability. A criteria used in all of UTC's industrial engine design studies has been to retain the capability of quick change out, usually by incorporating modular designs for the larger engine sizes. The SSBL engine represents one of many possible modular design concepts. For the SSBL design, the

compressor case is horizontally split, enabling access to the compressor blading for repair and replacement. The combustor section and turbine sections are vertically split cases. Access to the individual combustor cans is obtained by sliding the combustor case rearward. Also, the first stage turbine vanes can be removed and replaced easily. If more extensive turbine repair is required, the entire turbine module can be translated rearward after opening the exhaust duct.

Maintainability, reliability and cost are related and trades must be made to optimize the system. The vertically split cases, for example, have lower transient stresses than horizontally split cases, but may impair maintainability unless special design features such as translating the cases for maintenance are used. This thermal flexibility of the vertically split cases contributes to longer LCF life for the cases. The physical size and weight of the SSBL engine modules and other components were evaluated and found acceptable for maintenance. The nominal rating of the SSBL engine is 100 to 120 MW and the engine would weigh approximately 100 tons.

The 1972 study in Reference 8-13 concluded that the single spool free turbine would have the lowest maintenance cost. The data from which this conclusion was reached are shown in Table 8-5.

TABLE 8-5. TURBINE DATA SUMMARY

Number of Stages	3
Expansion Ratio	12.4
Work, BTU/LB	295
Mean Velocity Ratio	~0.59

CONTROLS

The control functions including response and acceleration, load following, starting, normal shutdown and drop load, all affect the operation, reliability, and maintainability of the SSBL engine. These functions differ considerably in concept between direct drive SSBL and free turbine engines. The SSBL engine is a direct

drive engine and rotates at 3600 RPM. Load variation and response, therefore, is limited mainly by thermal stress (thermal shock) considerations rather than rotor inertia.

Starting of the SSBL engine is rotor inertia dominated because the gas turbine rotor and generator rotor are connected. The starter power assumed in the 1972 study of Reference 8-14 was higher for the direct drive engine than for the free turbine engines to achieve enough airflow and speed to light off the combustor in the same length of time. Acceleration after light off is initially slower for the direct drive, but then more rapid because the entire turbine is available to provide the power to accelerate the rotor system. The total start time to full load was calculated to be 5 minutes for all systems in the 1972 study. The engine shows a variable inlet guide vane and three stages of variable compressor vanes, these variable vanes may not be required because a similar effect can be achieved with compressor bleed. Acceleration to synchronous speed and time to full load are set by thermal shock considerations.

Normal shutdown is accomplished by reducing power, disconnecting the unit from the line, and decelerating slowly to minimum fuel flow and shutdown. Barring would continue until the system is cold. Drop load is accomplished by rapid cutoff of fuel flow and coastdown of the engine. The control system would distinguish between false and real drop loads and backup mechanical overspeed devices would positively prevent rotor overspeeds beyond specified limits.

The SSBL engine concept is intended to reduce the needs for auxiliary functions. For example, the minimum number of bearings minimize the lubrication requirements. Experience indicates that reliability can be improved considerably by proper attention and analysis of the auxiliary systems.

The steady state and transient performance characteristics were also investigated in the 1972 study. It was concluded that the direct drive engines have excellent load pickup characteristics since the rotor does not have to be accelerated. The underspeed is much less. The direct drive engine is also advantageous during dropload or other potential overspeed conditions. The recommended overspeed margin was 10% for the direct drive versus 20% for the free turbine engines.

REFERENCE 8-1

PERFORMANCE

August 1972

Performance comparison of the Single Shaft Base Load gas turbine with direct drive and free turbine power output

Summary

This section presents the results of performance studies of an industrial gas turbine engine based on a scaled advanced aircraft engine compressor. Both direct drive and free turbine engines based on the single shaft gas generator concept were studied. This section discusses the design point selection, performance, and engine growth potential.

Discussion

Design Point Selection

The compressor match points for the Direct Drive and Free Turbine engines were selected based on requirements for adequate surge margin at the 0°F max peak rating, maximizing ISO peak power, and maintaining a cold day power lapse rate equal to single shaft direct drive engine data.

The following design points were selected for the Direct Drive and Free Turbine engines:

	SSDD	SSFT Max Scale
Pressure Ratio	14.3	14.0
Airflow	710	780
Speed - RPM	3600	3520
% AAC Corrected Airflow	98.3	100
% AAC Corrected Speed	98.5	100
Power Output	80.2	85.1
SFC - LB/HR HP	0.399	0.412

These engines were based on scaling an advanced aircraft engine high compressor which had a considerably lower design point corrected airflow and much higher rotational speed.

A study of direct drive engines, summarized in Figure 8-6, was conducted in which several match points were selected at pressure ratios of 14:1 and 15:1 with 100% and 95% of the original inlet corrected airflow. The compressors were scaled up from their respective match points to provide 3600 RPM while maintaining constant compressor aerodynamics.

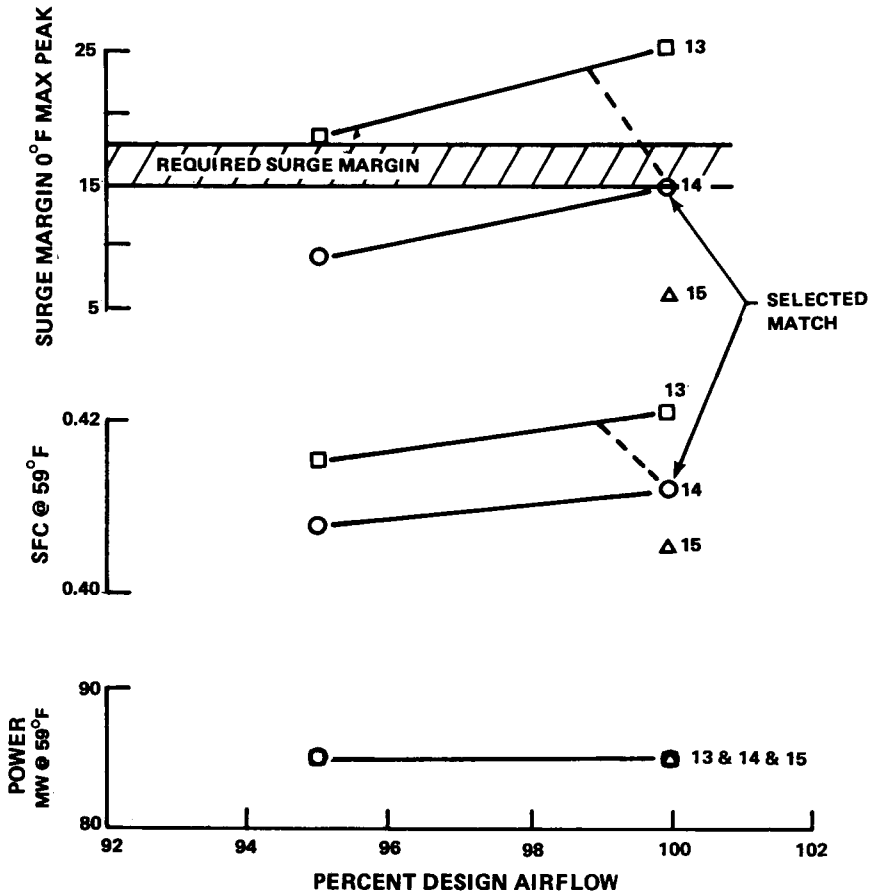


Figure 8-6. Compressor Match Single Shaft Direct Drive

Performance was evaluated for each engine with a fixed stator setting at 59°F and 0°F peak load rating, and adequate surge margin at the 0°F maximum peak condition. The SSDD power variation at off design conditions would depend primarily on the compressor efficiency lapse rate and the compressor speed-flow relationship. The power increase from 59°F to 0°F was smaller, with the higher flow and lower pressure ratio match points due to a poorer compressor efficiency lapse rate. The higher airflow match would provide greater power output at 59°F peak load condition due to the larger compressor scale factor. However, fuel consumption would be slightly poorer due to reduced compressor efficiency.

Design points at 14:1 pressure ratio at 100% and 105% of the reference Advanced Aircraft Compressor (AAC) corrected airflow were also considered with the compressor stator setting fixed open 10°. The increased stator opening reduced the compressor scale factor at 100% flow because of a lower corrected speed, and consequently decreased the power output. Fuel consumption was increased due to poorer design point efficiency, and 0°F maximum peak surge margin was only 8%. The 105% AAC corrected flow design provided sufficient 59°F power and 0°F maximum peak surge margin, but had poor 0°F peak load power. On this basis, it appeared that setting the stator opening beyond the AAC maximum open position was not desirable.

The direct drive compressor match was selected at 14.3 pressure ratio and 98.3% flow because it maximized ISO peak power while providing 22% greater power than ISO at 0°F peak load rating; and maintained 10% surge margin at the 0°F maximum peak condition.

The compressor stator vane schedule was held constant at the AAC max open position to minimize the power loss at non-standard day conditions. A variable stator schedule in which stators would be closed in the hot day operating range, according to the AAC schedule, would reduce flow and hot day power significantly. Opening the stators 10° and the 0°F peak rating (105% AAC design corrected speed) would reduce cold day efficiency and power output even though flow would be increased.

A similar study was conducted with the Free Turbine engine and is summarized in Figure 8-7. The operating line shape and flow excursions from 0°F to 104°F ambient temperature require a variable stator vane schedule throughout the operating range. Compressor match points at pressure ratios of 13 and 14:1 were evaluated at 100% and 95% AAC corrected flow and 15:1 pressure ratio at 100% corrected flow. While the compressor scale factor for the direct drive engine was dictated by the 3600 RPM speed requirement, the free turbine engine allowed greater flexibility in compressor scaling. Since the twin spool gas generator could be scaled up 5% to provide 85.5 MW at 59°F peak load and improve its cost effectiveness, the free turbine engines were scaled to provide the same power output. Similar trends result in cold day power variations and surge margin at maximum peak load as the direct drive machine. The selected match point for the free turbine engine was 14:1 and 100% AAC corrected flow. This would provide a favorable cold day power lapse rate and 10% surge margin at 0°F maximum peak load condition.

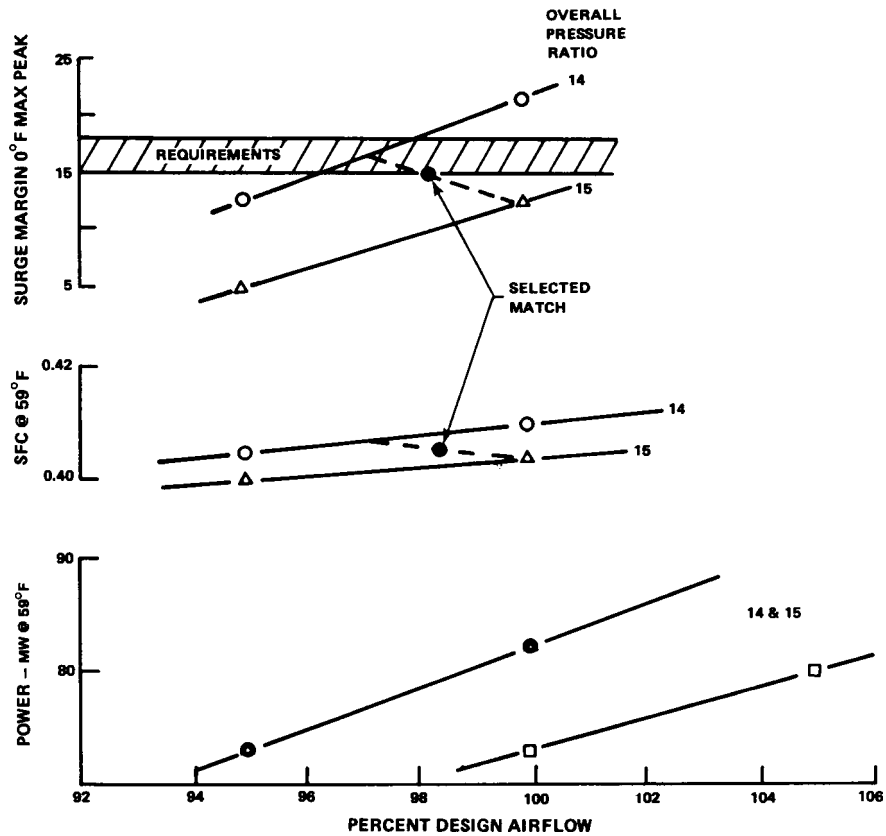


Figure 8-7. Compressor Match Single Shaft Free Turbine

Performance

The performance comparison of the single shaft direct drive (SSDD) and single shaft free turbine (SSFT) with the twin spool free turbine (TSFT) at peak load is presented in Table 8-6.

The SSDD engine would have 2.8% and 5% more power than the TSFT in simple and combined cycle configurations, respectively. Component efficiencies and secondary flows were handled in a consistent manner to provide a direct performance comparison between the AAC derivative engines and the twin spool design. The SSDD simple cycle fuel consumption is 1.8% poorer than the TSFT due to lower cycle pressure ratio, lower compressor efficiency and higher second turbine stage cooling flow penalties.

The SSFT engine would have 0.7% less power with the same compressor scale factor as the SSDD engine because of lower compressor efficiency, higher bearing losses and additional rotor thrust balance flow.

TABLE 8-6
PERFORMANCE COMPARISON
ISO CONDITIONS

	TSFT	TSFT Scaled to SSDD	SSDD	SSFT	SSFT Max Scale
Simple Cycle					
Power - MW	78.0	80.2	80.2	79.6	85.1
Heat Rate-BTU/KWH	9932	9950	10100	10380	10430
SFC	0.392	0.393	0.399	0.410	0.412
Pressure Ratio	16	16	14.3	14.0	14.0
Airflow - LBS/SEC	695	715	710	726	780
Combined Cycle with 2 Pressure Steam System					
Power	103.7	106.6	108.5	109.2	117.0
Heat Rate	7470	7477	7470	7570	7590
SFC	0.295	0.295	0.295	0.299	0.300

The gas generator of the free turbine engines could be scaled up to produce greater output since it would not have the speed requirement of the direct drive engine. Previous twin spool studies indicated that the free turbine mechanical limitations would prevent its being scaled up beyond its present size, but that it would be cost-effective to scale the gas generator up 5% before exhaust losses became excessive. For this study, consideration was given to scaling the TSFT gas generator 1.5% to obtain the SSDD power output, and the SSFT to produce 10% more power than the TSFT. The fuel consumption of these engines would be higher because of increased exhaust system losses.

The performance variation with ambient temperature for the SSDD and SSFT (max scale) engines compared to the TSFT would show that the SSDD maintains a greater hot day power output relative to 59°F than the free turbine engine because of a smaller reduction in corrected airflow at hot day conditions.

The combined cycle performance was calculated based on a two pressure steam system with the following assumptions:

- High Pressure Boiler Pressure 1000 psi
- Exhaust Stack Temperature 300°F
- Boiler Pinch Points 50°F
- Condenser Subcooling 0°F
- Gas Turbine Exhaust Pressure Loss 10" H_2O

The lower compression ratio AAC engines would produce more steam power than the TSFT engine because of their higher gas generator flows and exhaust temperatures.

Engine Growth Capability

The growth capability for the AAC engines and the TSFT is shown in Figure 8-8. The first growth step was defined as a 200°F increase in turbine temperature. Assuming that this growth step would be accomplished by hot rematching, then all engines show approximately 15% greater output.

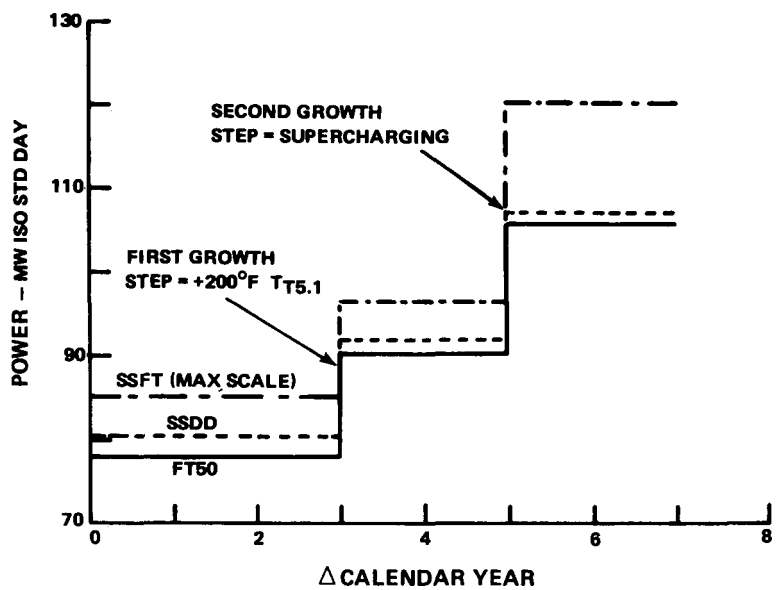


Figure 8-8. Comparison of Engine Growth Capability

The second growth step considered was the addition of a supercharging stage. A comparison of compressor characteristics is tabulated in Table 8-7 where it was assumed that the supercharging stage was added to the 1st growth engine.

TABLE 8-7
GROWTH COMPRESSOR COMPARISON

	TSFT		SSDD		SSFT Max Scale	
	Base	Supercharged	Base	Supercharged	Base	Supercharged
Supercharging Stage Pressure Ratio	-	1.25	-	1.37	-	1.37
Overall P.R.	16	20	14.3	16.3	14.0	19.2
Wa $\sqrt{\theta T}/\delta T$ lb/sec	695	839	710	859	780	1016
N1/ $\sqrt{\theta T}$	3720	3860	3600	3600	3520	3695

For this step it was assumed that the added supercharging stage was similar in pressure ratio to the first stage of the base engine. The AAC stage pressure ratio of 1.37, compared with 1.25 for the TSFT, gave the AAC compressor machine a potential for a much greater growth increment. The SSFT and the TSFT, because of the free turbine feature, could permit the compressor to increase in speed to allow full exploitation of the supercharging stage. The TSFT showed a 17% growth increment while the AAC SSFT configuration showed a 25% increase.

The SSDD engine, even with its 1.37 supercharging stage, showed a growth increment comparable to the TSFT because it is forced to run at a constant speed.

The supercharged free turbine engine could be matched down 115°F and deliver the same hot day power as the supercharged SSDD.

REFERENCE 8-2

PERFORMANCE

March 1974

Performance of the Supercharged Industrial Single Shaft Engine

Summary

The performance of an industrial engine based on a supercharged advanced aircraft engine compressor was studied. The selected cycle assumed a pressure ratio of 15.5 and a flow of 1028 #/second. The higher flow and pressure ratio resulted in higher power and lower heat rate than either the twin spool free turbine (TSFT) or the unsupercharged single shaft engine.

Discussion

A performance study was made assuming a linearly scaled supercharged Advanced Aircraft Compressor, with one rear stage removed and with two rear stages removed. The latter configuration included design pressure ratio of 14.3:1 and rematched to 15.5:1 at flows of 100% and 95% of design. Selection of the configuration with two stages removed and a design point of 100% flow at 15.5 pressure ratio was made on the basis of meeting performance objectives and maintaining sufficient surge margin.

Table 8-8 shows a comparison of the performance of the twin spool free turbine, the single shaft engine, and the August 1972 single shaft engine.

TABLE 8-8
PERFORMANCE COMPARISON
ISO CONDITIONS

	Twin Spool Free Turbine Engine	Single Shaft Engine	August 1972 Single Shaft Engine
Power MW	86.04	117.1	80.22
Heat Rate BTU/kW Hr	9940	9870	10110
Pressure Ratio	16:1	15.5:1	14.3:1
Airflow LBS/SEC	762	1028	710
Compressor Efficiency	0.872	0.872	0.860
Turbine Efficiency	0.912	0.915	0.912

REFERENCE 8-3

ENGINE MECHANICAL DESCRIPTION

August 1972

Single Shaft Direct Drive and Free Turbine Industrial Engines

Summary

The Single Shaft Direct Drive (SSDD) industrial engine would have an 11 stage compressor connected to a 4 stage turbine. The turbine drives the compressor and delivers shaft power to an output coupling. The 14.3 pressure ratio compressor was assumed to be a linear scale of an advanced aircraft engine compressor. The SSDD would weigh 85.5 tons, be 24.5 feet long, and have a maximum diameter of 10.5 feet.

The single rotor having a speed of 3600 rpm and weighing 22.5 tons would be supported by 3 radial hydrodynamic bearings. These bearings were located in the inlet, diffuser/burner, and exhaust case structure. A single thrust bearing was located in the inlet case. This location was chosen because it minimizes compressor tip clearance with no penalty to turbine tip clearance. Also, this region would be the coolest bearing location and provide ease of access to the thrust bearing. The engine modules consisted of: (1) Compressor, (2) Diffuser/Burner Section, and (3) Turbine. These basic modules would be located between inlet and exhaust cases to form a basic gas turbine package.

Discussion

Compressor

The compressor rotor construction studied consisted of 11 stages of compressor disks spaced apart by bore and rim spacers. Four tierods were assumed to pass through the disks and bore spacers to hold the compressor assembly intact prior to engine assembly. At engine assembly, a 142" long tiebolt would pass through the disk bores providing the rotor structural "tie" of the compressor. The aft end of the tiebolt threaded into the turbine front hub while the compressor front hub, disks, spacers were axially pinched by tightening a nut on the forward end of the bolt.

The 1st stage compressor rotor would have titanium blades which would be required to obtain allowable foil root stresses. The blade attachment stresses in the first

four stages of the SSDD were comparable to those in the TSFT low compressor. The stresses in the last few stages in the SSDD were lower.

The compressor disks were designed for a burst margin of 1.375 based upon a nominal speed of 3600 rpm while the TSFT had a value of 1.25 based upon first growth engine speed. This higher burst margin was required to account for overspeed during "drop load." Compressor disks in stages 4 thru 8 were sized to dead loads while sizing of the last three stages included effects of steady state and transient temperature gradients. Stages 1 through 3 were influenced by temperature gradients caused by 3rd stage bleed air washing the disk bores.

The TSFT compressor blades and vanes were designed with a minimum nominal leading a trailing edge radii of 0.020 at the root and 0.010 at the tip. A direct scaled version of the advanced aircraft compressor resulted in several stages having leading and trailing edge radii of 0.018 at the root. Slight thickening was recommended. This was not a problem for the tips because a scaled version resulted in greater thicknesses for the SSDD.

Systems

The diffuser case bearing and seal area was simpler than the TSFT design since the SSDD did not have an inner shaft bearing or a separate high pressure compressor thrust balance seal vent system. The SSDD had a single high pressure air seal discharging into the oil compartment vent system. The SSDD single seal in combination with the 4 stage turbine and 11 stage compressor gas loads resulted in a 20,000 lbs. bearing thrust load.

The bearing compartment was buffered by 3rd stage air that also cooled the compressor disks. This air would be fed through holes in the compressor rotor rear cone, thereby eliminating external plumbing. The major portion of the compressor cooling air is discharged through additional holes in the rear cone to enter the diffuser case vent region. The SSDD required a diffuser case flow vent area of 158 in.^{2} as compared to 115 in.^{2} for the TSFT design. The flow area increase was necessary to handle the cooling of the compressor and turbine disk bores.

The compressor and turbine rotors were assumed to be connected together by a splined joint under the diffuser case bearing. The long tierod that would hold the compressor together was also used to tie the turbine to the compressor. Since the

tiebolt would be easily accessible from the front of the engine, it simplified removal of the compressor from the turbine.

Combustor and Diffuser

The diffuser and combustor case design were similar to the TSFT designs, except the axial spacing between the compressor EGV and the turbine 1st stage nozzle was increased by 10" to a length of 54". Lengthening this area was necessary to prevent overlapping of the compressor bleed areas by the combustor section. An additional benefit of increased burner "window" area was obtained by this length increase. The SSDD burners were approximately 3% larger in size than the TSFT combustors. The O.D. of the combustor section was 10 ft. 6 in., which was 9 inches larger than the TSFT engine. The larger diameter was primarily attributed to the larger compressor.

Turbine

The SSDD turbine rotor would consist of 4 turbine stages spaced apart by bore spacers. 16 tierods passing through these members provided the axial structural tie of the turbine rotor. This turbine rotor was supported by two bearings, one located in the diffuser/burner case, and the other in the turbine exhaust case.

The SSDD would not require an inter-turbine case, which is a significant advantage over the TSFT engine. Mid-web spacers were located between the 1st and 2nd stage disks to form a cavity for transferring cooling air to the 1st and 2nd stage turbine blades. This cavity system eliminated the TSFT external piping system often used to deliver cooling air to the 2nd stage turbine blade. This cavity system also would result in minimizing the length of the blade cooling air holes located in the rim of the 1st stage turbine disk. A tangential on-board injection system, located forward of the 1st stage turbine disk, supplied air to this cavity which in turn supplied compressor discharge air to the 1st stage turbine blades. 2nd stage blade cooling air would flow from this cavity, through axial holes in the mid web spacers and the 2nd stage disk, to feed the radial holes located in the rear side of the 2nd stage disk rim. This feed system eliminated a seal and associated air leakage.

The lower wheel speed in the SSDD and SSFT engines would permit the 1st turbine disk bore to be smaller in thickness and diameter than the TSFT. The 2nd turbine SSDD disk would be larger and heavier than the 2nd stage TSFT. Lower rpm, increas-

ed stage loading, and larger diameter rims in the SSFT and SSDD engines resulted in nominal speed dead rim loads equivalent to the TSFT 2nd stage turbine. The 3rd and 4th stage turbine disks in the SSDD would be lighter than in the SSFT or TSFT because of the lower SSDD overspeed. These disks in the SSFT would be similar to the TSFT.

The 1st turbine blade would be cooled with compressor discharge air, but the reduced injector effectiveness would cause the SSDD cooling air temperature at the blades to be approximately 20°F higher than the TSFT during standard day operation. This temperature difference would increase to 55°F for hot day operation. While the TSFT 2nd blade was cooled with 13th stage air, the SSDD and SSFT would be supplied with compressor discharge air internally delivered to the 2nd disk in these engines. The 2nd vane would be supplied with 9th stage air in the single shaft engines, whereas the 2nd vane was supplied with 13th stage air in the TSFT.

The flowpath and speed changes in the SSFT and SSDD would make the 1st turbine blade stresses significantly lower than in the TSFT. The foil root stress alone was reduced by approximately 30%. Since the FT50 went from X-40 to In 738 because of excessive foil root stress, it would appear reasonable that X-40 can be considered for the SSDD and SSFT. However, the impact of this modification on disk material selection for attachment stress would also have to be appraised.

The turbine exhaust case for the SSDD would be similar in size and design to the TSFT engine, except for elimination of features required for mounting the TSFT free turbine thrust bearing.

Bearings

The radial bearings for the SSDD engine would be similar in design to those used on the TSFT. The SSDD inlet case bearing would be 15.00" in diameter and a 16.00" diameter bearing would be used in the exhaust case. To obtain critical speed margin for the SSDD rotor, it would be necessary to use an 18.00" shaft diameter in the diffuser/ burner section. This bearing would have a maximum steady state velocity of 284 ft/sec as compared to 286 ft/sec for the TSFT Number 4 bearing. The SSDD diffuser/ burner bearing would have a radial load of 24,900 lbs. as compared to 12,180 lbs. for the TSFT Number 4 bearing. As a result, the TSFT Number 4 bearing required an oil flow of 59 gpm as compared to 115 gpm on the SSDD diffuser case bearing.

3rd stage air supplied from inside the rotor system would be used to buffer the aft side of the inlet case bearing and also to buffer the diffuser/burner section bearing compartment. 4th stage air piped externally would be used to buffer the turbine exhaust case bearing and cool the turbine disk bores.

The version would have a gas generator 3.7% larger in linear scale than the SSDD engine. The compressor would be driven by a two stage turbine. The SSFT would have a gas generator speed of 3570 rpm compared to 3600 rpm for the SSDD. This combination of size and rpm was selected to obtain the maximum power output before reaching the size limitation of a free turbine the same size as the TSFT. The inter-turbine and exhaust case designs are similar to the TSFT engine.

The diffuser/burner section bearing would be moved 10" further forward in the SSFT than in the SSDD to maintain rotor critical speed margins. This would be required on the SSFT because the gas generator turbine rotor would be lighter and its stiffening contribution on compressor deflection mode would be reduced.

The diffuser case vent air flow area requirement of 58 in.^{2} in the SSFT would be much less than 158 in.^{2} required for the SSDD. This difference resulted from the compressor thrust balance seal on the SSFT, which was not required on the SSDD, since the free turbine thrust load nearly balances the compressor thrust load. The vent area for the thrust balance seal in the SSFT would be lower because of the higher vent pressure which would discharge to the free turbine.

Supercharged Versions of SSDD and SSFT

The supercharged version of the SSDD would be achieved by adding a compressor stage in front of the compressor. The rotor speed would remain at 3600 rpm for this direct drive machine. The additional stage would increase the span across the bearings from 138" to 156". To obtain critical speed margins, it would be necessary to have compressor bore spacers 3.00" thick as opposed to 0.8" thick for the non-supercharged version.

Supercharging of the SSFT would be accomplished by adding a similar stage. The rotor speed would increase from 3570 rpm for the SSFT to 3740 rpm for the supercharged version. The additional stage would increase the bearing span across the compressor from 128" to 146".

REFERENCE 8-4

ENGINE MECHANICAL DESCRIPTION

March 1974

Single Shaft Direct Drive Engine Using a Supercharged Advanced Aircraft Engine Compressor

Summary

The single shaft direct drive engine based on a supercharged advanced aircraft compressor would be mechanically similar to the August 1972 (Reference 8-3) engine except for the supercharging stage and incorporation of a front drive of the generator. The supercharging stage would increase the mass flow significantly, and the front drive would allow more flexibility in the design of the exhaust system (Table 8-9).

Discussion

The single shaft engine would incorporate a scaled up supercharged advanced aircraft compressor with the 2 rear stages removed. The compressor would be matched at 15.5:1 pressure ratio and 100% flow.

The compressor of the study engine would be scaled up from a supercharged advanced aircraft compressor with the 2 rear stages removed, resulting in a total of 10 stages. The burner would be larger than in the earlier study because of the larger flow. The turbine, as in the previous study, would have 4 stages to drive both the compressor and electric generator.

Mechanically, the engine would be similar to the previous study except that it would be updated to reflect the latest experience of the TSFT design and revised to drive through the front. The front drive would allow freedom to optimize the exhaust system without compromising the length due to the shaft critical speed requirements. The turbine discharge area would be kept the same as the TSFT and the earlier single shaft direct drive engine to avoid excessive stresses. An increase in exhaust losses due to higher exit Mach numbers would result.

Other configuration and operational characteristics would not change significantly from those described in the previous study of August 1972 (Reference 8-3).

TABLE 8-9
COMPARISON OF TSFT AND THE SCALED SUPERCHARGED COMPRESSOR

Parameter	TSFT LPC	TSFT HPC	TSFT Overall	Supercharged SSDD
Corrected Flow (lb/sec)	766	240		1033
Corrected Speed (RPM)	3522	3407		3600
Pressure Ratio	3.8	4.18	16.0	14.3
Discharge Mach Number	0.46	0.27	0.27	0.33
Average Mach Number	0.649	0.567	0.608	0.72
Number of Stages	7	10	17	10
Hub/Tip Diameter				
Inlet	0.541	0.790		0.567
Exit	0.790	0.902		0.916

REFERENCE 8-5

DESIGN ANALYSIS

August 1972

Analytical Design Aspects of the Industrial Single Shaft Gas Turbine

Summary

This section reports the results of the design analysis efforts carried out in support of the single shaft gas turbine studies. Both direct drive and free turbine versions were considered in most of the analyses. The design analyses were done for what were assumed to be the most critical aspects of the conceptual design. In most cases, the single shaft designs easily met the established industrial requirements. In a few cases, the criteria could not be met and alternative designs were recommended.

Discussion

The following discussion records the results and conclusions reached during the Industrial Single Shaft Engine Study.

Rotor Dynamics

Two single shaft engines, incorporating a scaled advanced aircraft compressor were studied with the objective of satisfying the rotor dynamics design criteria of 50% speed margin on stiff bearings and less than 25% rotor strain energy for soft bearing critical speeds. The single shaft direct drive concept (SSDD) and the single shaft free turbine concept (SSFT) would meet the critical speed criteria for both of the configurations studied.

The SSDD engine design would be a single shaft three bearing machine with eleven compressor stages and four turbine stages connected directly to an electrical generator. The SSFT engine design would be a single shaft three bearing gas generator with eleven compressor stages and two turbine stages. A separate 2 stage free turbine would drive the electrical generator system. The free turbine would be the same one which has been designed for the TSFT dual spool engine.

The SSDD design would meet all the critical speed criteria shown in Table 8-10. The supercharged configuration would have a 49% stiff bearing margin which would be just below the required 50%. This would be acceptable.

TABLE 8-10. CRITICAL SPEED CRITERIA COMPARISON

	Base			Supercharged		
	TSFT	SSDD	SSFT	TSFT	SSDD	SSFT
Stiff Bearing (% Margin)						
1st Low Rotor	45*	57	71	43*	49*	53
2nd Low Rotor	60	88	100	49*	89	88
1st High Rotor	40*	--	--	40*	--	--
2nd High Rotor	104	--	--	104	--	--
Soft Bearing (% Rotor Strain Energy)						
1st Low Rotor	23	12	15	22	12	15
2nd Low Rotor	36*	23	22	36*	22	20
1st High Rotor	8	--	--	7	--	--
2nd High Rotor	20	--	--	20	--	--
Low Rotor Speed	3969	3600	3570	4051	3600	3740
High Rotor Speed	4496	--	--	4574	--	--

*Does not satisfy critical speed criteria,

The SSFT design and the supercharged configuration would meet all the critical speed criteria.

Table 8-10 shows the comparison of the critical speed analyses of all the single shaft configurations discussed above with the dual spool engine. The TSFT at that time did not meet all of the critical speed criteria, but with continued work, acceptable rotor dynamics were obtained.

The single rotor gas generator systems of the SSDD and SSFT would be inherently simpler from a critical speed standpoint than the dual spool TSFT design.

Bearings

For optimum 4th stage turbine blade tip clearance control, the SSDD thrust bearing is located at the front of the compressor. Using TSFT temperatures and thermal response rate estimates as a base, the single shaft engine temperatures and thermal response rates for the 4th stage turbine blade transient and steady state tip clearance were estimated. The results are shown on the curve in Figure 8-9. The pinch point is shown at steady state conditions.

If the thrust bearing were to be located at the turbine exhaust case, axial thermal transient growths (rotor relative to cases) would not be a factor. Therefore, the pinch point would be twelve minutes into acceleration, requiring an increase in gap at steady state of 39 mils.

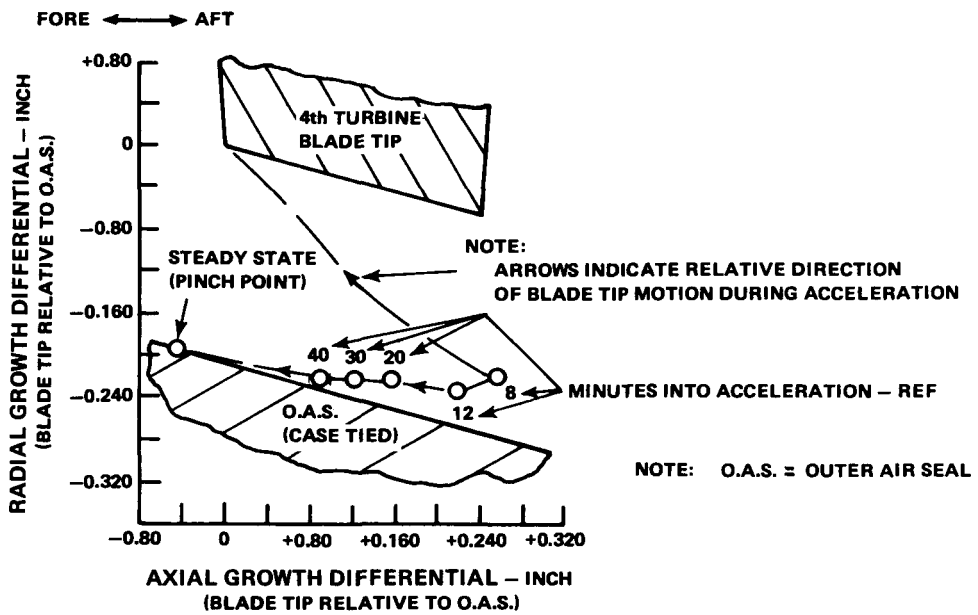


Figure 8-9. Turbine Blade Thermal Transient Tip Clearance Response

Deceleration would not be limiting for either thrust bearing location, since relatively large airfoil radial growth reduction would be almost instantaneous.

Radial bearings (tilting-pad type) were designed for the SSDD and SSFT industrial engine studies. The designs met all design criteria. The SSDD engine scheme from the bearing viewpoint was the most attractive for the following reasons:

- least amount of bearings required
- lowest values of unit loading and surface speed
- lowest power loss and oil flow requirement.

Industrial engine design practice was employed in the design of the hydrodynamic bearings for the SSDD and SSFT industrial engine studies. All bearings would be of the tilting-pad type to allow for a basis of comparison between the three engine designs.

Table 8-11 illustrates in tabular form the comparison of the three bearing designs. The major advantage of the SSDD and SSFT designs over the TSFT, which is not evident from the table, was that an intershaft bearing would not be required. The intershaft bearing design in the TSFT was questioned primarily due to being outside

of our industrial experience and, difficulty in analytical design. Consequently, both the SSDD and SSFT designs were favored over the TSFT.

The SSDD would definitely be the more attractive of the two single shaft designs. The pertinent design parameters, unit loading and surface speed, would be well within the state-of-the-art; only two journal bearings and a journal/thrust combination bearing would be required; the oil flow requirement is down 20% from the SSFT resulting in an appreciable cost savings in the lubrication system.

TABLE 8-11. BEARING DESIGN COMPARISONS

	TSFT	SSDD	SSFT
No. of Bearings			
Journal	8	3	5
Thrust	3	1	2
Maximum Unit Load			
Radial	260 psi*	193 psi	260 psi*
Thrust	346 psi	325 psi	346 psi
Maximum Surface Speed			
Radial	286 fps	283 fps	265 fps
Thrust (@ O.D.)	440 fps**	320 fps	360 fps
Total Power Losses	792 HP	599 HP	735 HP
Total Oil Flow	440 gpm	329 gpm	406 gpm

*Exeeds design criteria (F/T rear journal bearing)

**Exceeds design criteria (thrust bearing - front HPC)

Thermal Gradient Analysis

A preliminary analysis of the SSDD/SSFT disks was conducted using estimated thermal gradients. The following materials would be satisfactory for the SSDD engine based on this analysis:

- IM3005 - Limited to temperatures below 600°F, it would appear to be suitable for compressor stages 1-6 and turbine stage 4.
- IM3000 - Limited to temperatures below 850°F, it would appear to be suitable for compressor stages 7-11.
- IM152 - Limited to temperatures below 850°F, it would appear to be suitable for the 2nd and 3rd turbine stages and a candidate for the 1st turbine stage provided suitable rim cooling is possible.

The front compressor stages would be more highly stressed than the TSFT LPC stages. These stages were limited in burst and fracture mechanics stresses.

The first turbine stage would have bore stresses slightly in excess of the fracture mechanics allowable of 90 ksi and a low rim life (5000 cycles). The TSFT HPT was similarly stressed.

The second turbine stage would be stressed to the same levels as the first stage, and accordingly, would be stress limited at the bore and rim. TSFT disk stresses were substantially lower.

Transient gradients were unavailable for Stages 3 and 4 of the SSDD engine and the power turbine of the TSFT. Disks, however, would appear to be comparably stressed at steady state.

Industrial Duty Cycle for SSDD Engine

In order to minimize acceleration gradients in the turbine and compressor disks, a recommended start-up cycle shown in Figure 8-10 would consist of:

- ½ minute cranking
- 7½ minutes light-off to synchronous idle
- 5 minutes synchronous idle to peak

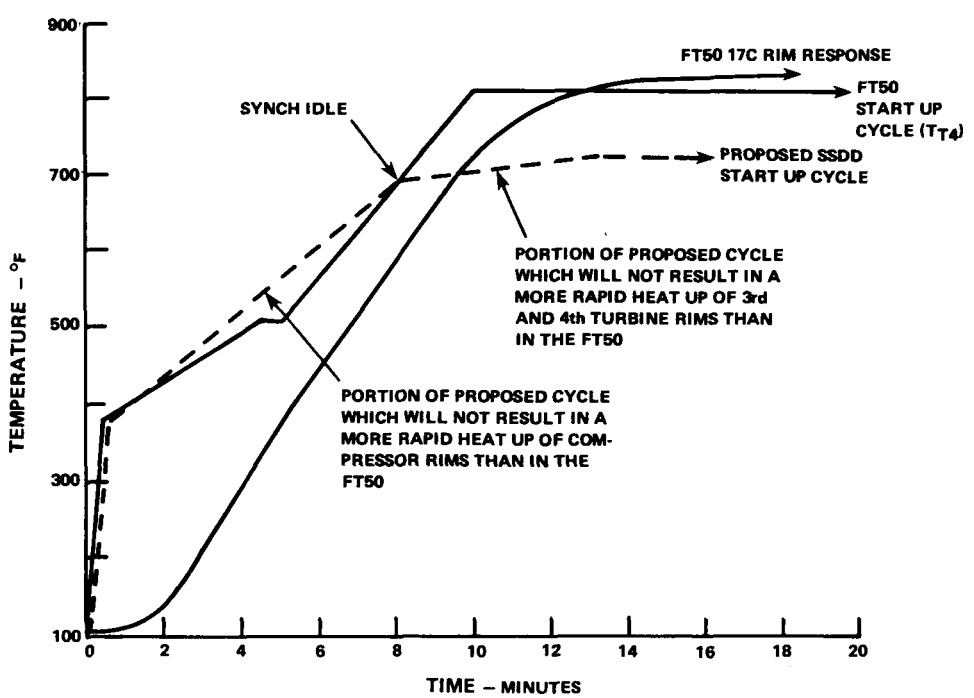


Figure 8-10. Comparison of Proposed SSDD Acceleration Schedule with Respect to Present FT50 Schedule

The 7½ minutes to synchronous idle limits gradients in the compressor and turbine stages 1 and 2. The five minutes spent accelerating to peak power would limit gradients in stages 3 and 4.

In order to minimize deceleration gradients, the following deceleration schedule shown in Figure 8-11 was recommended:

- Five minutes peak power to synchronous idle (five minutes to gas generator idle for TSFT
- Five minutes synchronous idle to gas generator idle.

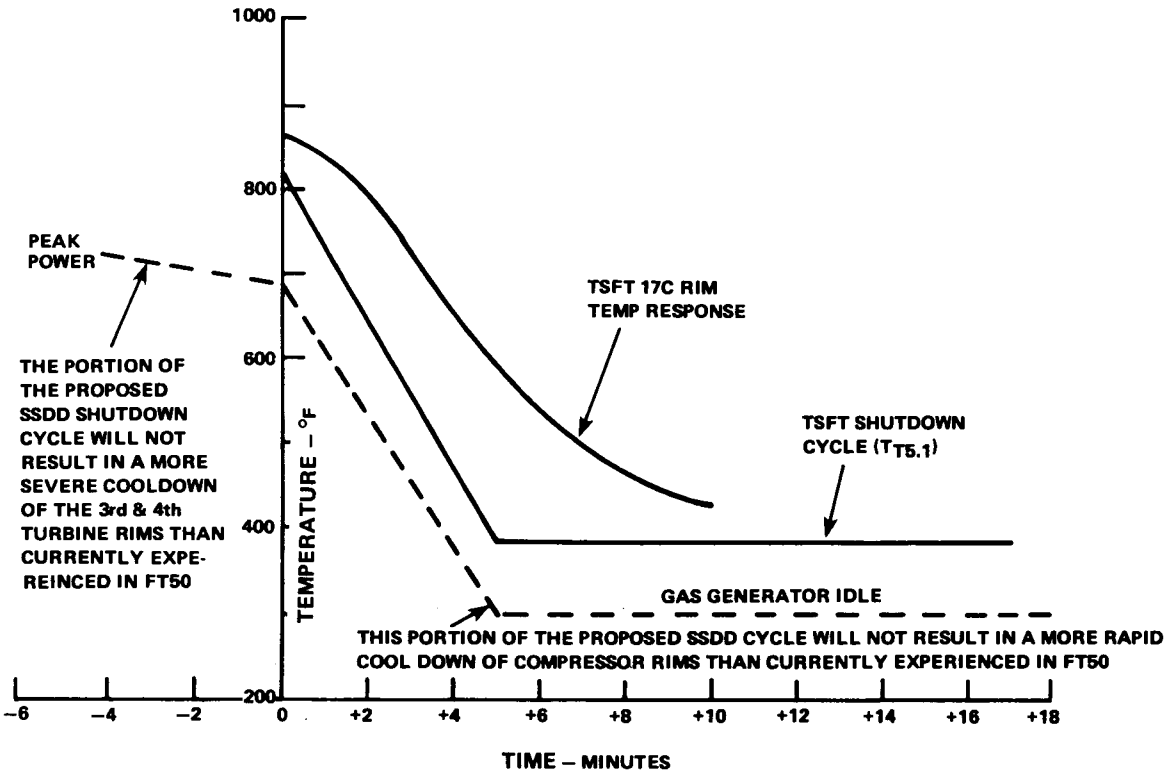


Figure 8-11. Comparison of Proposed Decel Schedule for SSDD with Present FT50 Deceleration Schedule

The five minutes deceleration to synchronous idle would limit stresses in the turbine Stages 3 and 4. The five minutes spent decelerating to gas generator idle would limit gradients in the compressor and turbine stages 1 and 2.

Due to the similarity of the SSFT and SSDD performance tables, the same disk problems would be expected throughout the engine, that is, most of the disks would be bore, bolt hole, or rim stress limited, rather than limited by burst. Therefore, disk weights and sizes could not be based only upon the SSFT burst requirements since, in general, the disk would be larger. The actual sizes could not be estimated without a transient analysis, but would be between the SSDD disks scaled to the SSFT and disks sized for SSFT burst requirements. The free turbine stages should correspond to the TSFT free turbine.

Due to similiarity to the basic design of the TSFT, the SSFT industrial cycle should also correspond to the TSFT.

Airfoil Vibration

The structural evaluation of the blades and vanes of a single shaft industrial engine was completed for a scaled Advanced Aircraft Compressor (AAC). The following comments were made as a result of this evaluation:

The first compressor blade in the base engine would have an airfoil root combined P/A and untwist stress of 75,000 psi at 3600 rpm, assuming a steel blade. To reduce the stress to an allowable level (65,000 psi) would require a 20-25% increase in aspect ratio. A material change to titanium would reduce the stress to an allowable level in the present scaled aspect ratio. The supercharging rotor would also require the use of titanium or a substantially higher aspect ratio.

Erosion and FOD resistance of compressor leading and trailing edge radii scaled from the AAC would meet the requirements established for the TSFT for protection against erosion and FOD.

Variable cantilevered stator construction represented an increase in structural risk compared to the TSFT. There was limited experience with this type construction, and the vibrational response, i.e., damping, wear, etc., was likely to require extensive development. It was not clear that the AAC development program would offer solutions to the structural problems associated with these stators since the operation of the two compressors would differ in many ways.

Blade and vane tip chordwise bending resonances would exist below redline speed for 7 of the blades and 3 of the vanes. In the free turbine configuration, several of the resonances would exist in the estimated gas generator idle speed range.

The airfoil tips (blades at the O.D. and vanes at the I.D.) would have to be thickened to avoid resonances at idle. In the direct drive configuration, these "tip modes" would result in resonances in speed ranges of transient operation only. In either case, the structural risk would be higher than in the TSFT, where the blades and vanes were designed with the "tip mode" resonances above redline speed to avoid possible tip fatigue failures.

The scaled AAC 1st, 2nd, and 3rd stage compressor blades would have 10E resonances at 3600 rpm. This would be an unacceptable situation with the original 5 strut configuration. For this reason, the number of struts was increased to six.

No new serious problems were anticipated in the turbine blades. Minor airfoil modifications might be necessary to "tune" the natural frequencies of the blades, but the chords as shown should be structurally adequate. The elimination of the turbine intermediate strut in the direct drive configuration would be a real plus for that system, since it is inevitable that such a strut would introduce flow disturbances which would excite the blades fore and aft. The synchronous idle condition in the direct drive engine was to be favored over that of the free turbine configuration since the last stage blade tip incidence angle was much improved. In the free turbine engine, as with the the off-design incidence was beyond our experience and cascade tests were recommended to assure no tip separation exists.

Materials

The materials selected for the single shaft design were kept as close to those of the twin spool as temperature and stress requirements permitted. They are summarized for the engine components in Table 8-12.

The front compressor case was changed from a wrought structure to a casting to accommodate the numerous holes for the variable stators.

The rear compressor would be fabricated from machined rings produced by flash butt welding or fusion welding and forming. The outer case would be cast or wrought based on cost.

TABLE 8-12. PRELIMINARY MATERIAL SELECTION FOR SSDD AND SSFT

	TSFT		SSDD & SSFT
	Low Rotor	High Rotor	
Cases	ASTM A515 Gr 60	ASTM A217 GR WC6	(Over stages 1-4) ASTM A216 GR WCA (Over stages 3-11) ASTM A387 GR B
Disks	PWA-IM 3005	(1st) PWA IM-3005 (2nd-8th) PWA-IM 3000 (9th-10th) FV 535	(1st-5th) PWA-IM 3005 (6th-10th) PWA-IM 3000 (11th) FV 535
Vanes	(IGV-3rd) 2389-05 (4th-7th) PWA-IM 4010	PWA-IM 4010	(IGV-2nd) 2389-05 (3rd-11th) PWA IM 4010
Blades	(1st-3rd) 2389-08 (4th-7th) PWA-IM 4010	PWA-IM 4010	(1st) PWA 4928 (2nd) 2389-08 (3rd-11th) PWA-IM 4010
Tie Rod	AMS 6415		AMS 6415
Rear Hub		PWA-IM 3000	PWA-IM 3000
Combustor		Hast X	Same as FT50
Turbine			
Vanes	X-45	X-40	1st X-45 & 2nd X-40
Blades	IN 738	IN 738	1st X-40 2nd IN 738
Disks	M152	M152	1st & 2nd M152
Case	ASTM A387 GR B	ASTM A387 GR B	ASTM A387 GR B
Free Turbine			
Vanes	X-40		X-40
Blades	1st U-500 2nd IN 718		1st U-500 2nd IN 718
Disks	PWA-IM 3005		3rd SSDD M152 3rd SSFT PWA-IM 3005 4th SSDD & SSFT PWA-IM 3005

1st stage compressor blades for the single shaft would be Titanium 6Al-4V (PWA 4928) because of higher untwist stresses than the TSFT.

Rim temperature of the SSDD third turbine disk would be higher than the TSFT by 100°F, and M-152 was selected for this disk in place of PWA 3005.

Secondary Flow

The secondary airflow for the single shaft engine in total (excluding airfoil cooling) would be 0.5% WAE and 0.2% WAE lower than the TSFT for the direct drive and free turbine versions, respectively. Since the individual cooling air sources were not the same, the lower cooling flows would not necessarily mean a performance benefit relative to the TSFT.

Secondary flow differences relative to the TSFT are:

	TSFT	Single Shaft Direct Drive	Single Shaft with Free Turbine
Turbine Seal Leakage and Cooling	Base	+0.12% W_{AE}	+0.12% W_{AE}
Bearing Flows & Rotor Drum Cooling	Base	-0.15%	-0.32%
Thrust Bearing Leakage	Base	-0.45%	0%

The SSDD diffuser vent pipe size would be 35% greater than the TSFT since both compressor disk bore and turbine disk bore cooling air would be vented from the diffuser compartment. The single shaft free turbine diffuser vent area would be almost 50% smaller than the TSFT due to the venting of only one bearing compartment and dumping of the thrust balance air to the free turbine.

The SSFT design would have the same 50,000 LBF load in the gas generator and free turbine as in the TSFT. The SSDD thrust load would be 17,800 LBF.

The bearing compartment protection systems for the single shaft designs were equivalent to the TSFT since they would utilize buffer air at the same temperature as the FT50.

Due to their increased size, the single shaft designs would be more difficult to seal relative to the FT50 for the following reasons:

- In both single shaft versions, the flow between the first and second stage turbine disks would be 0.3% W_{AE} greater than the TSFT due to a change in work split causing a higher pressure difference across the second vane inner seal and due to the increased size of the engine.
- Larger differences in axial thermal growth due to the greater distance from the thrust bearing to the turbine relative to the TSFT, would cause more difficulty in maintaining axial overlaps and good sealing.

The SSDD cooling system would result in the following disk temperature differences relative to TSFT:

	Relative Difference
Compressor	Disk Rims as much as 80°F < TSFT Disk Bores as much as 65°F > TSFT
Turbine	1st and 2nd Disk Rims 15°F and 60°F > TSFT, respectively. 3rd Disk Rim 100°F > TSFT in Direct Drive Version Only 1st and 2nd Disk Bores 300°F < TSFT in the Direct Drive Version and 265°F < TSFT in the Free Turbine Version

REFERENCE 8-6

COMPRESSOR

August 1972

Aerodynamic Study of the Application of a Scaled Advanced Aircraft High Pressure Combustor to Industrial Single Shaft Gas Turbine

Summary

An aerodynamic study was conducted to determine the suitability of using a scaled Advanced Aircraft Compressor (AAC) in a single shaft gas turbine proposed as an alternate to the TSFT. Comparisons of the AAC and TSFT compressors were also made so that their relative aerodynamic performance could be assessed using consistent methods. The following conclusions and observations were drawn from this study:

- The AAC compressor aerodynamics should not be significantly altered by the different inlet profile and bleed requirements of the industrial single shaft application.
- The compressor rig development required to optimize the different variable geometry schedule of the industrial AAC would not be substantial.
- The aerodynamic loading (D factor) requirements of the AAC single shaft industrial compressor and the TSFT compressors were essentially equal.
- The TSFT compressor aerodynamic design parameters were within our experience whereas the AAC exit hub/tip diameter ratio would not be. The AAC rig program with adjustable strators would diminish this risk and no analytical penalty was assumed.
- The direct drive industrial AAC compressor was 0.9 percentage points lower in adiabatic efficiency than the TSFT at their respective design speed operating points.

Discussion

AAC Modifications

Although the scaled AAC compressor would be aerodynamically suitable for the industrial engine, differences arising from this new application were noted as potential risks.

Inlet Profile

The AAC was designed as a high pressure compressor of a two spool engine and consequently would have a 3.0 percent total pressure profile (greater % at O.D.) and a non-axial 60° swirl angle at its inlet. The industrial AAC engine would have a flat inlet total pressure profile and axial flow requiring a new inlet guide vane which would cause a different radial match in the front stages of the compressor.

Bleed

The aerodynamic design of the AAC compressor would be adjusted to account for the steady state bleed locations and quantities of the AAC engine. The bleed requirements for the industrial single shaft would be different and constitute an aerodynamic change to the compressor.

Match

The aerodynamic matching effects of the inlet profile and bleed changes were computed using the AAC design geometry in an analytical performance prediction system (AFCC) which included cascade tunnel data and end wall effects. This system indicated that the industrial version would overflow the AAC slightly (~0.4 percent) with about the same efficiency at 14.0 pressure ratio. Internally, the inlet profile change would have a negligible effect on aerodynamics. The bleed change would increase loading (D factor) in the 6th through 9th stages by about .01 and move incidence towards stall by less than a degree. These changes were small aerodynamically and required only small changes of the adjustable stators to be acceptable.

Variable Geometry Schedule and Maps

Maps were generated for single shaft free turbine and direct drive compressors and a typcial map is shown in Figure 8-12. Changes in surge line and speed/flow near 90 percent speed on the AAC map were based on a less severe vane angle opening versus speed relationship than the aircraft engine match. In addition, the direct drive version would lock stators above 90% speed to prevent power fluctuations and the free turbine version would continue opening beyond nominal. The changes in surge line, speed/flow and efficiency/flow relationships were incremented using trade factors from rig testing.

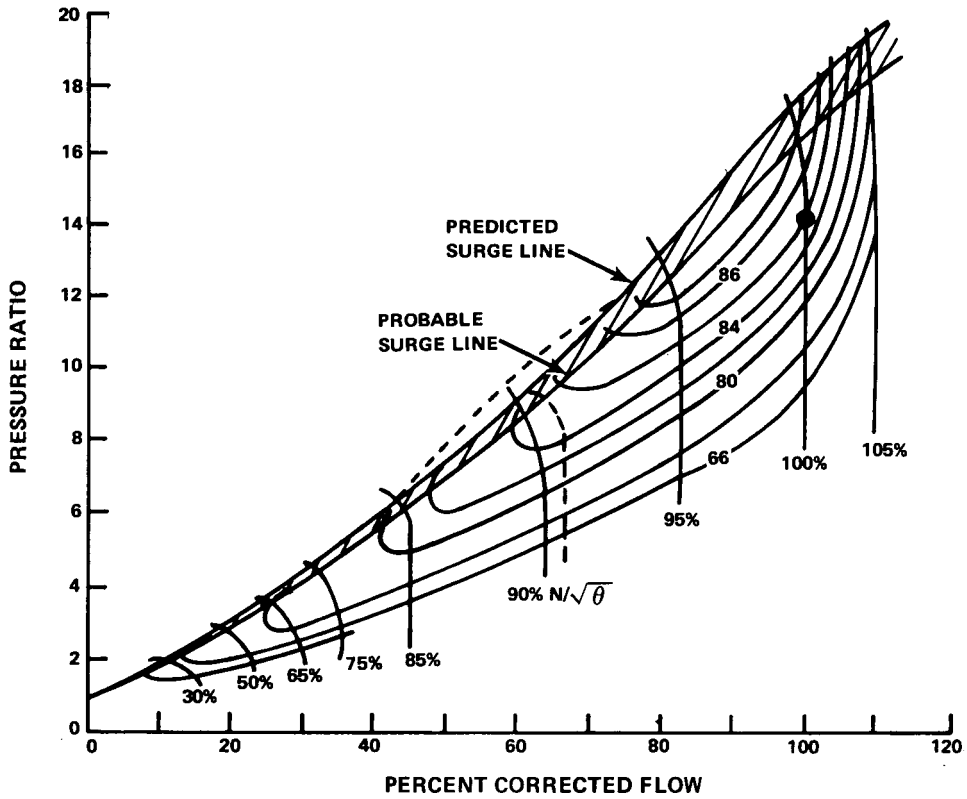


Figure 8-12. AAC Compressor Characteristics With Industrial Engine Vane Schedule

Comparison of the Industrial AAC to TSFT

The TSFT and AAC compressors were designed consistently with the same analytical system and blading philosophy and differed only in the selection of parameters resulting from their different program requirements.

A general comparison of the TSFT and scaled AAC compressors is presented in Table 8-13. The scaled AAC would be lower in pressure ratio and efficiency than the TSFT and have fewer stages and airfoils. The lower predicted efficiency of the AAC was a consequence of the tighter gap/chord, higher Cx/U and higher exit Mach number of that compressor. The reduced number of stages resulted from these parameter differences as well as the higher D factor of the AAC. The lower average aspect ratio of the AAC also contributed to the reduction in number of airfoils. The reduced number of airfoils and number of stages would lead to lower cost and greater reliability.

TABLE 8-13. COMPARISON OF TSFT AND INDUSTRIAL COMPRESSORS

Parameter	TSFT LPC	TSFT HPC	TSFT Overall	Scaled AAC*
Corrected Flow (lbs/sec)	766	240	---	710.
Corrected Speed (RPM)	3522.0	3407.0	---	3600.
Pressure Ratio	3.8	4.18	16.0	14.3
Adiabatic Efficiency (%)	91.1	89.5	88.2	87.3
Discharge Mach Number	.46	.27	.27	.30
Average Mach Number	.649	.567	.608	.711
Number of Stages	7	10	17	11
Hub/Tip Diameter				
Inlet	.541	.790	---	.614
Exit	.790	.902	---	.921
Gap/Chord (Average)	.95	.95	---	.791
Aspect Ratio (Average)	2.00	1.49	1.69	1.46

*Scaled up from and advanced aircraft engine high pressure compressor.

Loading

The emphasis in choosing the loading in the TSFT was to maintain efficiency and minimize risk. Design parameters were selected to benefit efficiency, but were within established experience. The AAC goals would exceed experience in surge line loading and exit hub/tip ratio. The surge margin goal for the original AAC compressor would be 30 percent, exceeding the surge margin required for an industrial single shaft engine. A comparison of the surge line and operating line loadings of the TSFT and industrial AAC showed that although the AAC had a greater D factor on the operating line than the TSFT, the surge line loading extremes at high and low flow point would be less than that of the TSFT. The AAC variable geometry would permit nearly equal balancing of high and low flow leading distributions whereas the TSFT had no such control over the same flow range (100 to 70% flow) and had to compromise its match between the two.

Efficiency

The AAC efficiency would be 0.9 percentage points less than that of the TSFT. The difference was based on predictions for both compressors using the same analytical system. Cascade, Mach number, Reynolds number, and tip clearance effects calibrated with PWA experience were consistently included for both compressors. The predicted efficiency of the scaled industrial single shaft AAC would be greater than that of the original AAC because of relatively smaller tip clearance effects. The scaled AAC had 0.4 points non-standard leading and trailing edge radii penalty debited relative to the TSFT.

Hub/Tip Ratio

The AAC exit hub/tip ratio would be higher than the general experience level of 0.9. There was no apparent aerodynamic reason why this difference should be detrimental to the AAC. Although wetted annulus area/flow area increases for a unit axial length as hub/tip ratio increased, chord would be decreased for a given aspect ratio making total wetted surface the same. However, a given tolerance on dimensions would be a larger percentage of the shorter high hub/tip ratio span which would produce a possible penalty in surge margin and efficiency.

REFERENCE 8-7

COMPRESSOR

March 1974

Aerodynamic Study of the Application of a Supercharged Advanced Aircraft High Pressure Compressor to an Industrial Gas Turbine

Summary

The August 1972 report on the application of a 14:1 pressure ratio 11 stage AAC compressor to a single shaft, industrial gas turbine for 60 cycle electric power generation was updated. The major difference was assuming the addition of a supercharging stage to the compressor and the removal of two compressor rear stages which would increase the pressure ratio to 14:5 and the power from 81 MW to 117 MW.

Discussion

At the time of the previous study, the initial AAC compressor had not been tested. Its design exceeded Pratt & Whitney Aircraft experience in surge line loading and exit hub-tip ratio and was, therefore, considered a risk to achieve the predicted efficiency and surge margin. Between the two reports two tests of the rig took place, the latest one showing about 1% below predicted efficiency and nearly matching the predicted surge line. This resulted in a higher degree of confidence in the predicted performance for the proposed level of compressor technology.

The Supercharged Advanced Aircraft Compressor (SAAC) would add a new stage to the front of the initial configuration and revise the flowpath and blading of the first two stages of the original compressor. Although there was growing confidence in the design system used for the AAC compressor, the higher surge line loading and exit hub-tip ratio in addition to the new high pressure ratio front stages make this compressor a greater risk than the TSFT compressors.

The single shaft engine would incorporate a scaled up SAAC compressor with the 2 rear stages removed and matched at 15.5:1 pressure ratio and 100% flow. A description of this compressor and a tabulated comparison with the TSFT compressors is shown in Table 8-14.

Mechanically, the engine would be similar to the previous study except that it has only 10 compressor stages and was updated to reflect the latest experience of the TSFT design. It was also revised to drive through the front or inlet end of the

engine. The front drive would allow complete freedom to optimize the exhaust system and would not compromise the compressor. The turbine discharge area was kept the same as the TSFT and the earlier single shaft direct drive engine to avoid excessive stresses.

Other design parameters such as D factor and operational characteristics did not change significantly from those described in the previous study of August 1972.

Efficiency

The selection of the compressor design point after consideration of the steady state surge margin resulted in an increase of adiabatic efficiency from 87.3% in the August 1972 report to 88.2% in the 1974 report.

TABLE 8-14. COMPARISON OF TSFT COMPRESSORS AND THE SCALED SUPERCHARGED ADVANCED AIRCRAFT COMPRESSOR

Parameter	TSFT LPC	TSFT HPC	TSFT Overall	Scaled SAAC (less 2 rear stages)
Corrected Flow (lbs/sec)	766	240	---	1033
Corrected Speed (RPM)	3522.0	3407.0	---	3600.
Pressure Ratio	3.8	4.18	16.0	14.3
Adiabatic Efficiency (%)	91.1	89.5	88.2	88.2
Discharge Mach Number	0.46	0.27	0.27	0.33
Average Mach Number	0.649	0.567	0.608	0.72
Number of Stages	7	10	17	10
Hub/Tip Diameter				
Inlet	0.541	0.790	---	0.567
Exit	0.790	0.902	---	0.916
Gap/Chord (Average)	0.95	0.95	0.95	0.791
Aspect Ratio (Average)	2.00	1.49	1.69	1.46

REFERENCE 8-8

COMBUSTOR

August 1972

Combustion Section for a Scaled Advanced Aircraft Compressor
Applied to an Industrial Single Shaft Engine

Summary

The combustor for the single shaft direct drive engine based on an advanced aircraft compressor would be essentially the same as for the TSFT. Using the same design criteria as for the TSFT, a canted can-annular design using eight 15.65" diameter cans was recommended.

Discussion

The single shaft engine combustion section would be a canted can-annular design consisting of eight 15.65" diameter, low emission burners with segmented transition duct section. The design criteria would be the same as that for the TSFT dual spool engine. The combustor section would be designed for 10,000 hour life goal between removals, ΔTVR goal of 1.2, with the initial engine to be designed for a ΔTVR not to exceed 1.3, and would be designed to achieve 1978 EPA emission regulations.

The burner and transition duct construction would be the same as the TSFT, which incorporated the shrouded louver concept. The louvered construction featured a low risk-proven cooling concept, low cost, and easy development.

The straight section diffuser for the single shaft engine would be 21" long, 3" longer than the TSFT diffuser. Both were designed for .14-.15 diffuser exit Mach number to minimize dump loss, but the higher diffuser inlet Mn (0.3Mn) for the single shaft engine would result in a longer, higher area ratio diffuser than that of the TSFT. 4" thick service-carrying diffuser struts were required in both engines.

The single shaft engine would have a dual-fuel fuel system designed for natural gas and No. 2 home heating fuel. The ignition system would used 2 ignitor cans and cross-over tubes.

The combustion section for the single shaft engine would have the same degree of development risk as the TSFT.

Table 8-15 presents the pertinent combustor section parameters.

TABLE 8-15
COMBUSTOR SECTION PARAMETERS

Peaking Power - 59° Day

	TSFT	SSDD	SSFT
Airflow - lbs/sec	695	710	780
Burner Inlet Mach No.	0.26	0.30	0.30
Diffuser Length	18"	21"	22"
Length Comp. Exit to TIGV	44"	54"	56"
Burner Section Total Pressure Drop	6.5%	6.5%	6.5%
Required Burner L/D	3.4	3.4	3.4
Burner Diameter	14.3"	14.8"	15.65"

REFERENCE 8-9

COMBUSTOR

March 1974

Combustion Section for a Supercharged Advanced Aircraft Compressor Applied to an Industrial Single Shaft Engine

Summary

The combustion section for the single shaft direct drive engine using a supercharged AAC was essentially unchanged from the combustor studied in 1972. Additional studies were completed on a low temperature engine adapated to residual fuel and a study of low BTU gasified fuels was made for the 1974 report.

Discussion

Performance studies for the low temperature dirty fuel engine resulted in choosing a turbine inlet temperature of 1600°F and either a pressure ratio of 15.5 or 12:1 for the single shaft engine. The 15.5:1 pressure ratio could be handled in the combustor designed for No. 2 home heating oil by only a nozzle change. The 12:1 pressure ratio would require a longer, larger diameter combustor.

The use of an external silo type burner was considered, but there was insufficient time to adequately define this type of combustor. A preliminary look at this burner indicated it could be very difficult to adequately cool the large liner surface areas required at the firing temperatures considered for this engine.

All of the single shaft engines studied would be adaptable to and could burn low BTU gasified fuels with modifications to the fuel system and with rematching. The out-of-the flowpath design of the combustors would enable changes to be made to the fuel systems with little impact on the engine design.

REFERENCE 8-10

TURBINE SECTION

August 1972

Turbine for the Advanced Aircraft Compressor Applied to an Industrial Single Shaft Engine

Summary

Turbine flowpaths for both direct drive and free turbine engines were considered. The requirement for high turbine efficiency led to the choice of four stage turbines for both the direct drive and free turbine engines.

Discussion

Flowpath

The turbine flowpath studies of the direct drive and free turbine versions of the single shaft gas generator showed that the number of turbine stages would remain unchanged from the four required in the TSFT. A three stage turbine for the direct drive engine was evaluated and eliminated from further consideration due to the performance penalty of 0.7 points in overall turbine efficiency. Other performance disadvantages of the three stage turbine would be increased tip clearance sensitivity due to the increased slope on the OD wall and higher exhaust losses due to the 20 to 25° additional swirl entering the diffuser.

The direct drive and free turbine gas generator flowpaths were compared to the TSFT. Both of the single shaft flowpaths would be out at a larger diameter than the TSFT. This was required to hold the wheel speed and velocity ratio at approximately the TSFT level, at 3600 rpm, while the TSFT has a 4200 rpm high rotor. The direct drive flowpath reflects an engine airflow of 710 lb/sec. The gas generator flowpath of the single shaft free turbine version would be generated by photographically scaling the first two stages of the direct drive turbine by 3.8% to a flow of 780 lb/sec. The free turbine flowpath would remain essentially the same as that of the TSFT.

The number of foils was estimated by holding the airfoil loading coefficient at the TSFT levels. The foil weights and pulls were estimated by accounting for size and chamber differences from the TSFT. The total number of foils was estimated to be 404 for the SSDD and 370 for the SSFT compared to 352 for the TSFT. The greater

number of airfoils was due to the higher average hub/tip ratio of the single shafts. The aerodynamics of the free turbine version showed that approximately 15° of swirl leaves the gas generator and enters the third vane. By counter-rotating the free turbine, the turning requirement of this vane could be reduced by twice the swirl angle, or 30°. The direct drive could not take advantage of this, so the number of third stage vanes would be increased to accomplish the additional turning.

Efficiency

Efficiency estimates of the single shaft engines were obtained by accounting for differences in aerodynamics and cooling flows from the TSFT levels. The first two stages of both single shaft engines would operate at higher efficiency levels than the TSFT. This resulted from the ability to optimize the work split between the two stages. The last two stages of the direct drive would operate at a lower efficiency than the free turbine version, since the third vane did not have the reduced turning benefit of counter-rotation. The net effect of the cooling and aerodynamics was that the overall turbine efficiency of the direct drive and the free turbine version were both slightly better than the TSFT. The cooling flows used in the study assumed about 1/2% more cooling flow for the single shaft engines than the TSFT. The airfoil cooling flows would be somewhat changed from the TSFT levels because of slight differences in the cooling air temperatures and the relative gas temperatures. The increased seal leakage in the cavity between the first and second disks resulted from the turbine flowpath being at a larger diameter and the ΔP across the second vane being up relative to the TSFT. The increased diameter implied that the sealing diameters were also increased. The increased ΔP was a result of the work split differences between the single shaft turbine and the TSFT. The best engine from a SFC standpoint would probably have more work being done in the first stage than the initial assumption indicated. This would lower the ΔP across the second vane and thereby reduce the seal leakage. Another benefit of another work split would be lower gas path temperatures to reduce the airfoil cooling requirements. Both of these benefits would be at the expense of turbine efficiency.

The thrust bearing would be located at the front of the engine. Initially, this caused some concern over the ability to hold tip clearance control in the last two stages of the direct drive turbine. The concern centered about the axial motion of the case relative to the blade tips. A preliminary study showed that the thermal response rates of the rotor and cases were similar in both the compressor and turbine minimizing the tip clearance problem. The response of the burner case was

found to be rapid. When this result was combined with the quick thermal response of the airfoils, it was concluded that the minimum clearance would occur at steady state and the thrust bearing should be located at the front of the engine.

REFERENCE 8-11

TURBINE SECTION

March 1974

Turbine for the Supercharged Advanced Aircraft Compressor Applied to an Industrial Single Shaft Engine

Summary

The turbine for the single shaft engine was studied for the SAAC. The turbine inlet annulus was increased to pass the increased flow of the supercharged engine, but the exit annulus area was held to TSFT levels to maintain the stress levels. This would increase the exit Mach number and exhaust pressure losses.

Discussion

In this study, emphasis was placed on reducing the number of turbine stages from 4 to 3. The reduced number of stages would permit a smaller number of airfoils to be used, reducing the turbine cost. The reduced number of stages would produce an estimated loss in turbine efficiency of 0.9% due to increased wall slope and decreased velocity ratio. In addition, the exit Mach number and swirl would be increased, which would cause an estimated 2% greater exhaust duct loss. The predicted losses were estimated to offset the potential cost advantage of the 3 stage turbine and the 4 stage turbine was recommended.

REFERENCE 8-12

RELIABILITY

August 1972

Reliability Considerations Pertaining to the Single Shaft Direct Drive/Free Turbine Engine Designs and the TSFT Design

Summary

Improved reliability and lower operating cost per hour would favor the single shaft configuration over the twin shaft configuration. Both single shaft configurations would have four turbine stages and all combustion systems would be similar which tends to minimize the differences between the engines.

Between the two single shaft designs, the free turbine configuration was the one preferred. The reasons for this preference were the lower maintenance material cost, the lower starting torque requirements, and the higher power rating of the free turbine configurations which would lower the mils/kw-hr.

Discussion

Elimination of bearings, seals, and their associated lubrication requirements would heavily favor the single shaft design. While the free turbine design would have two more bearings than the direct drive, it still showed an improvement over the TSFT. The reliability comparison of the various engines is presented in Table 8-16.

The single shaft configuration reduced barring complexity by eliminating the need for a clutch system in the TSFT.

Since the three candidates all would have cantilevered vanes, this feature was not considered in the assessment. The variable geometry in the single shaft design represented an increase in complexity. However, in view of the heavy industrial construction, there would be no reason to believe they would not be reliable.

Historically, there have been problems associated with exit to entrance twin spool interaction. The single shaft design would preclude this type of problem.

TABLE 8-16. PRELIMINARY RELIABILITY COMPARISON OF TSFT, SINGLE SHAFT DIRECT DRIVE, SINGLE SHAFT FREE TURBINE

Eng. Section	Failure Rate/10^6 Hrs TSFT	SSDD	SSFT	Remarks
Bearings, Seals,	16.5	7.5	12.0	SSDD has the fewest bearings, seals, and least complex lube system. SS design eliminates the barring clutch.
Compressor	19.6	13.5	13.5	SS has one less compressor. SS cantilever vanes no more risk than TSFT HPC. Variable vanes should be within the state-of-the-art.
Burner	35.5	35.5	35.5	Low NO_x burner common to all 3 designs.
Turbine	31.0	31.0	31.0	Primary problem is sulfidation, common to all 3 designs. SS takes cooling air through disks, TSFT 2 TOBI systems. Considered equal.
Main Static Structure	23.3	14.3	21.8	SSDD eliminates 2 intermediate cases, SSFT eliminates only 1.
Power Turbine	10.8	10.5	10.8	SSDD eliminates 1 radial bearing.
Non-Chargeable	32.7	32.7	32.7	No anticipated change.
Total Failure Rate/ 10^6 Hrs.	169.4	145.0	157.3	
Mean Time Between Failures	5,900	6,900	6,350	Primary difference in reliability due to the bearings, compressor, and structures.

While the single shaft would have slightly larger burner cans, there would be no reason to presume any difference in burner life. The initial cost of the cans might be higher, but the cost of repair should be approximately the same. The fuel manifolds and nozzles were expected to have the same effect.

Turbine airfoil sulfidation and its impact on airfoil scrap rates would tend to control the Maintenance Material Cost of the engine. Since all three designs would have the same metal temperature, airfoil durability should be the same. The increase in number of airfoils would be offset by the decrease in cost, resulting in no appreciable difference in the three designs. A summary of the Maintenance Material Cost is shown in Table 8-17.

The single shaft cooling scheme would require taking the air through holes in the disk and diaphragm. While this would not be a plus for reliability, it had been demonstrated on the JT9D. The TSFT utilized a double tangential on-board injection system for turbine cooling flow management. These two systems were considered even.

The single shaft direct drive design would eliminate both the compressor and turbine intermediate cases and their bearing support systems. While the free turbine single shaft design would have a turbine intermediate case, it still had an advantage over the TSFT design.

From a Maintenance Material Cost standpoint, the turbine, burner, and power turbine sections for all three designs would have cost approximately the same per operating hour. The single shaft design would have an advantage in the compressor and bearing area since the cost of replaceable cold section parts would be approximately 35% less.

The higher megawatt ratings for the single shaft engines would further reduce the cost when calculated in mils per kilowatt hour.

TABLE 8-17. PRELIMINARY MAINTENANCE MATERIAL COST COMPARISON OF TSFT, SINGLE SHAFT DIRECT DRIVE, AND SINGLE SHAFT FREE TURBINE

Eng. Section	Cost $/Oper. Hour			Remarks
	TSFT	SSDD	SSFT	
Turbine Airfoils and Seals	8.71	8.70	8.70	Average scrap life the same for all 3 designs. Increase in total number of airfoils in SS design offset by decrease in cost.
Remainder of Hot Section	5.30	5.30	5.30	Part lives and costs about the same in all 3 designs.
Remainder of Gas Generator	6.44	4.19	4.19	SS replaceable gas path parts and bearings approx. 65% the cost of TSFT.
Power Turbine	0.73	0.73	0.73	Increase in total number of airfoils offset by decrease in cost.
Total Cost Per Operating Hour	21.18	18.92	18.92	Lower cost per operating hour due to lower cost of cold section and bearings.
Peak Power Rating NEMA-MW	68.5	71.2	74.1	
Cost Mils/kW-Hr	0.309	0.266	0.256	Differences in MMC for the SSDD and SSFT are due to the Peak Power Rating.

REFERENCE 8-13

MAINTAINABILITY

August 1972

Maintainability Considerations of the Single Shaft Gas Turbine Designs and the TSFT

Summary

From a direct maintenance cost standpoint, the single shaft/free turbine would be the preferred configuration since it had the lowest predicted maintenance cost. The single shaft/direct drive version had the next lowest maintenance cost.

The 1972 design evaluation did not reveal a clear preference from an overall Maintainability standpoint which considered not only direct maintenance costs, but also the economic impact of reduced engine availability. A potential problem would exist in both single shaft designs in that some of the modules would exceed maximum shipping limitations and would require special shipping permits. Further study was recommended to determine if the economic impact of special shipping permits and delays in receiving spare modules was enough to offset the lower maintenance costs of the single shaft engines.

If the need for special shipping permits did have a significant impact on maintenance costs, it was recommended that the single shaft engines be designed so that the compressor and turbine could each be separated into two modules which would be small enough for shipment without special permits. Incorporation of this recommendation would permit making a definite choice from an overall maintainability standpoint for the single shaft engines, the free turbine version being the first choice.

Discussion

A preliminary evaluation was made of the relative maintainability potential of the single shaft/direct drive and free turbine versions of the TSFT engine in comparison with the baseline twin spool/free turbine version. The evaluation included considerations of the predicted maintenance costs and the impact on the modular maintenance concept of each of the configurations.

Maintenance

Table 8-18 shows the predicted direct maintenance costs for the three versions of the TSFT engine. The SSFT version had the lowest predicted maintenance costs for both material and labor.

TABLE 8-18. DIRECT MAINTENANCE COST COMPARISON

Engine Configuration	Maintenance Cost (mils/kW Hr) Material	Labor	Total	% Change from Base
Twin Spool Free Turbine (TSFT - Base)	.309	.065	.374	-
Single Shaft Direct Drive	.266	.067	.333	-11%
Single Shaft Free Turbine	.256	.064	.320	-14%

From a maintenance material cost standpoint, the turbine, burner, and power turbine sections for all three designs would cost approximately the same per operating hour. The single shaft design had an advantage in the compressor and bearing area since the cost of replaceable cold section parts was approximately 35% less.

The higher kilowatt ratings for the single shaft engines would further reduce the cost when calculated in mils per kilowatt-hour.

As indicated in Table 8-18, the maintenance labor costs were essentially the same for all three configurations. Both single shaft versions would retain many of the maintainability features of the base line engine. All would have the capability for borescope inspection and replacing hot section parts (including 1st and 2nd stage turbine blades) through the combustion case. There were, however, differences between the configurations in the man-hours required to correct unscheduled maintenance actions, as indicated in Table 8-19. The on-site maintenance labor and the predicted frequency of unscheduled maintenance actions would be less for the single shaft versions. This, combined with the higher power output of the single shaft machine, would counteract in terms of mils/kW-hr, the increased shop repair man-hours for the larger modules of the single shaft engines. Thus, the net effect was that all machines were approximately equal from a maintenance labor standpoint.

TABLE 8-19. AVERAGE MAN-HOURS REQUIRED FOR UNSCHEDULED MAINTENANCE

Configuration	Average Man-Hours Required: Module/Parts Replacement - On-Site	Module/Parts Repair - Shop	Total
TSFT (Base)	168.0	359.0	527.0
SSDD	151.0	530.0	681.0
SSFT	153.0	479.0	632.0

Modular Maintenance

Special emphasis was placed on assuring that all modules in the base line TSFT design could be shipped by truck without the need for special permits. Limits for over the road shipment without permit are 8'0" wide x 11'0" high x 50' long, with a maximum weight of 43,280 pounds. Tables 8-20 and 8-21 provide a listing of the modules considered for the base line and single shaft engines, respectively.

The SSDD design configuration had two modules - the compressor and the turbine. Both of these modules would exceed the weight limitation and the turbine would also exceed the width restriction. Thus, shipment of these modules would require special permits. In the SSFT design configuration, there would be three modules - the compressor, gas generator turbine, and the power turbine. The compressor module, which would exceed the weight and width limitations was the only module which would require a shipping permit. The full impact of the need for special permits could not be evaluated. However, if the need for special permits resulted in significant delays in delivery of spare modules, there would be a loss of availability and subsequent loss of revenue for the utility company or extra charges incurred for the purchase of power from another utility company.

TABLE 8-20. TSFT BASE ENGINE MODULES

Module	Width	Length	Height	Weight
Low Compressor	7'3"	8'7"	7'3"	29,100 lbs
High Compressor	6'0"	5'11"	6'0"	16,700 lbs
High Turbine	6'1"	6'1"	6'9"	4,600 lbs
Low Turbine	6'4"	4'8"	6'4"	4,700 lbs
Power Turbine	7'10"	10'0"	10'0"	34,300 lbs

TABLE 8-21. SINGLE SHAFT ENGINE MODULES

Module	Single Shaft Direct Drive				Single Shaft Free Turbine			
	Width	Length	Height	Weight	Width	Length	Height	Weight
Compressor	7'11"	12'2"	7'11"	47,000	8'2"	11'10"	8'2"	50,100 lbs
Turbine	10'8"	10'7"	10'8"	58,300	7'0"	10'0"	10'0"	25,500 lbs
Power Turbine	-	-	-	-	8'0"	9'10"	9'10"	34,300 lbs

It was noted that it was possible to design, at some increase in cost and complexity, the single shaft engine so that the compressor and turbine could each be separated into two modules which would be small enough to permit shipment without special permits. If either single shaft configuration was selected the need for these provisions should be assessed, taking into account the impact of shipping permits on maintenance costs and the cost to the utility for any delays in receiving spare modules when required.

Another alternative to avoid the need for special permits would be to dissassemble the engine stage by stage on-site and replace only those detail parts requiring replacement. This would be a poor solution from a maintenance standpoint since it would mean abandonment of the modular maintenance concept. Man-hours and elapsed time for on-site labor would increase, thus having an adverse effect on station availability to generate power. Piecemeal assembly also would increase the possibility of assembly error and would require a larger on-site spare parts inventory. Piecemeal disassembly could also have an adverse impact on the installation cost since a larger house might be required to provide an area for disassembly and storage of parts.

REFERENCE 8-14

CONTROLS

August 1972

Report on Controls and Operational Characteristics of the Industrial Single Shaft Engines Compared to the TSFT

Summary

Most of the control functions were similar for all three engines, but there were identifiable differences that could be significant and result in decidedly different operational characteristics. The free turbine engines would have slower response times in load pickup, and reach higher overspeeds during load drop. The direct drive engine would require higher starting power because of the added turbine and generator mass, but would be more responsive to load changes when connected to the grid.

Discussion

Control requirements relative to the TSFT that were identified as different for the Single Shaft Direct Drive (SSDD) engine included: (1) A stator vane control (SVC). This extra control function would use a speed, N, signal and a temperature, T_{t2}, signal to compute the $N/\sqrt{\theta_{t2}}$ which would then be used to schedule the stator vane angle. (2) The control would delete the gas generator speed protection since this function would be provided by the electric generator speed controller. (3) The control would delete the low rotor, N_1, and high rotor, N_2, speed sensing because of the single rotor.

Control requirements that were identified as different for the Single Shaft Free Turbine (SSFT) engine included: (1) A stator vane control which would be the same as for the SSDD engine except that more accuracy would be required in the power region, and (2) Only one rotor speed could be eliminated because the machine still had two shafts.

All other control requirements would be the same as for the TSFT.

Starting

The SSDD and SSFT engines were analyzed and the following Table 8-22 shows the estimates made for operation between "start" and full load based data using estimated rotor inertias:

TABLE 8-22. ESTIMATES FOR OPERATION

	TSFT	SSDD	SSFT
Time to ignition speed	30 sec.	30 sec.	30 sec.
Motor Size	500 HP	1500 HP	750 HP
Time from ignition to "Sync"	150 sec.	240 sec.	150 sec.
Time from "Sync" to full load	120 sec.	30 sec.	120 sec.
Total time full load	300 sec.	300 sec.	300 sec.

Thus, it was estimated that it would take 5 minutes for each engine to perform the emergency start. It was not known what the temperature levels and the surge margins would be during the start.

Steady State and Transient Performance

The steady state surge margin was assumed to be the same as for the TSFT. This margin would be sufficient to account for externally induced inlet distortion and normal levels of fouling and deterioration of the compressor. In addition to the steady state factors considered in determining surge margin requirements, the following transient factors should be considered.

The transient excursions for the load pickups for the TSFT, SSDD, and SSFT were computed and plotted on their compressor maps. As shown in Table 8-23, the free turbine engines would use up a large amount of the available surge margin as they accelerated from low power level. The remaining surge margin, however, would be sufficient to avoid surge during these transients.

TABLE 8-23. LOSS IN SURGE MARGIN DUE TO TRANSIENTS

	Loss in Surge Margin Due to Transients		
	TSFT	SSDD	SSFT
0-25% Load Pickup	16%	None	14.5%
75-100 Load Pickup	5%	3%	7.3%
Drop Load	Requires Compressor Bleed	None	None

Underspeed

During sudden load pickup, the engine could slow down momentarily before accelerating to constant speed. A comparison of the under speeds obtained during sudden load pickup operation is summarized in Table 8-24:

TABLE 8-24. UNDERSPEED COMPARISON

	TSFT	SSDD	SSFT
Inertia: J1, low spool	1132. slug-ft^2	5000.	3200.
J2, high spool	1023.	----	----
J3, power turbine	2050.	----	----
J4, generator	1675.	1675.	1675.
Overtemperature Allowed	±250°F	+100°F	+250°F
0-25% pickup underspeed	2%	under 2%	over 3%
75-100% pickup underspeed	2%	0.2%	1.6%

Note that that SSFT would not be able to meet the 35% load pickup requirements in the 0-25% region because the gas generator rotor inertia is too high. The inertia could be lowered by redesign of the rotor.

Overspeed

Overspeed could result from several causes, including drop load, fuel control malfunctions, and other mechanical failures.

There were other factors studied which could influence the degree of overspeed. For drop load conditions these factors included:

- Power level
- Gas generator inertia
- Driven inertia
- Minimum fuel level
- Bleeds
- Generator windage and friction losses
- Gas generator efficiency lapse rates
- Free turbine compressoring
- Trip setting for failed control operation

For the single shaft application study, specific configurations were chosen for the purpose of comparison, thereby establishing relative overspeed criteria. For the rotor inertias shown above in the underspeed Table 8-24, the following Table 8-25 shows overspeed results obtained:

TABLE 8-25. OVERSPEED COMPARISON

	TSFT	SSDD	SSFT
Minimum f/a ratio	0.003	0.003	0.003
Bleed	11.5% Sta 3	None	None
Trip Speed Setting	12%	8%	12%
Overspeed:			
1. Normal Control	7.2%	2%	4.9%
2. Failed Control	15.0%	8%	14.3%
Recommended Overspeed Design Level	20%	10%	20%

The overspeed design levels were based upon the TSFT where 25% margin was added to the overspeed. The free turbine was set at the same overspeed design level as the TSFT.

The SSFT simulation produced less overspeed (during drop load) than the TSFT. Variable geometry compressor speed/flow characteristics accounted for the difference in results.

SECTION 9.0
BACKGROUND CONCEPTUAL DESIGN INFORMATION

This section includes three separate documents which were prepared during Phase I in anticipation of follow-on development of the new design during subsequent phases of the program. Their preparation represents an important part of the natural progression of the design process. They include:

- Gas Turbine Design Criteria
- Ancillary Systems Design Criteria
- Reliability Plan

Gas Turbine Design Criteria - To establish guidelines for design of the turbine engine specifically for its intended operating environment, a design criteria document has been drafted. This document defines structural standards, operating environment, fuels and emissions specifications, maintenance requirements, etc. It is intended to reflect not only the manufacturers' standards but also the utility users' requirements. Consequently, it is expected that the draft document formulated in conjunction with the initial conceptual design phase will be coordinated with EPRI during the preliminary design phase to follow.

The Gas Turbine Design Criteria is intented is to lay down specific guidelines for the engine designer to follow in converting a broad design concept to detailed layout drawings of individual parts to be fabricated. These guidelines reflect the intended application and as such specify environmental and installation requirements. In addition, detailed design practice for structural and aerodynamic limits and for tolerances or margins reflect accumulated past gas turbine operating experience. Application requirements should represent the needs of the utility operator as can be met by a practical design. Consequently, the design criteria document should, in its final form, include the collective requirements of the utility industry. This can be accomplished by review of the document included in this report, which can be considered to serve as an initial draft.

Ancillary Systems Design Criteria - In the conceptual design phase of the HRGT-CC plant, Design Criteria Documents were prepared covering the following systems which support the HRGT:

Fuel Supply System
Hydraulic Servo System
Lube Oil System
Start System
Ignition System
Water Wash System
Fire Protection System
Gas Turbine Controller

The Design Criteria Documents contain the design philosophy and criteria necessary to guide the system preliminary design. Included are requirements stemming from the other ancillary systems, an outline of the various operating modes, available services such as electrical, hydraulic, and pneumatics for control, normal and abnormal environmental requirements, structural and packaging requirements, reliability and maintainability criteria and specific design guidelines such as applicable design standards and codes, and specific requirements or exclusions stemming from past experience. The design critiera is specific for each system. However, many of the general guidelines and critieria are common for all the systems.

The Gas Turbine Ancillary Equipment Design Criteria serves a function similar to the "Gas Turbine Design Criteria, that is, as a guideline for the design of the equipment by specifying applications and environmental constraints on operating limits. However, in the case of ancillary equipment, the document will serve to guide the equipment vendors who supply the various ancillary system elements to the gas turbine manufacturer. In many cases, equipment component elements are already designed and manufactured. The design criteria in that case acts to guide the proper selection of component parts for the gas turbine ancillaries among those currently available.

Reliability Plan - To infuse reliability into the complete engine development program, an overall reliability plan was formulated to carry the effort from design through inital prototype testing. The plan is designed to be implemented through the program organization and integrated with the total program plan.

The preparation of Reliability Program Plan is a part of normal United Technologies' procedures in the initiation of a new engine design and development program. Its intent is to adapt to a specific program, existing general procedures and organizational structure for product integrity assurance. A reliability program plan is especially relevant for this program, which so strongly emphasizes high reliability in the equipment design. As is the case with the other documents, the reliability plan is a working document which, to be useful, must be subjected to continuous review, up-dating and refinement. Thus the document presented here as a byproduct of the Phase I effort should be considered as an initial draft to be modified during subsequent phases of the HRGT-CC program.

GAS TURBINE DESIGN CRITERIA

Contract RP1187-1

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SECTION I

INTRODUCTION

A. PURPOSE

The purpose of this Design Criteria Document (DCD) is to define the initial design philosophy and guidelines for the gas turbine in a combined cycle power plant for electric utility baseload application. The design criteria emphasizes reliability over functionality, performance and cost in keeping with the engine's intended application.

This Design Criteria Document, developed as a part of the Phase I effort, is subject to review and revision as the requirements and design evolve during subsequent phases of the program.

B. ENGINE DEFINITION

The High Reliability Gas Turbine (HRGT) will be designed to direct drive a 60 Hz A/C generator for electric utility base load application.

As an initial objective, the major features of the design will include:

- Modular construction to maximize availability by reducing maintenance time; rapid replacement of components.
- Single shaft direct drive configuration to minimize the number of parts. Single speed operation reduces vibrating failure modes.
- Cold end drive to eliminate the need for shaft seals in the hot gas discharge duct.
- Totally factory assembled engine substantially reduce on-site installation time and expense and enhance reliability.

<u>Compressor</u>

- Stiff, thick airfoils highly resistant to erosion, foreign object damage, and vibration.

- Rapid access for maintenance. Horizontally split case will be used to provide rapid access.

- A welded drum rotor. A short, stiff rotor shaft will be used with provision for field trim balancing to allow vibration-free operation.

Combustor

- Readily accessible combustor cans and fuel nozzle assemblies.

- A common annular combustor and transition section to provide more uniform turbine inlet temperatures.

- Adaptibility for distillate, residual, and coal-derived gaseous and liquid fuels.

Turbine Section

- Only three stages used for minimizing parts while used providing satisfactory efficiency.

- Compact rotor directly coupled to the compressor.

- Double wall case for durability and reliable tip clearance control.

- Rapid maintenance. Front stage vanes and blades are to be replaceable without engine disassembly. Entire turbine module to be removable as a unit.

Turbine Airfoils

- Airfoil metal temperatures are to be held to levels lower than present engines.

- Airfoil cooling will be provided by all-convective designs. No film cooling holes will be used.

- Cooling air from the compressor may be cooled externally to the engine before flowing to the airfoils.

- Airfoils to be constructed of conventional turbine alloys with hot corrosion resistant coatings applied.

Rotating Elements

- Rotor critical vibration modes will be as far as possible from operating speed.

- Journal bearings with high damping will be used for long life.

- Bearing/seal compartments will be located in lowest temperature regions. They will be directly accessible for maintenance.

Accessories

- All accessories will have external electric drives. There will be no gearbox or tower shaft system.

SECTION II

GENERAL REQUIREMENTS

A. GOALS OF ENGINE DESIGN

The primary objective of the program is to produce a high reliability gas turbine with significant improvements in maintainability and durability without compromising performance and cost.

To ensure the achievement of the reliability goals state-of-the-art industrial fabrication, conventional manufacturing methods and materials technology will be utilized. Low stress structural configurations will be implemented in the design (see Section V).

B. PERFORMANCE

The basic performance requirements for the gas turbine will be:

Power, MW	107 (ISO)
Engine Speed RPM	3600
Air Flow, lbs/sec	840
Pressure Ratio	14:1
Firing Temperature °F	2150
Simple Cycle Heat Rate (Btu/kWh)	10,700 (HHV)

C. FUTURE GROWTH

Parts will be designed so as not to preclude an upgrading of the engine. However, no special features will compromise the engine design in anticipation of this upgrading.

D. ELECTRIC UTILITY APPLICATION

The following application related general engine considerations will be used as a basis for design:

1. The gas turbine is one of two units in a combined cycle power plant for base load application.

2. The gas turbine will be operated continuously except for scheduled maintenance. However, nothing in the engine design should preclude use in an intermediate duty application where the load factor is 30 - 50%, or peaking duty where annual operation would be under 1500 hours.

3. Startup time of the gas turbine will be minimal in keeping with the primary application and the desire to maximize availability.

4. During operation the gas turbine will be operated at rated conditions but will have the capability of operating at partial power outputs.

5. During normal operation there will be no snap acceleration or deceleration imposed by the controls. Snap deceleration will be experienced during an emergency shutdown or drop load condition.

6. Operation with high ambient air temperatures is a requirement consistent with summer peak load conditions at many utility sites.

7. Operation at site altitudes up to 7000 Ft is a requirement.

8. Operation at very low ambient temperature is a requirement.

9. The gas turbine will be operated from a central plant control room. Operation will be fully automatic with provision for the operator to take manual control to avoid spurious shutdowns. The automatic controller will have protective circuits without overrides.

10. There are no limitations on dimensions except shipping and maintenance considerations.

11. There are no limitations on weight except as it impacts shipping individual modules or the entire assembled gas turbine.

E. INTERFACE

1. The gas turbine, with minor retrofit modifications, will be required to operate on a wide range of liquid and gaseous fuels. When fuels for the plant contain contaminants such as salts, metals, and other corrosives, a fuel treatment system will be incorporated to reduce contaminants to the gas turbine as much as practical. When necessary, atomizing air will be provided to assist combustion of heavier liquid fuels.

2. Inlet air will be filtered but may still contain residual dirt and salt.

3. Low levels of pollutants (NOx, smoke, etc.) are required consistent with the electric utility application. NOx control will be accomplished without steam or water injection.

4. Fuels, lube oil, hydraulic servo fluid, fire protection ignition, engine heat during shutdown water wash, and start provisions will be provided by systems external to the gas turbine.

5. Exhaust from the gas turbine flows to a waste heat boiler which imposes a back pressure on the gas turbine exhaust of about 12" H_2O.

F. DURABILITY

The hot section components will be designed to have a minimum life of 18,000 hours while the rest of the engine will have a minimum of 100,000 hours life. All engine components will have a minimum low cycle fatigue life of 12,000 cycles, and high cycle fatigue life of 10^8 cycles. The engine will have an overspeed requirement of TBD*%.

*TBD = To Be Determined

SECTION III

OPERATING REQUIREMENTS

A. Overall Specifications

	Goals
Unit Size, Net Electrical Output, MW	107*
Available for Commercial orders, year	1989
Nominal Capacity Factor, %	70
Heat Rate (Simple Cycle)	10700 Btu/kW/hrs) (HHV)
Availability (Full/Equiv.)	95/98
Operating Reliability (Full/Equiv.)	97/99
Starting Reliability	95
MTBF	9000 Hrs.
MTTR	TBD
O&M Costs (excl. fuel cost) mils/kWhr	1.0
Capacity Factor	TBD
Use Factor	TBD
Hours between major maintenance	18,000 hrs
Hot Section Parts Life	50,000 hrs
Cold Section Parts Life	100,000 hrs
Fuel Flexibility Dist. #2 & #6	yes
Dry NOx Capability for Dist. #2 & #6	yes
Firing Temperature (Combustor Exit)	2150°F max.
T_{metal} (hot spot maximum)	1500°F
Water Wash Capability (On-Line)	Yes
Cold Start, Time to Full Load	20 min.

*(To meet 300 MW Combined Cycle Requirement)

B. Fuel

The gas turbine will have multi-fuel capability including gaseous fuels with a heating value as low as 100 Btu/SCF, alcohols including methyl fuel and ethanol, and hydrocarbon liquid fuels including naphtha, petroleum distillates and selected crudes, residual fuels, coal derived and shale oil derived fuels containing fuel bound nitrogen. This multi-fuel capability may accommodate two or more changeable combustor geometries for the total range of gaseous and liquid fuels considered. Combustor technology and other technology improvements will be utilized to expand capability to heavier and higher nitrogen content fuels. Atomizing air will be available for burning heavier fuels. Fuel treatment will also be provided to minimize contaminants.

1. GASEOUS FUELS

The gas turbine will operate with gaseous fuels having a heating value of 100 Btu/SCF or higher. The fuel supply system will supply gaseous fuel at the following conditions:

(a) PRESSURE AND TEMPERATURE.

The gas supply pressure (measured at the burner case fuel connection) will be at least 50 psi above the maximum burner pressure and with a gas supply temperature between -40°F and 400°F.

(b) PARTICULATES.

Particulate matter in gaseous fuels will be filtered to minimize turbine deposits and erosion. A maximum limit of 0.01 grains/cubic foot is allowed. The maximum size distribution for the particulate will be 40 microns and the mean will be less than 25 microns. (This specification is subject to modification pending advanced technology programs).

(c) CONTAMINANTS

The total amount of sulfur and the alkali metals, sodium, potassium, and lithium will be less than required to form the equivalent of 5 ppm of alkali metal sulfates in the fuel. (This specification is subject to modification pending advanced turbine technology programs).

Sulfur content will be less than that which produces 150 ppm sulfur dioxide by volume in the turbine exhaust at 15 percent oxygen on a dry basis.

The gaseous fuel will contain no liquid constituents.

2. LIQUID FUELS

The gas turbine will operate with liquid fuels including methyl fuel and ethanol, naphtha, petroleum distillates, and selected crudes, residual fuels, coal derived and shale oil derived fuels containing fuel bound nitrogen. The fuel system will supply liquid fuel at the following conditions:

(a) CONTAMINANTS

Sulfur content will be less than that which produces 150 ppm sulfur dioxide by volume in the turbine exhaust at 15 percent oxygen on a dry basis.

The following list is a guide. (These specifications are subject to modification pending advanced technology programs).

Lead	1 ppm max.
Calcium	2 ppm max.
Sodium	1 ppm max.
Potassium	1 ppm max.
Vanadium	0.5 ppm max.
Ash	300 ppm max.
Water and Sediment	1.0% by volume
Particulate	Filtered to 40 microns max. with mean less than 25 microns.

(b) VISCOSITY

Maximum viscosity at the fuel nozzle will not exceed 20 centistokes with fuel heated to 250°F.

(c) TEMPERATURE

Temperature of the fuel at the nozzle will range from -10°F to 250°F.

C. Environmental Performance

The gas turbine will be designed so that during operation there is a minimum amount of degradation to the environment. It must, therefore, not exceed the standards specified in the latest "Standards of Performance for New Stationary Sources - Stationary Gas Turbines 40 CFR Part 60 Subpart GG." This EPA document will be continually updated and the gas turbine must be capable of conforming at all times.

1. Exhaust Emissions

(a) The gas turbine will be designed to obtain a non visible stack emission which includes the white smoke of unburned hydrocarbons. To be considered non visible on engine exhaust it will produce a Von Brand smoke reflectance number greater than 99 (at a sampling rate of 0.108 a scf/in^2 of Whatman No. 4 filter paper). This requirement will apply to engine operation at all powers except at start-up.

(b) Particulate Matter

Fuel contaminants including ash will be kept to a minimum by the fuel treatment system. Combustor soot will be kept to a minimum by the engine in order to limit the quantity of particulate matter in the exhaust. Design will be to the more restrictive of 10 lb/hour (max) or 0.1 grain/SCF (max.) or 0.1 lb/10^6 Btu heat input for 300 to 1000 million Btu/hr.

(c) Oxides of Nitrogen (NOx) and Sulfur (SOx)

United States Environmental Protection Agency limits as outlined in the "Gas Turbine New Source Performance Standard" can not be exceeded. All hydrocarbons and CO must be kept to a minimum.

(d) Odor - No emission of any substance will cause an objectionable odor.

2. Noise

Gas turbine emitted sound will be held as low as possible. Acoustical treatment applied directly to the surface of the engine or in the form of an easily removable hood is preferred. The acoustical treatment will be consistent with maintainability goals.

3. Impact Due to Failure

All components will be designed with due regard for personnel safety and environmental impact. All electrical components must be suitable for National Electric Code Class 1, Division 2, Group D Service.

D. Installation Environment

The gas turbine will be designed to operate in an environment typical of electric utility power plant installations. The external contaminants and ambient conditions for design are listed below:

1. Temperature - The engine will be designed for rated power at 59°F conditions. However the engine will be designed for operation over the range of -30°F to 120°F ambient temperature.

2. Altitude - The gas turbine will be rated at seal level but must be capable of operation at altitudes up to 7000 Ft.

3. Corrosive Protection Provisions (Gas Path Protection) - Inlet air filtration will be provided to protect the engine against performance deterioration and reduced life caused by erosion, corrosion and fouling. Filtration will be provided to insure that the following levels are maintained:

   Particle Concentration - (MAX) 0.1 grains/1000 SCF
   Particle Size - all particles < 10 microns

   The filtration design for each site will be established on the basis of good industrial practice and utility site requirements.

4. Cleaning Provisions - The gas turbine will be designed to accommodate water wash of the compressor, the turbine and the combustor nozzles. It is a goal of this design to minimize the frequency and engine outage time associated with water wash to 1000 hour intervals and 5 hours for the wash cycle including engine cooldown. During cool down and water wash the engine can be rotated by the turning gear at TBD RPM or the start motor and hydraulic torque converter at TBD RPM. A drain system must be provided to handle a water flow rate of TBD gpm and must insure proper engine operation after cleaning.

Design of the engine intervals should not preclude optional solid internal particle cleaning (carbo blast).

5. Attitude Limits - The gas turbine will operate satisfactorily under the following installed conditions: maximum of 4 inches on 100 feet longitudinal and/or 3 inches on 20 feet transverse. This would accommodate the worst soil/foundation situation.

6. Clearance Envelope - The gas turbine with its generator and auxiliaries will be located in a gas turbine building. The gas turbine itself will have acoustical treatment consisting of either acoustical insulation directly on the surface of the engine or an easily removable hood. A platform will be provided around the engine to assist with maintenance.

   In order to minimize the height of the gas turbine building, removal of an engine module should not require a vertical lift more than 1/2 the engine diameter. An overhead crane capable of at least a 25 ton lift will be provided within the gas turbine building.

7. Allowable Interface Loading - In order to minimize the impact on the engine, the coupling to the generator will be designed so as to limit the load on the engine to a maximum of 1500 lb axial force.

8. Interchangeability - All newly designed or modified engine components and modules must be interchangeable with previous components and modules they are intended to replace.

SECTION IV

DESIGN REQUIREMENTS

A. General Criteria

The gas turbine will be designed primarily to meet the requirements of the electric utility industry. It will satisfy as closely as possible the following customer-stated design life and operating cycle requirements:

1. Starting & Barring

(a) Starting

The gas turbine with its direct coupled generator will be rotated during startup by an electric motor and hydraulic torque converter. The start time will be 20 minutes-consistent with base load or intermediate operation. The gas turbine will be designed to accommodate jacking oil pressure. Jacking oil will be supplied by the lube oil system to minimize breakaway torque.

(b) Barring

The turning gear and barring motor will be located on the opposite end of the generator shaft from the gas turbine. The turning gear will be designed for TBD RPM considering both the barring function and requirements related to water wash.

- The barring system will rotate the engine to provide uniform cooling to prevent the rotor from seizing after shutdown.
- It must be capable of rotating the engine from a standstill and barr at a minimum of TBD rpm to assure the formation of a bearing oil film.

2. Safety

The gas turbine will incorporate the following features for safety.

(a) Emergency Shutdown - Parts will be designed to avoid failure, yielding, distortion and buckling during the induced thermal transients of emergency shutdown.

(b) Blade Loss and Containment - All structures will be designed to withstand blade loss loads. (Sec. V.C.2.b.) without major failure. Blade containment will be provided in the gas generator and power turbine. (Section V.D.2.d.)

(c) Shaft Failure - In the event of gas turbine shaft failure, energy will be removed from the rotor, in order to prevent a disk from reaching burst speed. This energy is to be dissipated by the following sequencing of events:

1. blade failure
2. blades meshing, if compatible with the aerodynamic and mechanical configuration
3. rubbing of shafting and end hubs against static members

The gas turbine will be desiged for short time operation at least TBD % speed without damage.

(d) Bearing Overtemperatures or Oil Loss - The gas turbine will be shut down immediately if bearing overtemperature or oil loss occurs. On loss of oil supply, the gas turbine will be tripped by the emergency shut down control system and the emergency lube oil pump will be energized to protect the bearings during coast down.

(e) Fire Protection - The gas turbine design will recognize the four basic elements of fire safety: prevention, containment, detection, and extinguishment.

- All fuel to the gas turbine will be shut off in case of fire followed by the balance of the standard emergency shutdown procedure.

- The gas turbine and accessories will be designed to minimize ignition sources such as hot surfaces, air leaks, and sparks generated by electricity or friction.

- Flammable fluid systems will be designed, shielded, or located to minimize the likelihood of leakage coming in contact with potential ignition sources.

- The gas turbine will be designed to minimize the likelihood of uncontained internal fires.

(f) Foolproofing - Where possible, hardware will be designed so that parts which are not structually or functionally interchangeable are physically non-interchangeable.

(g) Failsafe Overspeed and Overtemperature Protection System - The gas turbine will be protected from an adverse overspeed or overtemperature condition by a failsafe protection system. The system will shut down the gas turbine in the event of:

Loss of Flame
Overspeed (Due to Loss of Electrical Load)
Excessive Vibration
Turbine Overtemperature
Loss of Lube Oil
Loss of Turbine Cooling Air
Bearing Overtemperature

The protection system will be foolproofed against manual override, independent of the normal control system, and provided with automatic protection system testing.

3. Bleeds and Bleed Air

Bleeds and bleed air will be kept to a minimum to avoid the attendant complexity and sources for failures, decreased system efficiency and extra initial and installation costs. Compressor air used for cooling purposes will be extracted and reintroduced so as to minimize entry of particulate matter.

4. Anti-icing

Rugged construction of the gas turbine and favorable atmospheric site conditions usually preclude icing problems in heavy duty industrial gas turbines. Therefore there will be no anti-icing provisions.

5. Materials

All material design specifications and data must originate or be approved by Materials Engineering before utilization in the gas turbine design.

6. Manufacturing Considerations

(a) Quality Specifications

- Materials
  The use of low cost and durable materials in the form of castings, forgings, and plate will be emphasized wherever feasible. Materials will be controlled through industrial standards and specifications.

- Surface Finish Requirements
  Surface roughness allowables will be liberal on all hardware except where fatigue strength is jeopardized on highly stressed parts or where good finish is required for aerodynamic performance.

- Tolerances
  With unit weight not being a significant design criteria, noncritical tolerances on parts will be significantly liberal to insure minimum level of scrappage and parts rework.

(b) Fabrication Methods

Welded steel plate and rolled ring structures along with iron and steel castings will be used wherever applicable in the interest of economy.

(c) Standard Hardware

- Locking features associated with threaded members will use lockwires. Lockwire will be a minimum 0.032 in diameter wire in a larger hole to minimize assembly time.

- Standard industrial off-the-shelf hardware in the form of bolts, nuts, seals, flanges, etc., can be utilized throughout the engine with emphasis placed on maintaining high quality and minimizing the number of different sizes of hardware.

- Special Tooling
  Where feasible, the design will provide built-in support equipment and tooling to minimize the special tooling required to assemble and disassemble the engine. Required special tools will be designed for use in an electric utility environment. Special consideration will be given to designing simple, low cost, minimum weight tools.

  The design of the gas turbine will emphasize and allow for the use of existing heavy industrial tools such as cables, slings, turn buckles, anchor shackles, and trunnions whenever possible for assembly and disassembly. Incorporation of special maintenance tools will be considered only after the use of the aforementioned tools has thoroughly been investigated.

- Unified Thread System
  The unified thread system, UNC Class 2, will be used.

7. Standard Codes

(a) ASME Boiler and Pressure Vessel Code (Section VIII), Division II. All calculations will be done consistent with Division II, referring to Division I only for material allowables where applicable.

Relative to Division I, the rules of Division II are more restrictive in the choice of materials which may be used, but permit higher design allowables in the range of temperatures where yield or ultimate strength is limiting. A more precise analysis is required and certain design details are prohibited. This code will be used in the gas turbine case design (See Sec. V.D.).

(b) Code for Pressure Piping ANSI B31.1 will be used on all support equipment.

(c) If expansion joints are used, they will conform to the Expansion Joint Manufacturers Association (EJMA) guidelines.

(d) All electrical components will conform to National Electric Code Class 1, Division 2, Group D.

8. Instrumentation Provisions

The following instrumentation will be supplied for control:

- Supply oil pressure and temperature at one location within the engine
- Oil temperature out of each bearing compartment
- Breather pressure at the source
- Vibration indicator at each bearing location

Each sense point will have these sensors for reliability. Instrumentation sensors will be replaceable without engine disassembly. In addition provisions will be made for instrumentation for the on-line condition monitoring system which is independent of the control system. The requirements for condition monitoring instrumentation are:

Vibration

- Vibration at each bearing
- Case vibration in 6 places

Temperature (3 sensors at each location for reliability)

- Compressor inlet
- Disc cavity
- Cooling air inlet to turbine
- Compressor exit
- Exhaust rake at each burner

Pressure (Total)

- Compressor inlet
- Burner
- Exhaust rake (10 locations)

B. Design Support Requirements

1. Maintainability

(a) Maintenance Concept

The gas turbine will be designed to permit a maximum amount of on-site, installed maintenance. It will be designed to accommodate an on-condition maintenance concept with no scheduled removals for overhaul of either the complete engine or modular sections. Sufficient inspection of parts and provisions for monitoring critical engine parameters will be provided to permit utilization of this concept. Modular assembly and

disassembly of the engine will be performed with the engine mounted horizontally in its running stand.

- Scheduled Maintenance

  Scheduled maintenance activities will include of routine inspection during engine operation, Major Field Inspection (MFI) and Hot Section Inspections (HSI).

  Hot Section Inspection will be performed annually. This will consist of borescope inspection of key compressor and turbine rotor blades, fuel nozzles, combustion cans, transition sections and first stage turbine vanes.

  Major Field Inspection will be performed every two years. This inspection will require disassembly of combustion section with removal of the combustion cans to permit a thorough visual inspection of all fuel nozzles, transition sections and first stage turbine vanes and blades.

  At alternate major field inspections (every 48,000 operating hours, or approximately, eight years) all stages of the compressor will be inspected by borescope. The necessity to visually inspect any or all bearings will also be determined at this time based on trends which have been established by bearing temperatures, vibration indications, etc.

- Unscheduled Maintenance

  Unscheduled maintenance requirements will be established by unsatisfactory trends indicated by engine monitoring equipment (i.e., vibration, bearing temperatures, oil consumption, etc.). Unscheduled maintenance is categorized herein as light or heavy maintenance.

  Light maintenance is that maintenance which can be accomplished without removal of the engine or engine modules from the operating mounts. Individual parts which should be replaceable or repairable on site in the light maintenance category include:

- Combustors
- Fuel Manifolds and Fuel Nozzles
- Bearings
- Bearing Seals
- Fluid Line Leaks

Heavy maintenance is that maintenance which will require the removal of the engine or engine modules to gain access to the part requiring repair or replacement. Disassembly of the engine will be accomplished utilizing the modular concept. After removal of an engine module, the module may either be repaired on site or shipped to a major repair shop at the customer's discretion. In the heavy maintenance category, it is desirable that all modular sections of the engine plus the following individual parts should be replaceable on-site:

- Complete Rotor - (Engine Split at Compressor/Diffuser Plane)
- Inlet Case Module
- Compressor Module
- Diffuser Case Module
- Turbine Module
- Turbine Exhaust Case Module
- Front Outer Airseal
- Rear Bearing Front Oil Seal
- Rear Bearing Rear Oil Seal
- Rear Outer Air Seal
- Combustion Cans
- Transition Sections
- Turbine Nozzle Guide Vanes
- Rear Bearing
- Compressor Blades (Stages R_7, R_8, R_9)

(b) Modular Construction

- The gas turbine will be designed such that each major section is a separate independently replaceable module. Rotor and stator modules will be serialized to facilitate tracking of part time.

- Any gas turbine module will be replaceable within 78 hours elapsed time.

- Convenient break points will be provided in engine plumbing to facilitate module separation and allow translation of the gas turbine or its modules during assembly and disassembly.

- The gas turbine will be designed to permit replacement of front end modules without removing aft end modules. Upon forward translation of the gas turbine, modules from the turbine rearward will be replaceable without disassembly of front modules.

(c) Borescope Provisions

Provisions will be incorporated for borescope inspection of all compressor and turbine blades, fuel nozzles, combustion cans and all 1st turbine vanes. Where possible, inspection ports will be accessible without case disassembly.

Adequate borescope inspection of the gas turbine requires that the rotor have the capability of rotating at very low speeds with the capability to stop for a thorough inspection of each blade.

2. Reliability and Availability

Reliability will be a prime consideration in all phases of the design to assure low frequency of power outage for repair, low operating cost, and a high level of availability.

The overall engine reliability including the engine, but excluding accessory components, will be aimed at achievement of the quantitative goals listed in Section 3A.

C. Performance

1. Performance Table - The gas turbine will be designed to the performance Table TBD. Included in the P&T table are the assumptions made on engine cooling air leakage and bearing compartment bleed flow. This table will be updated periodically and changes will be reflected in Section V of this document.

2. Variation and Deterioration - Compensation for gas turbine variation and deterioration will be accounted for by including increments in temperature as follows: TBD

3. Design Life Requirements

(a) Part Life Goals

A thirty year equipment life is a value frequently used for major equipment in the electric utility industry. The term life, as used in this context, refers to that time period when the average part usefulness would have been expended through obsolescence, not to being worn out.

In general, parts will be designed for infinite life. The following are the major part life goals:

PART LIFE GOALS

	Hourly Life Goal*
Hot Parts Burner Cans, Transition Duct Turbine Blades, Vanes and Seals	50,000
Cold Parts Compressor, Turbine Cases, Disks	100,000

* All lives quoted are without repair. Lives may be extended with repair procedure.

4. Operating Cycle

Duty Cycle

The normal cycle consists of: a 20 minute acceleration rate from cold to 95% power; continuous operation at rated load until the next scheduled maintenance such as water wash; fast unloading of the engine and cooldown over a 2-3 hour period for maintenance; then, restarting and continued rated power operation.

SECTION V
STRUCTURAL REQUIREMENTS

A. GENERAL

1. Design Philosophy - The gas turbine will use conservative design procedures to insure high reliability. Design margins will be based on experience with commercial and industrial gas turbine engines using appropriate increases in margins to give the extended life and high reliability required by the program goals.

2. Design Codes - Engine cases will conform to the ASME Boiler and Pressure Vessel Code (Section VIII-Div. 2) where practicable. Longitudinal flanged (split) cases are not specifically covered by this code, but should be designed to the same level of conservatism. All plumbing will be designed to meet the requirements of ANSI B31.3. If expansion joints (bellows) are required for the plumbing, they should be designed using the Expansion Joint Manufacturers Association (EJMA) guidelines.

3. Handling, Shipping, and Seismic Loads

   (a) Shipping Loads

   The gas turbine will be designed for the following shipping loads:

   3.5 g's vertical
   1 g horizontal
   4 g's axial

   Some modules may not be shipped on the horizontal so caution must be taken to rotate the shipping loads 90°. Special protection will be incorporated to protect the thrust bearings under axial shipping loads.

(b) Shipping Limitations

Truck - In order to be shippable by truck, engine modules will be designed within the following limits:

Weight - 70,000 pounds
Height - 11.5 feet
Width - 10.0 feet
Length - 25.0 feet

Rail - If possible, the engine will be designed to the following rail shipment limitations which will allow shipment of the entire engine by rail and will eliminate the need to assemble module on site.

Weight - 150,000 pounds
Height - 11 feet
Width - 10.5 feet
Length - 50.0 feet

Maximum weight and dimensions shown are for the shipped equipment excluding railroad car or truck.

(c) Handling Features

- Features will be provided for shipping and for on-site/ off-site maintenance.
- All parts over 50 lb. will have provisions for lifting and trunnioning.
- The design handling loads are 3 g's in any orientation.
- Permanent lifting lugs suitable for slinging will be provided on all sub-packages wherever possible.

(d) Seismic Loads

The engine will be designed to withstand the following seismic conditions.

Horizontal (Ground Motion)	0.33G (2.1G max)
Vertical (Ground Motion)	0.22G (1.4G max)

Relative motion between supports of a structure or between adjacent structures will be 1" to 3".

B. ROTOR DYNAMICS

1. General - Rotor assemblies will be designed for maximum bending and shift resistance within the limits governed by rotor inertia.

2. Lateral Vibration

(a) Stiff and Flexible Bearing Criticals

The engine will be analyzed for stiff and flexible bearing critical speeds. If possible, all major criticals will be designed outside of the normal steady state engine operating spectrum. In addition, the rotor bending strain energy will be evaluated in relation to total engine strain energy.

(b) Forced Response Analysis

A forced response vibration analysis will be conducted to ensure that reasonable vibration limits are not exceeded with the maximum anticipated rotor unbalance.

3. Torsional Vibration

(a) Generator Short Circuit

Torsional vibration analysis will include the complete engine rotor and generator. The torsional modes will be governed by the electrical excitation frequencies produced at the generator air gap during short circuit.

4. Instrumentation - Linear vibration pickups will be provided for and will be placed at designated bearing locations as determined by the Engine Rotor/ Frame Vibration Analysis (Section V.B.2a).

5. Balance Provisions

(a) Rotor Assemblies

Production rotor balance provisions will be provided at the front and rear of the compressor and turbine assemblies.

(b) Trim Balance

The front of the compressor, the rear of the turbine and the compressor drive-shaft will have trim balance provisions. Access to the trim balance locations in the compressor and turbine will be provided without disassembly of the rotors. If possible the same provision will be made for the intershaft balance plane.

C. ROTATING COMPONENT DESIGN

1. General

Inertia

For single shaft, constant speed engines, overall rotor polar moment of inertia is not critical. The effect of rotor mass and polar moment of inertia on rotor vibration will be considered.

2. Rotor Integrity

General Philosophy - Sequential Failure Scenario

In order to promote rotor integrity, the design will be such to ensure that the weakest element will cause the least damage in case of failure.

a) Shaft Failure

Disk burst margins will be set high enough so that in the event of an overspeed due to shaft failure, the blades will fail before the disks.

b) Blade Loss

All main rotor structures will be designed to withstand the blade loss loads without catastrophic failure. The engine will be designed for a static load equal to 10% of any stage airfoil centrifugal pull.

c) Overspeed

All rotating elements will be capable of sustaining a TBD % overspeed condition without failure.

3. Disks

a) Stress

All disks will be designed to provide adequate yield and burst margins at TBD % overspeed.

b) Low Cycle Fatigue

All parts evaluated for LCF life will have a minimum life of 12,000 cycles.

c) Other Criteria

All disks will be evaluated for other factors including creep, web stress and fracture mechanics where applicable.

4. Compressor Blades

a) Resonances

All stages will be tuned to avoid damaging resonances. Among the frequencies to be avoided are those which relate to adjacent vane stages and, where applicable, to inlet guide vane and diffuser struts.

b) Flutter

All stages will be designed to avoid flutter throughout the operating range.

c) Blade Tilt

Blade tilt will be set to minimize gas bending stresses.

d) Steady State Stresses

Airfoils will be designed to steady stresses consistent with the table of Section 3A. Both tensile yield and creep rupture stresses will be evaluated.

e) Axial Clearance

Axial clearances should allow for blade tip and disk rim deflections due to surge.

f) Foreign Object Ingestion

To ensure against a root failure, foreign object ingestion capability will be provided for the first stage blades.

5. Compressor Blade Attachment

As discussed under the rotor integrity section, the philosophy will be to design the root attachment more conservatively than the airfoil and yet weaker than the parent disk. To do so the attachment area will be subjected to a detailed stress analysis which will include the following items:

a) Neck Width Ratio

The relative strengths of the disk lug neck to the blade attachment will be compared.

b) Allowable Steady Stresses

The following blade dovetail and disk by steady stresses will be evaluated.

1. Tension
2. Shear
3. Bending
4. Bearing
5. Combined bending and tension
6. Torsion

6. Turbine Blades

(a) Design Target

The turbine blades will be designed with a target creep-fatigue life of 50,000 hours. All calculations will be based on a maximum turbine inlet temperature of 2150°F.

(b) Creep-Fatigue Life

The calculated creep-fatigue life, targeted for 50,000 hours with stress concentration factors, will be based on the thermal cycle defined by the gas turbine duty cycle (Section IV.)

c) Hot Corrosion Life Criteria

The design hot corrosion life (sulfidation and oxidation) will be 50,000 hours.

d) Shroud Design

If airfoil shrouds are required, the shroud contact angles will be selected with consideration for damping, wear, airfoil stress, and thermal interference characteristics.

e) Tilts

Blades will be tilted to minimize combined steady state thermal and gas bending stresses for standard day operation.

f) Drop Load Overspeed

The allowable blade effective stress and maximum tensile stress is the yield strength at TBD % overspeed.

g) Flutter

First coupled bladed-disk modes and the 1st torsion mode will be analyzed for flutter to insure stability.

h) Resonance

All stages will be tuned to avoid critical resonances in the operating range.

All stages will be tuned to avoid resonances at the burner can order.

Stages adjacent to bearing support struts will avoid strut resonances.

An attempt will be made to avoid resonances with adjacent vane rows for all modes.

Platform extensions to provide flow path overlaps will not be resonant with the adjacent number of stator vanes.

7. Turbine Blade Attachment

The turbine blade attachment will be subjected to a detailed stress analysis which will include the following items:

a) Neck Width Ratio

The relative strengths of the disk lug neck to the blade attachment will be compared.

b) Allowable Steady Stresses

The following blade firtree and disk lug steady stresses will be evaluated.

1. Tension
2. Shear
3. Bending
4. Bearing

8. Shafting

(a) Blade Loss

Shafting will be designed to the 10% blade loss loads defined in Section V.C.2b. Resultant shaft maximum stress from blade loss will not exceed the tensile yield strength when combining bending and torsional stresses.

(b) Short Time Allowable Stress

Shafting will be designed to safely transmit the predicted shaft maximum torque using 0.58 of tensile yield based on outer fiber shear stress.

(c) Long Time Allowable Stress (Creep)

Shafting will be designed for 100,000 hours creep life.

(d) LCF

The required LCF life of the shaft is 24,000 cycles.

Shaft LCF lives will be calculated including axial and bending load effects in addition to torsion.

(e) High Cycle Fatigue

Fatigue life will equal or exceed 10^8 cycles.

f) The shaft will be investigated for buckling.

g) Vibration

The shaft will be investigated for flutter.

9. Spacers

(a) Shaft Criteria

Spacers are to be considered as shafting when they carry the rotor torque. Spacers will therefore satisfy the shafting criteria, Section V.C.8, as well as the following items (b) through (e).

(b) Burst Limit

The spacer will be designed for a burst margin beyond the TBD % overspeed drop load condition.

(c) Spacer Cross Sectional Area

The axial load capacity of the spacers will be in excess of the total load capacity of the tiebolts.

(d) Blade Loss

Spacers will be designed to the 10% blade loss loads defined in Section V.C.2B. Resultant spacer maximum stress from blade loss will not exceed the 0.2% yield strength when combining bending, compression and torsional stresses.

(e) LCF

All spacers will be designed to provide a minimum life of 24,000 cycles.

10. Rotor Tiebolts

Rotor tiebolts will be designed to preclude stack separation during either transient or steady state conditions.

a) Stress Allowables

Tiebolts will be analyzed for tensile and torsional stresses at assembly as well as all operating conditions.

b) Stress Concentrations

Stress concentration factors will be calculated and evaluated for all applicable areas. (i.e., bolt heads, grooves etc.)

c) Tiebolt Preload

Tiebolts will be analyzed for creep to ensure that excessive relaxation does not occur in service.

d) Buried Threads

Where possible, buried threads will be used.

d) Friction Coefficient

The friction coefficients applied to the bolts threads and washer face surface will be consistent with the standard thread procedure.

f) Central Tiebolt Vibration

Central bore type tiebolts will be analyzed for vibration. The critical frequency will be above the maximum nominal operating speeds.

D. STATIC COMPONENT DESIGN

1. General

(a) Stress Allowables

Where practical, pressure loaded cases will be designed to meet the ASME Boiler and Pressure Vessel Code (Section VIII-Division 2). The most severe actual operating condition of temperature and pressure should be recognized under fully deteriorated performance conditions. The intent of the code will always be provided.

(b) Buckling

Conservative design will in general preclude buckling problems. Parts should be checked for short time buckling and also creep buckling as detailed in item (c).

(c) LCF

The engine life goal of at least 12,000 cycles is also applicable to Static Structures. In general, this will be accomplished by limiting the maximum allowable alternating (cyclic) stress that occurs during a normal duty cycle, recognizing the combined effects of mechanical and thermally induced stresses and the appropriate notch stress concentrations.

2. Engine Cases

(a) Stress Allowables

It would be desirable to have engine cases meet Section VIII of the "Code" using the accepted Code procedures, materials and allowable stresses. However, deviations from the Code are acceptable if increased durability can be justified, and documented. Refer to Table 1 for allowable stresses.

(b) Buckling

Adequate buckling margin will be provided.

(c) LCF

The Static Structures have a minimum life goal of 12,000 normal duty cycles. Thus they will be reviewed for transient cyclic conditions and the alternating (cyclic) stress $Sa = 1/2\ (\sigma\ max - \sigma\ min)$ limited to the value given in curves in the ASME Boiler and Pressure Vessel Code. For materials not included the Boiler Code or for temperatures higher than the Boiler Code is intended will be considered on an individual basis.

(d) Blade Containment

The engine cases will be designed to contain a failed compressor or turbine blade.

3. Engine Mounts

(a) Spring Rate and Location

Mounts will be designed to provide sufficient stiffness to minimize the effect of mount spring rate on engine loading. The mount system will also be designed to be structurally determinant with known load paths to ground. This will help insure that the proper spring rate is maintained and that engine thermal distortions will be minimized for optimum blade tip clearance.

(b) Blade Loss

Mounts and mount rings will be designed to withstand 10% blade loss load.

4. Bearing Supports

(a) Spring Rate

Bearing supports spring rate will be sufficiently stiff to provide sufficient mechanical spring rate between the bearing O.D. and the heavy case structure.

(b) Blade Loss

Bearing supports will be designed to withstand 10% blade loss loads.

(c) Thermal Distortion

Steady state thermal distortion (radial growth) of the bearing O.D. will be minimized so as to hold the desired average oil film thickness.

5. Combustion Chambers and Transition Duct

(a) Creep and Creep Buckling

Combustion chambers and transition ducts will be analyzed in creep and creep buckling for 50,000 hour life goal.

(b) Life Requirements

The creep life requirement is 50,000 hours. The thermal fatigue life requirement is 12,000 typical duty cycles (lives quoted are without repair). Lives may be extended with repair procedure.

6. Compressor Stators

(a) Allowable Steady Stress

The compressor stators will be analyzed for steady stresses (i.e., tension, bending, compression, etc.) Braze material if used will be analyzed in a similar manner.

(b) Flutter

All vanes will be analyzed for flutters stability.

(c) Axial Deflection Limit

Deflections will be limited to avoid rotor interference rubs during surge and normal operation. Limits will be based on combined stator and case deflections.

(d) Construction

Whenever possible the I.D. and/or O.D. vane-shroud joint will incorporate a construction which achieves mechanical damping for vibration control.

e) Resonance

Blade passing resonances in all modes will be tuned to provide frequency margin above redline speed or frequency margin below minimum normal operation speed.

7. Turbine Vane Airfoils

(a) Design Target

The turbine vanes will be designed to meet the hot section parts life requirement of 50,000 hours.

(b) Creep-Fatigue

The calculated creep-fatigue life (50,000 hours with stress concentration factors) will be based on maximum operating temperatures with a thermal fatigue cycle as defined by the duty cycle (Section IV).

c) Hot Corrosion Life Criteria

The design hot corrosion life (sulfidation and oxidation) will be 50,000 hours for the base metal, with allowance for protective coating which need not be refurbished before 18,000 hours operation.

8. Turbine Vane and Case Attachments

Allowable Stress

The turbine case will be analyzed for all major stresses including stress rupture.

9. Plumbing - Plumbing will be designed to comply with the provisions of ANSI B31.1. Tubing will be stiffened or brackets provided as necessary to insure tube resonances margin are above rotor excitations. All expansion joints will be designed using EJMA guidelines.

10. Labyrinth and Knife Edge Seals (Static and Rotating)

(a) Vibration

Seals will be analyzed for resonant and coincident frequencies. Natural frequency margin will be maintained so that any given mode is either below or above the normal operating engine speed. Seals will be designed to be dynamically stable.

(b) Stress

A burst margin beyond the TBD % over speed condition is required for all rotating seals.

A 24,000 normal duty cycle LCF life is required.

(c) Clearances

Engine hardware will be designed to accommodate, without major engine failure, variation in leakage flow through labyrinth seals should rubbing occur.

The effect of deteriorated seals on critical parts will be established.

ANCILLARY EQUIPMENT DESIGN CRITERIA DOCUMENTS

Contract RP1187-1

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INTRODUCTION

An important requirement in achieving the goals of the HRGT program is to significantly improve the reliability of the ancillary equipment. In general the ancillary equipment is off-the-shelf so that improving the reliability depends on system design thru the use of redundancy and component reliability by proper definition of equipment requirements. These requirements include consideration for local environment and margins to accomodate occasional extreme conditions. Another approach to improved reliability is attention to the arrangement and accessibility of components to minimize the time for corrective maintenance.

In order to provide the proper design philosophy, priorities and general guidance in the design of the ancillary systems, Design Criteria Documents were prepared under Tasks 10 and 14 of the HRGT Phase 1 program.

A separate Design Criteria Document section was prepared for each of the systems. In many areas quantification of the criteria was beyond the scope of the Phase 1 program, requiring more detailed study in Phase 2 or experimental evaluation.

The specific reliability goals for each system were also left for definition as part of the overall reliability goal setting in the initial task of the Phase 2 program.

Many of the specific requirements for the ancillary system which stem from the gas turbine design have also been left for definition during the gas turbine preliminary design in Phase 2.

DESIGN CRITERIA DOCUMENT
GAS TURBINE CONTROLLER

I. INTRODUCTION

The purpose of this document is to define the initial design philosophy and guidelines for the Gas Turbine Controller which supports the Gas Turbine in a combined cycle powerplant. The design criteria gives reliability equal priority with functionality, performance and cost in keeping with the goals of the program, EPRI 1187 High Reliability Gas Turbine Combined Cycle Development.

This design criteria is subject to review and revisions as the requirements and design evolve.

Present gas turbine installations have two separate control units for fuel control and station sequencing. This arrangement leads to unnecessary complication of communication tasks and requires a great deal of cables and wires. Electrical isolation of arcs and spikes from the relay operation of the sequencer has been a problem.

The state of the art of programmable control systems makes it possible to select a single, off the shelf unit, to perform both the gas turbine fuel control and sequencing. For this reason, only one design criteria document is written, covering both fuel control and sequence operation. This control system will be referred to as the integrated gas turbine controller. (GTC)

Improved reliability can be realized with the excellent electrical noise isolation of programmable systems and the substitution of software for relays and electronic analog circuits. Further reliability improvements are achieved by the elimination of wires between units, the reduction of communication tasks, and critical circuit and sensor redundancy. Reliability backup for the controller is provided by manual control modes and delayed automatic shutdowns to enable manual takeover.

II. GENERAL DESIGN OBJECTIVES

The objective of the gas turbine controller is to provide reliable, fully automatic and safe operation of the gas turbine from startup to line synchronization, rated power, and shutdown. The powerplant design will be for baseload service, operating primarily at rated power. The system will accept local operator inputs and display corresponding operating states. All critical operating conditions and standardized performance parameters will be continuously displayed to the operator. Operating limits will be annunciated and timed out prior to automatic shutdown. There will be 2 independent gas turbine protective controls. One is part of the gas turbine controller with alarm and shutdown limits, timed shutdowns, and subject to selected overrides. The other protective system will be totally independent, including sensors and monitors, not subject to overrides, and the shutdown limits will be outside those defined by the first protective system.

The following are further control system objectives for the normal operation of the gas turbine:

- The GTC will provide stable operation and sufficient surge margin (at least TBD%) at all operating conditions.
- The GTC will prevent rotor overspeed (not to exceed TBD RPM), and assure positive emergency fuel cutoff.
- The GTC will prevent exhaust gas overtemperature (TBD % of schedule), or excessive thermal gradients in the hot section.
- The fuel control will accomodate a wide range of gaseous and liquid fuels.
- The GTC design will provide for safe shutdown from any possible operating conditions. A control system failure will not lead to major gas turbine damage.
- The sequencing and analog control of all necessary ancillaries supporting the gas turbine will be provided by the GTC.
- The GTC will have provisions to permit uninterrupted service by allowing maintenance on redundant sensors or monitors and discrete circuits while the overall controller is operational.
- Sensor circuits which can cause shutdown will include redundancy and the necessary logic to ignore failed sensors.
- Control loops will have provisions to assume direct manual control of both input and output signals.

III. SCOPE

This design criteria covers the requirements of a supervisory and process control system for the gas turbine and its supporting ancillary systems. The primary elements to be controlled are:

- Gas turbine fuel control
- Gas turbine operational sequence control
- Compressor bleed valve control
- Compressor inlet vane control
- All ancillary sequence control
- Selected ancillary process control
- Line synchronization and output breaker control
- Alarm annunciation
- Status and performance displays

The GTC does not include control of the following system:

- Balance of plant
  - Heat recovery steam generator
  - Steam turbine generator
  - Steam plant systems
- Electric generator voltage control
- Emergency DC electrical power system
- Auxiliary power system

IV. OPERATING MODES

The GTC is primarily designed to support the normal gas turbine operating modes:

- Gas turbine off
- Startup including line synchronization
- Min power
- Rated power
- Normal shutdown
- Gas turbine barring
- Emergency shutdown

Some maintenance and test modes will be supported by the GTC:

- Water wash and drain
- Special tests
- Gas turbine maintenance

No provisions are made for a precise frequency mode of operation, black start capability, or a synchronous condenser mode.

As a minimum, the following control room mode switches will be provided:

Auto: Selection of "auto" allows the GT to sequence to a preset load limit after initiating a start.

Manual: Selection of "manual" allows the GT to sequence to synchronous speed after initiating a start. Synchronizing and loading are under manual control.

Test: GT test allows the GT to sequence to idle speed while further operation is under the control of the operator on rotor speed only.

Water Wash: This function will check GT permissives for water wash and initiate the start drive. No control is provided for the ancillaries.

Gas/Liquid: This switch is provided for selection of either liquid or gaseous fuel during or prior to GT operation.

V. GAS TURBINE IMPOSED REQUIREMENTS

The control of the gas turbine imposes the following requirements on the GTC:

- Fuel control
- Compressor bleed and guide vane control
- Sequence control
- GT permissives
- GT protectives

These functions will be supported by the GTC consisting of the following integrated devices:

- Programmable controller
  - Controller hardware
  - Controller software

- Sensors

- Instruments

- Monitors

- Synchronizer

A. Fuel Control - The fuel governor is that portion of the GTC software which regulates fuel flow, either liquid or gas to the GT. The fuel governor will accept analog input signals from GT sensors and discrete inputs from the sequencer. The fuel governor will provide analog output signals for the control of bleed valves and the inlet guide vane actuator according to pre-programmed schedules incorporated in various control loops. It will provide an analog output signal for the fuel valve positioner and discrete output signals for sequencing.

The fuel governor will provide the basic function of regulation and stability control under all conditions of operation, specific functions which will be provided are:

- Minimum fuel limiting
- Maximum fuel limiting
- Speed control for startup
- Speed control for shutdown
- Load rate control
- Maximum speed limiting

The required schedules include:

- Max. and min. fuel flow
- Rotor speed control
- Power control
- EGT limit
- Rate of change of EGT limit
- Power loading and unloading
- Rotor speed coast down

The flow of fuel will be controlled at all stages of engine operation to assure efficient and safe operation. A surge margin of not less than TBD % will be maintained at all times and thermal shock will be avoided to assure long engine life. Fuel flow will be monitored and compared to expected flow, high and low flow limits will be established and annunciated.

B. Compressor Bleed Valves - The GTC will provide a discrete output signal to control the bleed valves as a function of rotor speed, ambient temperature, compressor exit temperature, and requested fuel flow.

C. Inlet Guide Vanes - The GTC will provide an analog output signal for proportional control of the inlet guide vane angular position as a function of rotor speed, ambient temperature, compressor exit temperature, and requested fuel flow.

D. Sequence Control - The GTC will contain the logic equations to supervise the GT sequence. The sequence control will condition the GT and ancillaries for a start, allow a start if all GT permissive parameters are met, initiate starter action, initiate ignition and fuel flow, allow the GT to accelerate to design speed, allow automatic or manual synchronization and loading as a function of rotor speed, critical events and real time, operation of the ancillary equipment will be sequenced as a function of rotor speed, and critical events. The GTC will monitor all important events and characteristics and will cause a start abort if rotor speed or other critical events are not reached in allowable times.

The GTC will also provide the logic required to condition the GT through a normal stop from any steady state operating level. From full power, the GTC will allow the GT to unload, control the cooldown time and level, activate the braking equipment, monitor coastdown, deactivate the ancillary equipment and otherwise place the site in a ready to start condition. A normal start up sequence involves the following:

- Start
- Ignition
- Constant fuel for ignition
- Closed loop rotor speed acceleration
- Closed loop on frequency for synchronization
- Close breakers
- Closed loop on power for loading
- Closed loop on EGT for rated power

E. CT Permissives - The GTC will contain the logic equations to prevent a GT startup for loading, waterwash or test if any of the required GT support parameters are not within predetermined limits. Also, at any time during startup, the GT permissives will prevent the sequencer from advancing or initate a shutdown, depending on conditions.

F. CT Protectives - There will be two independent protective systems for the GT. One system will be contained in the logic equations of the GTC and will schedule GT shutdown for the following conditions:

- Bearing overtemperature
- Excess vibration
- Loss of flame
- Loss of lube oil
- Loss of turbine cooling air
- High rotor speed
- GT exhaust gas temperature limit

All of these conditions will be first annunciated, then alarmed with a time limit for shutdown. This alarm can be cleared within the allocated time to prevent a shutdown.

The backup protective system will be independent including sensors and monitors and will close a solenoid activated fuel valve if a GT trip condition is sensed. The GT trip set points will be beyond those of the primary system and will not be subject to overrides. The back up system will monitor the same conditions as the primary system.

G. Sensors - All GT critical sensors, required for control or protection, will be redundant and allow continued, uninterrupted operation if a single failure occurs.

H. Instruments - The following GT parameters will be clearly visible, at all times, in the control room and be displayed in engineering units with at least 2 sec. update frequency:

• Rotor speed	RPM
• GT temperatures	°F
• GT pressures	PSI

- Fuel valve position % open
- Power MW
- Frequency CPS
- Running time HRS
- Clock
- Sequence Status State No.

All other parameters will be displayed on operator request at the system terminal (CRT).

I. Monitors - Vibration monitors will be redundant at each location with independent sensors.

A combustible gas monitor will be provided with alarm limits for gas sampling in the engine enclosure.

J. Synchronizer - The synchronizer will have an auto/manual switch for automatic or manual synchronization. This unit will provide all the switches and instruments for manual synchronization.

VI. ANCILLARY IMPOSED REQUIREMENTS

The basic requirements imposed on the GTC by the ancillary systems and the main breaker are on/off and sequence control. The GTC will also provide proportional, integral, derivative control, or any combination thereof, as required. The control requirements of each ancillary are described in the respective ancillary design criteria. The GTC will also monitor ancillary system process variables such as consumable quantities, flows, pressures and temperatures. These variables will be displayed, in the control room, in engineering units. Selected analog signals will be compared with predetermined limits and "out of limit" conditions will be annunciated, or activate the GT protective system. Some of these conditions will require GT shutdown if the annunciation is not cleared within a preset time. The following table is an outline of the ancillary system and control requirements:

System	Signal from GTC	Signal From Ancillary	Function
Main Breaker	Open/Closed	Open/Closed	Synchronization Confirmation, Display

System	Signal from GTC	Signal From Ancillary	Function
Fuel Supply System			Sequence
		Fuel Quantity (Day Tank)	Display
		Fuel Temp.	Display
		Fuel Flow	Process, Display
		Fuel Pressure	Alarm, Display
		Annunciator	Alarm, Display
Lube Oil System	On/Off Main Pump		Sequence
	On/Off DC Backup Pump		Sequence
		Tank Outlet Temp.	Display
		Cooler Inlet Temp.	Display
		Cooler Outlet Temp.	Display
		Filter Press. Drop	Display Alarm
		Bearing Inlet Temp.	Display, S/D, Alarm
		Bearing Outlet Temp.	Display, S/D, Alarm
		Bearing Inlet Press.	Display
		Breather Pressure	Display, S/D, Alarm
		Metal Chip Detector	Display Alarm
Hydraulic Servo System	On/Off		Sequence
		Tank Outlet Temp.	Alarm, S/D, Display
		Cooler Inlet Temp.	Alarm, Display
		Cooler Outlet Temp.	Alarm, Display
		Filter Press. Drop	Alarm, Display
		Pump Outlet Pressure	Alarm, Display
		Regulated Pressure	Alarm, S/D, Display
Water Wash	On/Off		Mode, Sequence
		Water Level	Alarm, Display S/D
		Water Temperature	Alarm, Display
		Inlet Solution Conductivity	Display
		Effluent Solution Conductivity	Display
		Pump Outlet Pressure	Display
Engine Heater System	On/Off		Sequence
		Air Temperature	Display, Alarm
		Air Flow	Alarm
Start System	On/Off		Sequence: Start, Shutdown, Water Wash
		Motor Current	Alarm
		Motor Temperature	Alarm
Fire Protection System	None		
		Fire Alarm Active	Alarm, Shutdown
		Agent Storage Press.	Display, Alarm

VII. ELECTRICAL REQUIREMENTS

The primary input power requirements to the GTC will be 120 VAC, 60HZ from the site uninterruptible power supply. The control system design will be such that normal operation is not affected by a $\pm$ 20% deviation from nominal voltage. The GTC will have sufficient internal power to retain memory for at least 24 hrs.

VIII. ENVIRONMENTAL CONDITIONS

The GTC will, under normal conditions, be located in an air conditioned, clean environment of the control room. However, it will be required, that the controls will function properly and without interruption in case of air conditioning failure for at least 24 hours.

Normal Environment:

- Ambient temperature 65°F to 85°F
- Relative humidity 40% to 50%

Abnormal Environment:

- Ambient temperature -30°F to 110°F
- Relative humidity Up to 90%
- Shock Loads (Seismic)

  Horizontal 0.33G (2.1G max)
  Vertical 0.22G (1.4G max)

- Dust 180 gm/m^3
- Salt entrained in air 0.003-0.01 ppm

Under normal operating conditions, the air will be filtered and free of particulate matter. Special care will be taken with the location of the air intake ducts to avoid metallic particulates and engine exhaust gas containing SO_2.

The control hardware will be removed from any fire hazard area and the control room fire protection system will in no way affect proper operation of the controls. Also, the GTC cabinets will be secure from any liquid leakage from external sources.

Lightning protection for the control equipment will be provided by proper grounding of the control building and all ancillary structures and machinery with control wires and cables leading to the control system.

IX. STRUCTURAL AND PACKAGING REQUIREMENTS

The GTC cages and its input and output components will be designed for mounting in standard racks, suitable for use inside an operators control room. The racks will have both front and rear access doors. The cages will be removable from the enclosures from the front.

No special size or weight limitations are required for the GTC. For handling and shipping, the equipment must at all times be maintained in an protected environment not exceeding the limits given in the environmental section.

Materials and processes used in the assembly of the GTC will be of high industrial quality, will not contribute to personnel hazard, and will not be in conflict with the described use.

X. RELIABILITY AND MAINTAINABILITY CRITERIA

All control equipment will be designed with the objective of attaining a service life of 30 years when subjected to normal handling, maintenance and use. The designed mean time between failure on the operating system will be not less than 40,000 hours, and the mean corrective time will not exceed 2 hrs. The GTC will operate continuously for 9,000 hrs. without shutdown for maintenance. 40 hr. maintenance intervals will be scheduled every 9,000 hrs. to coincide with the CT overhaul period.

To assure this reliability and maintainability criteria, the GTC will be designed with redundant major control circuits, capable of taking command automatically and without disturbing the GT process. The GTC protective system will allow time for operator intervention prior to GT trip by a system of alarm timing and acknowledgment. All scheduled maintenance will conform to the specification of the manufacturer. However, such maintenance will be accomplished during normal system operation or in accordance with the GT overhaul schedule. A reasonable spare parts inventory for the GTC, sensors, transmitters and monitors will be available at all times and no part will delay control system repair or maintenance for a period in excess of 48 hrs.

XI. SPECIFIC DESIGN GUIDELINES

A. The following specifications and publications are applicable:

- JIC Joint Industry Conference
- NEC National Electric Code
- EIA Electronics Industries Association
- USASI US Standards Institute
- IEEE Institute of Electrical and Electronic Engineers Standards
- NEMA National Electrical Manufacturers Association Standards
- UL Underwriters Laboratories

B. Connectors, Cables And Signal Separation - All connectors to external wiring will have a locking mechanism to maintain engagement of mating pieces. To provide for easy rack removal, service and replacement, the modular cages will be interconnected by connectors and flexible cables. All cable connectors will be keyed and marked to prevent mismatched connections.

Cables will be capable of direct burial. Cables, connectors, and cable separation will conform to FCCS-DIST pallet electrical wiring and cabling specification and FCCS-0160 inter-pallet and power conditioning subsystem electrical cable specification.

All wiring in vicinity of rotating equipment will be protected from chafing due to vibrations. Thermocouples close to the high temperature sections will be protected from short circuits by the use of ceramic insulated wiring.

C. Electrical Noise Susceptibility - Proven grounding and shielding methods will be used to prevent internal inductive and capacitive coupling as well as to provide high common mode rejection throughout the control system. Internal circuits will be designed to be immune from undesirable transients and house. The control system will be designed to distinguish between a correct input signal and a spurious transient signal such as generated by reactive loads. All input channels will be filtered and all discrete input channels will include isolation to protect internal logic circuits from high voltage and noise as may be encountered in the control room. A common mode rejection ratio of 100 DB at 60 HZ will be provided at all double ended analog inputs. Light emitting diodes (LED's) will only be used if in conformance with the factory mutual requirements for intrinsic safety. All output

channels will isolate logic level current devices from external current devices and will include transient suppression to eliminate inductive feedback from power devices. All modules will be decoupled from the input power bus to eliminate inductive electrical noise. The control system will operate satisfactorily in the electromagnetic noise environment of the control room.

D. Update Frequencey - Each analog input and each control output will be scanned or updated at its own assigned frequency. Analog input and control output for selected control loops will be updated at least five times every second. All other analog input/output points in the system will have available scan periods ranging from 0.2 seconds to 1.0 minute. At least two input/output signals will be independent of the process scanning and act without delay.

E. Power Consumption - The power dissipated by the control equipment will not exeed 2KW.

DESIGN CRITERIA DOCUMENT
FUEL SUPPLY SYSTEM

I. INTRODUCTION

The purpose of this document is to define the initial design philosophy and guidelines for the Fuel Supply System which supports the gas turbine in a combined cycle powerplant. The design criteria gives reliability equal priority with functionality, performance and cost in keeping with the goals of the program, EPRI 1187 High Reliability Gas Turbine Combined Cycle Development.

This design criteria is subject to review and revisions as the requirements and design evolve.

II. GENERAL DESIGN OBJECTIVE

The function of the fuel supply system is to deliver fuel to the engine interface at a pressure, temperature, viscosity and flow rate required by the engine fuel control. It also insures that fuel constituents which will adversely affect turbine operation are removed or inhibited before delivery to the engine.

The system will be designed for baseload applications with operation on both liquid and gaseous coal derived fuels. It will also accomodate shale oil derived fuels, petroleum based distillates, crudes, and residuals, alcohols, natural and synthetic gas. The synthetic gases, whether coal or petroleum based, may have heating values as low as 100 BTU/SCF. The plant will also be capable of operating on natural gas.

Operation on this broad range of fuels may be accomplished with at least two changeable versions of certain internal engine parts, like combustors. One configuration can be initially installed with modifications easily installed in the field for a major change in the fuel.

A primary objective, along with functionality, of the fuel supply system is high reliability. Another important objective is maintainability while the system is operational to enhance plant availability.

II. SCOPE

The fuel supply system includes all of the equipment from the liquid fuel delivery terminal or gaseous fuel pipeline at the site interface, to the combustors on the gas turbines as well as supporting subsystems such as fuel treatment and atomizing air.

A. Liquid Fuels - The liquid fuel system for the reliable engine will incorporate a Tank Loading System, a Raw Fuel Storage System, an Integral Fuel Treatment and Forwarding System, which includes a Treated Fuel Storage System and a Fuel Pressurizing and Control System.

The Tank Loading System transfers liquid fuel from a tank car, truck or pipe line and delivers it to the raw fuel storage system.

The Raw Fuel Storage System consists of one or more storage tanks and their appurtenances for other on-site bulk storage of fuel. It contains necessary nozzles and accessories for filling, discharging, draining and venting. It also includes a level gage with high and low level electrical switch contacts, a shut off valve on full, and discharge connections and a floating suction.

The Integrated Fuel Treatment and Forwarding System is incorporated to accommodate the higher levels of trace metals to be found in the ash content of coal and shale oil derived fuels, as well as petroleum crudes and residuals. The heart of the system is either an electrostatic precipitator or centrifuge which separates the fuel from a water wash solution which is used upstream to extract the undesirable metals from the fuel. This system includes heaters to bring the fuel up to optimum temperature for extraction, water pumps, an injection system for adding demulsifier to the mixture, a vanadium inhibition system (to negate the corrosive effects of vanadium compounds which are not removed by separation), flow controllers, instrument air compressors and process monitors, and on the output, a diverter valve system which directs clean fuel to a clean fuel tank or, in the case of out of limits readings, back to the raw storage tank. The clean fuel storage tank will have fuel pumps and filters associated with it for final delivery of the fuel to the engine.

The entire fuel treatment system is operated by a programmable logic control system which is programmed according to the fuel to be treated and the number of separation cycles required. This controller directs all internal functions of the pretreatment system and receives only start and stop signals from plant supervisory control system.

Immediately downstream of the fuel treatment system is the final fuel filtration stage which will also contain a final fuel shut-off valve. The fuel system includes a liquid metering valve actuated by the Gas Turbine Controller (GTC), a flow dividing system for engines with multiple burners at differing elevations, fuel manifolds and selector valves for each combustor. An air atomization system is also included to supply high pressure compressed air to the combustor fuel nozzles to assist in the mixing and vaporization of the heavier ranges of fuel.

All of the above equipment applies generally to liquid coal, shale and petroleum based fuels. For alcohol type fuels, such as methyl alcohol, there are other considerations. The Fuel Pretreatment System will not be required because of the absence of ash content in these fuels, and the low levels of N_2 found will not require the advanced technology combustor required for heavy synfuels or petroleums.

Generally the balance of the system will remain the same, although detail modification will have to be made to items such as pumps and fuel nozzles due to the lower lubricity and lower heating value of these fuels. For methyl alcohol, for example, about twice the fuel flow rate will be required as compared to petroleum fuels for a given heat release.

B. Gaseous Fuels - Gaseous fuel is piped to the powerplant site interface. The "fuel supply system" in this case includes a gas pumping and condensation removal station, desulfurizer pressure regulators, filters, shut-off and relief valves, flow control valve, manifolds and piping to the engine selector valves at each combustor.

IV. OPERATING MODES

The liquid fuel supply system for a baseload powerplant will operate, as a total system, on a continuous duty cycle. Fuel forwarding and fuel flow control devices will be in operation whenever the plant is in operation. However, depending on

certain variables such as cleanliness of fuel as procured and size of clean fuel storage tanks, sub-systems of the fuel system like the fuel treatment system can operate on a more intermittent basis. The fuel treatment system will be sized to process and store TBD % more fuel in the worst case condition than the gas turbines require. Replaceable items, such as filters, which must be serviced at a frequency exceeding the engine minimum maintenance schedule will be installed in parallel with alternate units and provided with isolation valves for in-process replacement.

The interfaces between the powerplant supervisory control and the fuel supply system will be minimized, with only such items as the main fuel pump, main fuel shut-off valve, and main fuel control valve being directly actuated. The fuel treatment system will be controlled by its own controller and actuated on demand from a level switch in the clean fuel tank.

In addition to its main operating mode of continuous fuel supply while the powerplant is in operation, there are specific sub-modes required for start-up, normal shutdown, emergency shutdown, and barring.

The fuel supply system will be operated for a short time prior to start-up to completely fill the fuel system. The engine fuel metering valve can be positioned to a normal position for filling and the individual staging valves for each burner will have a pre-start position which accepts fuel, fills the valve, and bypasses it back to the fuel tank without passing it on to the fuel nozzles. On a normal shutdown command, the fuel system will provide a controlled reduction flow down to the idle condition followed by fuel shut-off. An air purge will then be initiated through the fuel nozzles to preclude the possibility of fuel coming in the nozzles. On an emergency shutdown command the fuel system will reduce flow from system maximum to zero in approximately 0.1 second.

The gaseous fuel supply systems will also operate on a continuous duty cycle. Unlike the liquid fuel system, there is no need to prefill the system for start, or air-purge the nozzles.

The gaseous fuel shutoff will reduce flow from maximum to zero in less than 0.1 second in the event of emergency. In both liquid and gaseous fuel systems the fuel system will be used to deliver water to the fuel nozzles for turbine wash.

V. ENGINE IMPOSED CONDITIONS

Following is a description of requirements imposed on the fuel supply system by the gas turbine's operating requirements only. This is to be distinguished from requirements imposed due to other ancillary system, site requirements, or regulatory agency health and safety requirements.

A. Fuels

1. Gaseous Fuels - The fuel supply system will be capable of delivering low BTU (100-155 BTU/SCF), medium BTU gas (255-300 BTU/SCF) or natural gas to the gas turbine.

2. Liquid Fuels - The fuel supply system will be capable of delivering a wide range of liquid fuels to the powerplant. These will be synthetic liquids derived from coal and shale oil, petroleum distillates, crudes and residuals and, with system modifications, alcohols, especially methanol. Liquids whose viscosity is above 20 cs. at 250°F may not be used directly but may be used as a blending stock with lighter fuels.

B. Flow Rates - The minimum, maximum, and peak flow requirements for each Gas Turbine are:

1. Gaseous Fuels - Flow requirements for each gas turbine

	Low BTU (typical)	Mod BTU (typical)	Natural Gas (typical)
	(BTU/SCF)	(BTU/SCF)	(BTU/SCF)
Heating Value (HHV)	(BTU/lb.)	(BTU/lb.)	(BTU/lb.)
Min (pph)			
Max* (pph)			
Peak (pph)			

*At Gas Turbine rating @ 107MW

2. Liquid Fuels - Flow requirements for each gas turbine

	#2 Distillate	CDL	Methanol
Heating Value (HHV)	19,780 BTU/lb.	BTU/lb.	BTU/lb.
Min (pph)			
Max* (pph)	58,320		
Peak (pph)			

*At Gas Turbine rating @ 107MW

C. Pressure - For gaseous fuels, the pressure as delivered to engine interface will be 50 psig greater than maximum operating burner pressure.

For liquid fuels, the maximum fuel pressure delivered to the fuel metering valve will be 1500 psig.

D. Temperature - Fuel temperature of gaseous fuels as delivered to engine interface will be $-40 \leq T \leq 400°F$.

Fuel temperature of liquid fuels as delivered to engine interface will be $-10 \leq T \leq 250°F$.

E. Filtration - For gaseous fuels, particulate matter will be filtered to minimize turbine deposits and erosion. Filters will be incorporated such that particulate entering the engine will have a mean of less than 25 microns with no particle greater than 40 microns. The total particulate weight will also not exceed 0.01 grains in a standard cubic foot. In the event of a fuel's having an exceptionally large concentration of small particles which collectively exceed the total weight limit without exceeding the above size guidelines, smaller filters will be required.

For liquid fuels, filters will have a nominal particle size of 10-15 microns with no greater than 30 microns.

F. Fuel Treatment

1. Gaseous Fuels - If necessary, a desulfurizer will be incorporated in the

system which will reduce the sulfur content to a value which is less than that which produces 150 ppm sulfur dioxide by volume in the turbine exhaust at 15 percent oxygen on a dry basis. In addition, the total amount of sulfur and alkali metals, sodium, potassium and lithium, will be less than that required to form the equivalent of 5 ppm of alkalii metal sulfates in the fuel.

It should be noted that data from current gasifier development programs has shown the amounts of trace metals to be not detectable and specific steps to treat for trace metals other than particulate scrubbing are not envisioned.

2. Liquid Fuels

Trace Metals - A fuel treatment system will be installed which removes or inhibit trace metals found in the fuel. The table below suggests trace metal limits for fuel both as procured and also after treatment, as delivered to the engine. These pre-treatment limits do not necessarily represent technical limits of the treatment systems but account for certain secondary undesirable effects. For instance, levels greater than the 100 ppm of vanadium specified can be inhibited but the resultant deposition from the inhibitors becomes in itself undesirable. In addition, the costs of treating higher levels of vanadium or other metals listed below becomes excessive for an on-site operation and is better handled at the refinery level.

	Distillate Grade Oil or Synfuel		Residual Grade Oil or Synfuel		Crude Oil	
	Ideal	Max	Ideal	Max	Ideal	Max
Sodium & Potassium (ppm)	*75/.05	150/1	75/0.05	150/1	75/.05	150/1
Calcium (ppm)	2/1	8/2	5/2	20/10	5/2	20/10
Lead (ppm)	1/0.5	1/1	1/0.5	1/1	1/0.5	1/1
Vanadium (ppm)	0.1	0.5	100	400	100	400

*x/y = before treatment/after treatment

G. Atomizing Air - Successful combustion of heavier liquid fuels will require fuel/air pre-mixing prior to combustion to insure homogenous load stoichometry and avoid droplet burning. To achieve this mixing, an external atomizing air compressor system will be incorporated into the fuel supply system and used to introduce high pressure air into the fuel nozzles for pre-mixing. The atomizing air flow to the engine combustors will be 2½% (TBD) of the engine air flow (~21 pps).

The inlet of the atomizing air compressor system is supplied with gas turbine compressor discharge air at approximately 200 psia, which has been cooled in the turbine air cooler to 400°F. In the atomizing air system the air is cooled to approximately 225°F and then supplied to a single stage, motor driven centrifugal compressor which has a pressure ratio of 2.

Compressor exit air is delivered to the fuel nozzles at a pressure of 410 psia and temperature of 422°F. The atomizing air system will include its own lubricating system for the compressor and a packaged control system to prevent the system from out of limits conditions such as low oil pressure, high oil temperature, high inlet air temperature, etc. The entire system will be skid mounted and truck transportable.

H. Special Transient Requirements - The fuel system will be able to respond to a demand for rate change in fuel flow equivalent to twice maximum flow per second. In the event of an emergency shutdown signal, the fuel supply system should be able to reduce fuel flow from maximum, or any intermediate level, to zero within 0.1 seconds.

I. Fuel Supply System Pressure And Temperature - The liquid fuel supply system will operate at a maximum system pressure of 100 psig in the preliminary stages such as storage tanks, fuel treatment sections, day tanks and main pump inlet filters. The elevation of system pressure to high pressure (≅ 1000 psig) will occur only in the final stages of the system such as main pump assembly, main metering and shut-off valves and final filter.

Fuel temperature will be maintained between 0°F and 130°F in the storage sections of the system. In the fuel treatment section temperatures may be elevated to as high as 250°F, in the case of residual fuels, to adjust viscosity for proper pumping, extraction, and separation, and delivered to the engine at these elevated temperatures to promote good fuel/air mixing and vaporization. The consequences of

handling these higher temperatures on final delivery devices such as main fuel pumps, metering valves, filters will be studied to see if the cost of the more expensive high temperature hardware is excessive or if reliability is compromised. It may then be necessary to cool the fuel exiting the pretreatment system and reheat it again at the fuel nozzles.

VI. AVAILABLE SERVICES

A. Electrical Power - Electrical power for the main pumps, blowers and heaters will be available from the 60 Hertz, 460 Volt three-phase power bus system. Single phase 115 volts 60 Hz also is available to power solenoid valves and sensors. The amount of power will depend heavily on the type of fuel used, and sufficient capacity will be allowed in the design of plant to allow for full flexibility. As an example, one manufacturers 200 GPM fuel treatment system for distillate fuel uses 20 kW for pumping and 1 kW for separation while an equivalently sized residual system uses 120 kW for pumping and 38 kW for separation.

For smaller components such as pilot solenoids, control sensors, and instrumentation, 115 Volt, 60 Hz single phase power will be available.

B. Heat Rejection - Heat rejection will be accomodated by the plant auxiliary cooling water system. This water will be available at a temperature of 95°F ± TBD°F and a pressure of TBD ± TBD PSI. The capacity of the cooling water system will be designed to accomodate the flow requirement of the fuel supply system in addition to the various steam plant cooling requirements.

C. Process Water - Process water for the fuel treatment system will be available from the plant service water system at TBD psig and TBD°F.

D. Fire Protection - The fire protection system will accommodate sensor signals indicating a fire condition in the area of the fuel supply system assembly. Suitable alarms will be provided and release of a Halon fire suppression agent will be triggered in the event of fire.

The fire control system will have immediate access to the main fuel shut-off valve. Shutdown signals from smoke or flame detectors will go directly to the main fuel valve without intermediary processing in the supervisory controller.

Parallel signals originating to these fire/smoke detecting devices will also be directed to the controller to initiate complete system shutdown procedure and back-up the direct signal to the shut-off valve.

E. Drains, Effluent Disposal, Vents - Liquid fuel supply metals treatment systems have an effluent consisting of wash water metal contaminants, filterable solid contaminants and trace amounts of liquid fuels.

The fuel treatment system will include effluent water treatment systems if environmental restrictions prohibit direct disposal.

Fuel storage tanks will have sump drains for condensate and sediment removal. This effluent will be treated if local environmental regulations require.

F. Pneumatic Supply Interface Conditions - Instrument air will be available to provide 60-100 psig oil free air for pneumatically actuated valves and instruments. Dewpoint of the air should be -30°F. Temperature of the air may be between 20°F and 180°F.

G. Hydraulic Supply - A hydraulic servo system is available in the plant for use by the fuel supply system main shut-off valves and metering valves. The supply pressure is 1000 psig of hydraulic oil pressure. Oil temperature will be $0^{o}F$ to $130^{o}F$. Oil flow rate to any one component will be less than 3 gpm.

VIII. CONTROLS AND INSTRUMENTATION

A. Automatic Controls - Startup and shutdown sequences associated with the fuel supply system will be controlled by the Gas Turbine Controller (GTC).

Automatic control of certain processes within the fuel supply system such as fuel treatment will be self contained.

Operation of the fuel pressurizing and flow control system will be automatically controlled by the gas turbine controller (GTC).

1. Pressure And Temperature - The liquid fuel metering valve assembly is the most critical component in the automatic flow control sequence. Internally it will consist of a metering valve combined with a throttle valve which maintains a constant differential pressure across the metering valve. With the constant ΔP across

the metering valve, the fuel flow for a given fuel specific gravity and temperature is a function of valve position only. The metering valve position is established by the GTC based on a current signal from the generator, indicative of desired power level. The liquid fuel metering valve will operate on a hydraulic actuator. The schedule of travel vs. current signal will be adjustable for variation in fuel density.

To enhance operation of the metering valve the main fuel pump will have a reasonably constant discharge pressure over the full range of fuel flow.

Gaseous fuel flow will be controlled in much the same way except that active control loops with pressure feedback will be used to maintain constant gas pressure at the metering valve.

Gaseous fuel supply control will include provisions to maintain temperature within upper and lower limits and the supervisory controller will sense this temperature and appropriately adjust flow control valve schedule. Liquid fuel temperature will be controlled to insure reduced viscosity for the heavier fuels, promoting good fuel vaporization at the nozzles. An active liquid fuel inlet temperature control loop will be incorporated into the system.

2. Shutdown - Fuel valves will shut-off from max flow to zero in 0.1 second or less. Provision will be made for an automatic sequence to secure the fuel system after emergency shutdown. Fuel downstream of the main fuel valve will be drained back to the fuel tank. A bleed system will be provided to safely vent off gaseous fuel likewise trapped between shut-off and metering valves. Reservoirs will be provided in both liquid and gaseous systems to accomodate the pulsations resulting from these shutdowns. For liquid fuels, air purges will be initiated immediately after shutdown to prevent fuel coking in nozzles.

Malfunctions in sub-systems of the fuel system will be identified by a preliminary alarm signal before a shutdown sequence is initiated. The same signals can be used to inhibit start-up until malfunction is corrected.

3. Fire Extinguishing System - Flame/smoke detectors will operate shut-off valves directly without intermediary signal processing in a supervisory controller. Parallel signals may be provided to the controller for back-up.

B. Manual Controls

1. Start-up/Shutdown - Following each shutdown the fuel shut-off valves will be electrically latched and the start-up sequence inhibited until the latching is released by manual operation of a switch. Manual re-start or operation of the final elements in the fuel delivery systems (main fuel pump, shut-off valves, metering valve) will not be permitted. However, manual control is permitted for preliminary subsystems like tank transfer to raw storage tanks and fuel treatment. Beyond the point of the clean fuel day tank, startup sequencing will be automatic.

A single large and readily identifiable switch will be provided to allow the operator to initiate a shutdown.

2. Temperature/Pressure Adjust - Active controllers in temperature pressure control loops will have field adjustable setpoints as well as gain and rate adjustment. In critical components such as the fuel flow control valve, mechanical adjustments will also be available to limit min/max flow and to provide rangeability on the internally controlled features like constant metering valve ΔP.

3. Drain/Shut-off - Fuel drain and shut-off valves will be configured for manual operation in addition to automatic sequencing. Manual control will have a preferential direction, for example a fuel valve can be shut off at an operator's discretion but not turned on outside the proper GTC sequence.

4. Fire Extinguishing - The fire entinguishing system in the area of the fuel supply system will be available at all times for manual operation in addition to its automatic modes.

C. Remote Controls - Electrical interfaces between the GTC and the fuel supply system will be minimized. Main fuel pump, liquid or gaseous metering valves, and shut-off valves will be operated directly by the GTC. However, the fuel treatment system will have a self contained control operation for which the supervisory controller need only provide initiation or termination signals.

For liquid fuels, the supervisory controller will initiate a process whereby the fuel treatment system is put in a ready state to be further activated on demand from the clean fuel day tank. For gaseous fuels, the desulfurizer processes will be initiated by instruction from the GTC controller at TBD time before gas turbine

start-up with the actual process control effected by the desulfurizer's own system. All fuel treatment system malfunction signals not requiring immediate shutdown will be tied together and routed on one alarm signal to the GTC. No response within TBD time will then cause shutdown or switch over to a clean back up fuel such as natural gas or light distillate but not before adequate time had been allowed for trouble shooting. The treated fuel storage tank will be sized with a capacity such that rated fuel flow can be provided to the system for a time comparable to the MCT of the fuel treatment system.

The fire extinguishing system in the area of the fuel supply system will be capable of remote operation. Any operation of this sytem should be an instruction for the supervisory controller to shut the fuel supply system down.

D. Condition Monitoring And Diagnostic Information - Parameters in the fuel supply system that will be monitored by the plant on-line condition monitoring system include:

Vibration and temperature of the fuel pumps and flow divider, levels in all tanks and pressure drops in all filters.

The system will maximize the number of parameters which can be monitored by bi-level devices (switches) rather than analogue devices.

Local instrumentation such as pressure gages and tank levels should be the tank transfer system for monitoring fuel delivery. Local instrumentation should also be available in the final treatment and clean fuel delivery systems to assist in trouble shooting system and/or signal wiring problems.

IX. ENVIRONMENTAL CONDITIONS

A. Normal Environment For Equipment

1. Location - The fuel supply system will be exposed to the local outdoor ambient conditions with the exception of the final delivery elements such as main shut-off valves, metering valves, and fuel pump which will be in an enclosure.

2. Temperature - The environmental temperature for normal operation will range from -20° to 130°F.

3. Pressure - Normal operation will be provided under environmental pressures associated with altitudes from sea level to 7000 feet.

4. Humidity - Humidity levels ranging from 0 to 100% will be accomodated.

5. Externally Induced Vibration - The engine mounted components and connecting plumbing will be designed so as to accept the vibration levels resulting from engine operation.

B. System Impact On Local Environment

1. System Leakage - The system will be designed to minimize the leakage of fuels. Provision will be made to collect any liquid leakage which may occur and to dispose of the fluid in a manner to preclude the possibility of contaminating the installation area. Any stagent areas in which gaseous leakage could accumulate will have monitoring systems capable of automatically activating ventilation systems. Design practice consistent with EPA and OSHA requirements will be followed.

2. System Noise - System noise will be within acceptable engineering practices and OSHA regulations.

3. Heat Loss - In order to minimize the amount of heat required to control the fuel temperature under cold weather conditions, the fuel tank pretreatment systems and forwarding components will be insulated to the extent permitted by cost effective design considerations.

C. Abnormal Environment For Equipment

1. Temperature Extremes - The system will be designed to withstand environmental temperatures ranging from -65°F to 130°F without damage in a conditioned non-operating mode.

2. Earthquake - The system will be designed to withstand seismic loads of 0.33 (2.1G max) horizontal (ground motion) on structure no damping. Similarly it will withstand 0.22G (1.41G max) vertical (ground motion) on structure no damping. Interconnecting piping will be designed in accordance with local requirements and building foundation design criteria.

3. Fire - In the event of indication of fire by the Fire Protection System, provision will be made for immediate isolation of all fuels by activation of suitably located solenoid valves. The system design will be such as to minimize the probability of its feeding any fire in the area.

4. Fire Suppression - The design of the system and selection of materials will be such that no damage to any equipment, wiring, plumbing or connections results from the release of the fire suppression agents.

5. Conditions During Maintenance - The system will be designed to be protected from accidental damage by maintenance procedures applied to this or any adjacent system. Workmen stepping on critical components, possible damage due to dropped tools or swing crane hooks must be taken into account in the system design.

6. Lightning - Provisions must be made for proper system grounding to minimize the impact of lightning strikes.

7. Dust And Dirt - All containers will be closed except for necessary vents which will be protected by screens and filters from wind blown dust and dirt.

8. Vermin And Insects - All system air intakes, exhausts, vents and breathers will be protected with screens. All cables and tubing will be armor protected against vermin.

9. Liquid Leakages From External Sources - The system will be designed and materials will be selected such that no system damage or impairment to life will result from exposure to any fluids employed in the powerplant installation.

X. STRUCTURAL/PACKAGING REQUIREMENTS

A. Packaging Objectives And Requirements - The system package will be designed for ease of maintenance, emphasizing minimum down time for repair or replacement of parts. The packaging will be such as to protect the system from the elements and the abnormal environments cited above with the minimum use of footprint area.

B. Factors Of Safety/Codes - Normal engineering practice will be employed in sizing components, plumbing, and wiring. Where design analysis shows component selections to be marginal for the service, the next larger component suitable for

the service will be employed. Where analysis shows that life or reliability can be enhanced by the use of devices having greater performance or capacity margins than normal practice would dictate, the greater margins will be employed. The design of all plumbing will be consistent with ANSI B31.1.

C. Handling/Shipping Loads - The system will be designed to withstand handling and shipping loads of up to 3 G's without damage, except for engine mounted components which must also tolerate up to 5 G's without damage.

D. Interface Requirements And Loads - All interfaces with the powerplant and the facility will be designed to hold stresses to components, plumbing and cabling well within safe limits as a result of thermal expansion, and normal and abnormal environmental conditions.

E. Provisions For Lifting/Mounting - An overhead crane and forklift trucks will be available for lifting and moving the system and its subassemblies. All parts weighing 50 pounds or more will have provisions for lifting and trunnioning. Eye bolts will be provided for removing covers and permanent lifting lugs suitable for slings will be provided on submodules where possible. The bulk of the system will be skid mounted.

F. Servicing Considerations - The system will be designed to be serviced by simple procedures and components will be easily accessible to minimize down time associated with repair or replacement.

G. Cleaning And Flushing Criteria - The cleanliness of fluids delivered by the system will be assured by the use of suitable filters. In the event severe contamination of fluids in the various holding tanks occurs for any reason, as indicated by clogging of filters in short periods of operations, it will be necessary to flush clean the system fluid loops involved. Appropriate valving will be provided to permit such flushing on an individual fluid loop basis.

H. Size Limits - The system will be designed to be truck transportable in several major sub-system skids. Preferred size limits are 8'W x 9'H x 20'L however dimensions up to 10'W x 11½'H x 30"L are negotiable.

I. Weight Limits - Individual module weights will be less than 43,000 lbs. consistent with limits permitted by truck transportation.

XI. RELIABILITY AND MAINTAINABILITY CRITERIA AND GOALS

A. Reliability Goals - All components will be selected for maximum reliability. Redundancy will be employed in the system design where it can be shown to improve system reliability. An MTBF of TBD will be used as a goal in the design of this system.

B. Maintenance Philosophy - To the maximum extent consistent with personnel safety, the fuel system will be designed to be serviced by simple procedures while the powerplant is operating. This may require back up or alternate devices such as fuel pumps and filters. Requirements for special tools and fixtures are to be avoided. The system can have routine "walk-around" visual inspection while the system is in operation. Maintenance which requires shutting down the system will be performed during one fifty hour period every year and one 100 hour period every two years.

C. Component Access And Replacement Times - Use of modular construction will be emphasized to minimize downtime for service and replacement of components. The design of the module interfaces will be such as to facilitate access to connections and allow replacement of the module by use of simple procedures. Downtime for repair will not exceed one working shift as a goal. All cabling, external accessories, and plumbing should be capable of being checked for security of mounting and evidence of leakage or chafing. Filters will be easily accessible for replacement without loss of operating time.

D. Scheduled Down Time Criteria - There will be no scheduled periods of downtime for routine service other than that specified under Item B, Maintenance Philosophy.

E. Spare Parts Philosophy And Guidelines - In the selction of components, preference will be given to equipment which is readily available. If long lead time parts must be employed, and system availability could be jeopardized, consideration will be given to the use of installed spares. Otherwise a policy of stocking spare long lead time components will be followed.

F. Design Life - The life of all structural and plumbing components should be 30 years minimum. The goal for all other components will be consistent with the planned maintenance schedule described in Item B, Maintenance Philosophy.

G. Restrictions On Cascading Failures - The fuel system will be designed to guard against the possibility of cascading failures initiated by the malfunction of any individual system component.

XII. SPECIFIC DESIGN GUIDELINES

A. Codes To Be Observed

1. NFPA 30 - Flammable And Combustable Liquids Code - Note that Methanol is a Class 1B, flammable, fuel and may require special handling and ventilation compared to petroleum based fuels which are generally class II combustable fuels.

2. NFPA 70 - The National Electrical Code - Electrical components will be suitable for handling group D fuels in hazardous locations classified as class I, Div II, except that Medium BTU gas by virtue of its hydrogen content in excess of 30% by volume is a Group B fuel. Not all components suitable for Group D are suitable for B and the additional expense required to procure Group B rated components is not warranted unless it is certain Medium BTU gas will be a fuel at a specific location.

3. ANSI B 31.1 For System Plumbing

4. OSHA regulations will apply to the entire system design.

B. Materials To Be Banned - The materials will be selected to be compatible with all fluids with which they come in contact. All materials and processes used will be of high industrial quality and will not contribute to personnel or environmental hazard. Restrictions on low melting or non-ductile materials, such as aluminum, copper, brass or cast iron, in fuel handling systems, as specified in NFPA 30 Sect. 3140 or ANSI B31.1 Sect. 323.32 will be observed.

C. Hardware Types Not Suitable - Electrical equipment not consistent with Class I, Divn. 2 category of the NEC is prohibited.

D. Safety Provision - The system will be consistent with NFPA No. 30 and OSHA regulations.

E. Other Guidelines - Maximum use will be made of standard off the shelf fasteners, flanges, connections and seals. Self locking features will be employed on connectors and fasteners to the maximum extent possible.

DESIGN CRITERIA DOCUMENT
LUBE OIL SYSTEM

I. INTRODUCTION

The purpose of this document is to define the initial design philosophy and guidelines for the Lube Oil system which supports the gas turbine in a combined cycle powerplant. The design criteria gives reliability equal priority with functionality, performance and cost in keeping with the goals of the program, EPRI 1187 High Reliability Gas Turbine Combined Cycle Development.

This design criteria is subject to review and revisions as the requirements and design evolve.

II. GENERAL DESIGN OBJECTIVES

The function of the lube oil system is to provide a source of oil at the proper pressure, temperature, and cleanliness for use as lubricant and coolant.

The lube oil system will be designed to service the lubrication and cooling needs of the gas turbine and generator bearings from a common controlled supply of appropriate oil. In order to avoid bearing damage, uninterrupted flow of lubricant to the bearings is essential under all conditions during which engine rotor-rotation is possible. For this reason, the reliability of this system is critical to the reliability and availability of the power generation system. The powerplant availability requirements make special demands on the lube oil system design. Maintenance and component replacement will be accomplished during system operation. The design of this system will not preclude potential future gas turbine performance up-rating.

III. SCOPE

The system consists of an oil storage tank which is capable of being electrically heated in cold weather, tank vent blowers oil level sensors and indicators, duplex

filters with transfer valves, centrifugal oil pumps (one main ac driven, one standing ac driven and one emergency dc driven), two water-cooled heat exchangers with transfer valves, temperature and pressure sensors and switches, gages, and valving for control, check, isolation and drain functions. In addition to the three supply pumps, two jacking pumps (one operating and one standby), may be needed to lift the rotor journals during start and barring operations. This lifting action will minimize starting torque requirements for the starter and barring motors. The system will be plumbed and wired to interface with all engine and generator bearing housings, the powerplant automatic controller, alarm annunciators and the condition monitoring system and site a-c and d-c power sources. Operation of the lube oil system will be fully automatic in terms of control of pressure and temperature and in terms of sequencing the operation of system component during startup, normal and emergency running, shutdown, barring and waterwash modes of operation.

IV. OPERATING MODES

The lube oil system will be required to operate continuously under all powerplant operating modes in which the gas turbine and generator rotors are rotated. During startup, if special jacking oil provisions are needed, the jacking oil pump will be energized to provide a slight rotor lift, and establish a lubricating film at each of the journal bearings. This oil flow is maintained until rotor speed reaches 300 RPM at which time the jacking pump is stopped, leaving the main oil pump to provide the normal oil flow through the remainder of acceleration, operation, deceleration, and barring periods. Oil flow must be maintained during any emergency condition, including loss of a-c power. For this reason, a battery-powered d-c bus is employed to operate the emergency components which will be needed to keep the lube oil system operational in terms of flow, pressure, temperature, and heat removal.

V. ENGINE AND GENERATOR IMPOSED REQUIREMENTS

A. Oil - The system will employ a turbine grade mineral oil selected from an approved list for service with the generator and the gas turbine. Provisions will be made in the system to permit drawing off oil samples for periodic analysis to determine acid number and viscosity.

B. Flow-Rates - The system flow rate will be dictated by the lubrication requirements of the bearing design for GT and generator. The system will be capable of delivering full oil flow to all bearing housings prior to start, and following

shutdown as well as during operation, including transient maximum usage. Allowances will be made for proper heat rejection in the cooler. A 20% margin of surplus flow will be provided in the design.

C. Pressure - The oil pressure at the bearing housings will be regulated at TBD plus/minus TBD PSIG. Transient cold start pressures will be limited to a maximum of 100 PSIG. The pressure regulating means will be selected to prevent excessive pressure rise that might result from having the main and standby pumps in operation simultaneously.

D. Filtration - A full flow duplex filter system will be provided on the oil line to the bearings. The filter will be rated for nominal 5 micron service and the maximum particle size will be limited to 20 microns. Manually operated transfer valving will be provided to allow filter cartridge replacement without interruption of service. A bypass valve will be provided for use during cold start conditions.

E. Shutdown Requirements - When the powerplant is shut down, it will be barred for a period of approximately 20 hours. During this period, and until the rotors come to rest, the lube oil system will be fully operational. Similarly, the engine and generator will be subjected to barring at selected intervals of any shutdown period. The lube oil system will deliver jacking oil to all bearings during all periods of barring. In cold weather, the tank heater and its control will be kept operational to maintain the minimum oil temperature required to permit a powerplant restart without excessive loss of time.

F. Temperature Regulation - The lube oil system will supply oil to the bearing housings at a minimum temperature of 20°F above solidification temperature. The maximum oil temperature leaving the bearing housings will be 180°F (TBD) and the maximum oil film temperature will be approximately 250°F (TBD). The temperature of the oil will be controlled by heat rejection via a water cooled heat exchanger with a bypass control valve on the oil side. For cold weather operation, electric heaters controlled by suitably located thermostats will be provided in the oil tank and on exposed plumbing lines to maintain the temperature above the minimum allowable value. These heaters will be sized so that a thermostat failure does not result in oil coking.

G. Storage Tank - The storage tank will be designed as a baffled, sealed enclosure capable of holding a slight vacuum pressure. A top sealed access port will be

provided for inspection and cleaning functions and the bottom will be sloped to a low point drain valve. The working oil capacity will be such as to hold in reserve enough oil for 5 minutes at normal flow rate or enough to accommodate engine coast-down time, whichever is larger. The tank size will be large enough to accommodate an additional amount of oil equal to the capacity of all system components, plumbing and bearing housings which can drain back into the tank as a result of any maintenance action. The tank design and oil inlet and outlet locations will be consistent with Section 2 of API Standard 614.

Scavenging - Special scavenging pumps will not be used to return the oil from the bearing housings to the reservoir. Gravity drain will be employed for this purpose. The storage tank will be vented through an air-oil separator by means of an a-c driven main vent blower and a d-c driven emergency blower. A suitable vacuum pressure control system will be provided to hold this pressure at TBD ± TBD in. H_2O. Each bearing housing will have a breather which will be connected to the reservoir by separate plumbing to maintain a slight vacuum pressure in these bearing housings. Since this configuration can result in air and hydrogen from the generator becoming intrained in the oil, the oil returned to the reservoir will be passed through swirl type static degassers located inside the reservoir tank.

I. Abnormal Operating Events - In the event of any abnormal operating condition such as loss of a-c power, aborted startup or emergency shutdown, the lube oil system will remain operational, capable of delivering oil to the bearings as instructed by the gas turbine controller.

J. Journal Jacking Requirements - To minimize engine starting torque requirements, the lube oil system must be capable of lifting the gas turbine and generator rotors slightly, easing the formation of a lubricating film on all journal bearings. The oil pressure needed for this jacking purpose is a function of the bearing designs. If necessary, separate jacking pumps may be employed in the lube oil system to meet this startup requirement.

VI. AVAILABLE SERVICES

A. Electrical Power - Electrical power for the main pumps, blowers and heaters will be available from the 60 Hertz, 460 Volt three-phase power bus system. Single phase 115 volts 60 Hz also is available to power solenoid valves and sensors. The emergency pump and vent blower will be tied to the battery powered d-c bus at 125 volts (TBD) plus/minus TBD volts.

B. Heat Rejection - Oil heat rejection will be accommodated by the plant auxiliary cooling water system. This water will be available at a temperature of 95°F ± TBD°F and a pressure of TBD ± TBD PSI. The capacity of the cooling water system will be designed to accomodate the flow requirement of the lube oil system in addition to the various steam plant cooling requirements.

C. Fire Protection - The fire protection system will accommodate sensor signals indicating a fire condition in the area of the lube oil system assembly. Suitable alarms will be provided and release of a Halon fire suppression agent will be triggered in the event of fire. Design analysis will be required to determine whether the lube oil flow should be interrupted immediately by a sensed fire to preclude feeding the fire with oil or whether a time delay should be used to maintain bearing lubrication and cooling during the engine coastdown period of the triggered shutdown.

D. Drains - No provision will be made for site oil drains. Portable drain tanks with associated valves and pumps will be used to drain oil from the system or components for ultimate oil disposal off-site in accordance with local regulations. Similarly, in the event of a cleaning operation, any flushing effluent will be drained into a portable system for off-site disposal.

E. Hydraulic Power - A hydraulic servo system is provided for certain control operations in the powerplant. If needed, this supply of hydraulic fluid, which will be TBD, may be employed in the lube oil system for control functions. The hydraulic supply pressure will be 1000 PSIG TBD plus/minus TBD PSI and a fluid flow rate not to exceed 3 (TBD) gpm will be available.

VII. CONTROLS AND INSTRUMENTATION

A. Automatic Controls - Startup and shutdown sequences associated with the lube oil system will be controlled by the gas turbine controller (GTC) which will also accept pressure, temperature, and liquid level signals and respond with suitable alarms. The automatic control of temperature and pressure within the lube oil system will be self-contained and not require the action of the GTC except to initiate startup and shutdown. The lube oil system will be interfaced with the fire protection system to produce alarm indications of fire and to respond to the triggered fire protection system in accordance with the strategy devised by system design analysis. By self-contained controls, the system will automatically start the emergency pump and backups vent blower in the event of low oil pressure or breather pressure indications.

B. Manual Controls - The lube oil system will be capable of being locally started and shutdown by manual means at the oil system assembly. Oil addition to maintain proper reservoir level will be done by manual means during operation if necessary. Control of temperature and pressure will have manual overrides available and set points will be manually adjustable. Since both local and remote manual triggering of the Fire Protection System are provided as part of that system, no special provision for this is needed in the lube oil system.

C. Remote Controls - Remote (plant control room) means for startup and shutdown of the lube oil system will be provided either in a manual or automatic mode. Adjustment of control settings for temperature and pressure will not be available at the remote location. However manual override of alarms and shutdown signals will be provided at the plant control room.

D. Instrumentation For Condition Monitoring - Local gages will be provided to indicate reservoir tank pressure, pump outlet pressures, filter pressure drop, oil flow rate and oil level in the tank. Sensors with electrical output signals will be provided for tank outlet temperature, cooler oil and water inlet and outlet temperatures, filter pressure drop, individual bearing outlet temperatures, bearing inlet pressure, breather pressure and metal chip build-up, engine bearing compartment or oil breather pressure, vibration from each pump and blower, and oil acidity (if applicable). The signals from these sensors will be sent to the gas turbine controller for use in alarm and shutdown functions and/or to the on-line condition monitoring and diagnostic system. In the case of signals which can cause powerplant shutdown, such as bearing temperatures and pressures, multiple sensors will be employed using signal voting requiring agreement of at least two signals, or the use of 3 sensors with median select provision to initiate shutdown. The alarm and shutdown trip levels are all TBD subject to systems design analysis and will have provision for manual override by the plant operator.

VIII. ENVIRONMENTAL CONDITIONS

A. Normal Environment

Location - The lube oil system will be installed in an unheated enclosure adjacent to the GT generator outside the gas turbine bay.

Temperature - The environmental temperature for normal operation will range from TBD to TBD°F.

Pressure - Normal operation will be provided under environmental pressures associated with altitudes from sea level to 7,000 feet.

Humidity - Humidity levels ranging from TBD to TBD will be accommodated.

Externally Induced Vibration - The engine mounted components and connecting plumbing will be designed so as to accept the vibration levels resulting from engine operation.

Externally Induced Electrical Noise - The system design will be such as to be unaffected by the presence of RFI and EMI radiated and conducted noise such as might occur because of d-c motor commutation, operation of high energy ignition systems, and the operation of power transformers and switchgear.

B. System Impact On Local Environment

System Leakage - Since severe oil leakage can result in powerplant shutdown and adversely affect availability, the lube oil system will be designed with the goal of being leak-free. The number of plumbing joints will be kept to a minimum and maximum use will be made of welded connections. In the case of pipe threaded connections, adequate space will be provided to assure proper tightening during assembly fittings will have positive locking provisions. All piping will be securely fastened to minimize the impact of engine vibration and stresses arising from shipping loads. The piping design will incorporate provisions for thermal expansion to minimize resulting stresses on plumbing fittings and component housings and include provisions to avoid fatique due to vibration at the equipment interfaces.

System Noise - Pump and blower speeds and generated noise will be within acceptable engineering practice and OSHA regulations.

Heat Loss - Heat rejection from the system will be made through the cooling water system which ultimately rejects the heat outside the enclosure. However, in the case of cold weather operation when electric heat will be needed in the reservoir and on some oil lines, design calculations will be made to determine if tank and plumbing insulation are effective in reducing the amount of electric heat required to keep the oil above the minimum temperature. Such insulation will be employed if analysis shows it to be effective.

System Induced Vibration - The system pumps and blowers will produce vibrations which can be transmitted to structures and to associated plumbing. A design analysis will be performed to assess the impact of the generated vibrations and to establish the need for isolation to preclude the possibility of stresses to components plumbing in this and any other installed system.

Ventilation Air Flow Requirements - The lube oil system is totally enclosed and sealed. Its internal vent flow will be expelled to the outside air via suitable piping through the enclosure. Since the lube oil is not toxic nor highly volatile, it is not expected that special ventilation around the lube oil console will be necessary over and above that provided in the enclosure.

C. Abnormal Environment For Equipment

Temperature Extremes - The system will be designed to withstand environmental temperatures ranging from -65°F to 130°F without damage in a non-operating mode.

Earthquakes - The system will be designed to withstand seismic loads of 0.33G (2.1G max) horizontal (ground motion) on structure no damping. Similarly, it will withstand 0.22G (1.41G max) vertical (ground motion) on structure no damping. Interconnecting piping will be designed in accordance with local requirements and building foundation design criteria.

Fire - In the event of indications of fire by the Fire Protection System, provision will be made for isolation of the lube oil supply by activation of suitably located solenoid valves. The desire is to minimize the probability of the oil feeding any fire in the area; however, lube oil flow will be required by the gas turbine and generator bearings during coast down following a shutdown trip. The need for a time delay and its magnitude will have to await analysis of the risks involved to evolve the optimum strategy.

Fire Suppression - The design of the system and selection of materials employed will be such that no damage to any equipment, wiring, plumbing or connections will result from exposure to the fire suppression agent.

Conditions During Maintenance - The system will be designed to be protected from accidental damage by maintenance procedures applied to this or any adjacent system. Workmen stepping on critical components, possible damage due to dropped tools, or swinging crane hooks must be taken into account in the design of the system.

Lightning - Provisions must be made for proper system grounding to minimize the impact of lightning strikes.

Dust And Dirt - All containers will be closed except for necessary vents which will be protected by screens and filters from dust and dirt.

Vermin And Insects - All system air intakes, exhausts, vents and breathers communicating with the system exterior will be protected with screens. All cables and tubing will be armor protected against vermin.

Liquid Leakage From External Sources - The system will be designed and materials selected such that no system damage or impairment to life will result from exposure to any fluids employed in the powerplant installation.

IX. STRUCTURAL/PACKAGING REQUIREMENTS

A. Packaging Objectives - The system will be designed as a packaged assembly with a suitable structural base plate. It will be capable of being factory assembled, shipped as a unit and installed with a minimum number of connectors. The pumps will be mounted external to the reservoir tank and at a level such that the pump inlet is gravity fed from the reservoir. If components are mounted on the tank top, it will be rigid enough to preclude sagging and propagation of vibrations. Pads will be used for mountings to prevent any hole penetrations into the tank. The structural base plate will have a drip rim with suitable drain connections to allow collection of oil leakage or spill from a fill operation.

B. Factors Of Safety/Codes - Normal engineering practice will be employed in sizing components, piping and wiring. Where design analysis shows component selections to be marginal for the service, the next larger component suitable for the service will be employed. Where analysis shows that life or reliability can be enhanced by the use of devices having greater performance or capacity margins than normal practice would dictate, the greater margins will be employed. The design of all piping will be consistent with ANSI B31.1.

C. Handling/Shipping Loads - The system will be designed to withstand handling loads of up to 3 G's without damage. The system also will be capable of withstanding the maximum loads imposed as a result of shipment by truck or by rail transportation.

D. Interface Requirements And Loads - All interfaces with the powerplant and the facility will be designed to hold stresses to components, piping and cabling well within safe limits as a result of thermal expansion, and normal and abnormal environmental conditions.

E. Provisions For Lifting/Mounting - A portable crane and fork lift trucks will be available for lifting and moving the system and its components. All parts weighing 50 pounds or more will have provisions for lifting and trunnioning. Eye bolts will be provided for removing covers and permanent lifting lugs suitable for slings will be provided on sub-modules where possible.

F. Servicing Considerations - The system will be designed to be serviced by simple procedures and clearance around components will be such as to make them easily accessible to allow repair or replacement while the system is operating. All electrical interfaces will be wired to a central easily accessible terminal junction box.

G. Cleaning And Flushing Criteria - The cleanliness of the oil delivered by the system will be assured by the use of suitable filters in a clean system. The components and plumbing must be flushed repeatedly and new filters installed prior to the initial start-up. In the event subsequent severe contamination of the oil occurs for any reason as indicated by clogging of filters in short periods of operation, it will be necessary to flush clean the system fluid loop. Appropriate valving will be provided to permit such flushing and collection of the effluent from this operation.

H. Size Limits - The system will be designed to be truck transportable. Preferred size limits are 8'W x 9'H x 20'L. However, dimensions up to 10'W, 11½'H and 30'L are negotiable.

I. Weight Limits - The system weight will be less than 43,000 lbs consistent with limits permitted by truck transportation.

X. RELIABILITY AND MAINTAINABILITY CRITERIA AND GOALS

A. Reliability Goals - All components will be selected for maximum reliability. Redundancy will be employed in the system design where it can be shown to improve system reliability. An MTBF of TBD will be used as a goal in the design of this system.

B. Availability Goals - Since the lube oil system is critical to the operation of the powerplant, special attention is required to its design in order to avoid down time for any reason. The use of redundant components and installed spares and provisions for servicing while operating will be necessary to maximize the lube oil system availability which will be TBD as a goal.

C. Maintenance Philosophy - To the maximum extent consistent with personnel safety, the lube oil system will be designed to be serviced by simple procedures while the powerplant is operating. Requirements for special tools and fixtures are to be avoided. The lube oil system can have routine "walk-around" visual inspection while the system is operating; maintenance which requires shutting down the system will be performed only on an "as-required" basis.

D. Component Access And Replacement Times - Use of modular construction will be emphasized to minimize downtime for service and replacement of components. The design of the module interfaces will be such as to facilitate access to connections and allow replacement of the module by use of simple procedures. Downtime for repair should not exceed one working shift as a goal. All cabling, external accessories, and piping should be capable of being checked for security of mounting and evidence of leakage or chafing. Filters will be easily accessible for replacement without loss of operating time, including the use of permanent staging for viewing and cleaning the inside of filter vessels.

E. Scheduled Down Time Criteria - There will be no scheduled periods of down time for routine service other than that specified for the engine. Maintenance windows of 50 hours every 9000 hours and 100 hours every 18000 hours will be available for component replacement if necessary.

F. Spare Parts Philosophy And Guidelines - In the selection of components, preference will be given to equipment which is readily available. If long lead time parts must be employed and system availability could be jeopardized, consideration will be given to the use of installed spares. Otherwise, a policy of stocking spare long lead time components will be followed.

G. Design Life - In general the system should be designed for a 30 year life; however, as a minimum, the lube oil system will be capable of 9000 hours of operation without major maintenance.

H. Restrictions On Cascading Failures - The lube oil system will be designed to guard against the possibility of cascading failures initiated by the malfunction of any system component.

XI. SPECIFIC DESIGN GUIDELINES

A. Codes To Be Observed - The system plumbing will be designed to be consistent with ANSI B31.1. The vessels will conform to the ASME code for unfired pressure vessesl, Section VIII and all electrical design will be consistent with NFPA-70. All electrical equipment will be explosion proof rated for Class I, Group D, Division 1 conditions per NFPA-70. The system design will be in general agreement with API STANDARD 614. OSHA regulations will apply to the entire system design.

B. Materials To Be Banned - Cast iron or copper will not be used in any oil wetted portions of the system. The materials will be selected to be compatible with all fluids with which they come in contact. All materials and processes used will be of high industrial quality and will not contribute to personnel or environmental hazard.

C. Hardware Types Not Suitable - Electrical equipment not consistent with Class I, Division 2 Groups B and D category of the NEC is prohibited. Multi-set point thermal switches will not be used in any control function. All piping will be of seamless stainless steel.

D. Safety Provision - The system will be consistent with NFPA No. 30, NFPA No. 70 and OSHA regulations.

E. Chip Detectors - Conventional chip detectors would not be effective in a system using journal bearings made of non-magnetic materials; however, provisions will be made in the design for an in-line strainer type metal electrical resistance detector. The decision to employ such a device will await its experimental evaluation for applicability.

F. Other Guidelines - Maximum use will be made of standard off-the-shelf fasteners, flanges, connections and seals. Self-locking features will be employed on connectors and fasteners to the maximum extent possible.

DESIGN CRITERIA DOCUMENT
WATER WASH SYSTEM

I. INTRODUCTION

The purpose of this document is to define the initial design philosophy and guidelines for the Water Wash system which supports the gas turbine in a combined cycle powerplant. The design criteria gives reliability equal priority with functionality, performance and cost in keeping with the goals of the program, EPRI 1187 High Reliability Gas Turbine Combined Cycle Development.

This design criteria is subject to review and revisions as the requirements and design evolve.

II. GENERAL DESIGN OBJECTIVES

The function of the water wash system is to provide for the storage and mixing of the necessary fluids and the delivery of these fluids at suitable pressure to the engine section being cleaned. The system also provides for proper collection and disposal of the fluid discharged from the engine during the wash process.

The water wash system will be designed to provide a means for cleaning the engine of deposits which can adversely affect performance and lead to the corrosion of critical engine parts. Two procedures for engine washing are possible. One involves flushing a cool engine with water or a detergent solution while the rotor is motored by the starter. The other involves the injection of a cleaning agent while the engine is running. Since the latter approach has not been evaluated for use in a large stationary gas turbine engine especially in terms of cleaning turbines in service with heavy fuels, it is not recommended for use at this time. The design criteria for this system therefore, is based on the use of the more conventional concept of engine motoring water/detergent washes preceeded by rapid cool down to minimize impact on powerplant availability.

The contamination buildup in the engine will be removed by washing with clean water while the rotating parts are turned over with the starter. A detergent may be added to the water on an optional basis. Provisions will be made to accommodate the several options that will be dictated by the unique requirements imposed by the characteristics of the fuel employed and the air contaminants experienced in the installation site. The operation of this system will be under the control of the powerplant sequential control system.

III. SCOPE

The major components making up the water wash system are holding tanks for water, detergent solution, alcohol and effluent fluids, a heater for the water tank, level sensors for each holding tank, shutoff valves, tank drain valves, check valves, a pump, mixing valves, filters, water inlet and outlet electric conductivity probes, compressor inlet manifold and spray nozzle assembly, an exhaust collector and an exhuast holding tank. This assembly of components interfaces with a potable water source and with the compressor inlet wash nozzles and the fuel nozzles on the powerplant. Similarly, there are electrical connections to the powerplant controller and connections to the site drain system.

IV. OPERATING MODES

This system will operate on an intermittent duty cycle called into use every TBD hours of powerplant operation. Actual time intervals between use will depend on the air cleanliness and the type of fuel employed in the powerplant. The water wash will be called into use after an engine cooling period of at least TBD minutes. Adequate cooling will be safeguarded by time and interlock provisions in the powerplant controller which prevent energizing the water wash system prior to the appropriate cooling period. A typical water wash cycle will require from 3 to 5 one minute periods of cleaning fluid ingestion while the rotating members are turned over by the starter. The one minute fluid ingestion periods are separated by 3 to 5 minute time to allow the fluid to penetrate the engine. If a detergent is employed, it will be necessary to follow these wash cycles with several clear water rinse periods of one minute duration. Following the rinse a waiting period of TBD minutes will be required to allow the water to drain from the engine before an engine start is attempted. The engine will then be run at idle speed for a period of TBD minutes to assure that the engine has properly dried.

V. ENGINE IMPOSED REQUIREMENTS

A. Detergents - If the engine service is such as to require the use of a detergent, a selection will be made from the approved list of suitable cleaning agents. This cleaning agent will be mixed with potable water to form a TBD percent solution, which will be stored in the detergent holding tank.

B. Flow Rates - The water wash system will be capable of delivering up to 120 GPM of water at 100 psi pressure at the engine interface. Suitable hand throttle valves will be provided for adjusting the flow rate over the range of 50% to 100% of the maximum flow rate.

C. Pressure - The system will be capable of delivering maximum flow at 100 psi at the engine interface.

D. Wash Cycle Requirements - The system will be capable of providing from three to five washing periods following by from two to five rinsing periods within the wash cycle.

E. Quantity Of Wash Constitutents - The water holding tank will be sized so as to hold all the water required by one maximum wash cycle. The detergent solution and alcohol holding tanks will be sized for 50% of the capacity of the water holding tank.

F. Temperature - A water heating element will be provided to raise the water temperature up to 190°F. The temperature will be adjustable over the range of 100 to 190°F plus or minus 15°.

G. Water/Detergent Mix Ratio - The system will be capable of mixing detergent with the water over the range of 100% water to 100% detergent solution.

H. Water Heat Time - The system will be capable of heating a full holding tank of water from 60°F to 190°F in a maximum time of TBD hours.

I. Abnormal Operating Events - The system will be capable of operating in freezing weather by mixing alcohol with the wash water. Provisions will be made to adjust the mixture from 0 to 50% alcohol.

J. Engine Connection Interfaces - The system will be connected to the engine with hard plumbing which is easily removable for purposes of maintenance.

VI. AVAILABLE SERVICES

A. Used Water Wash Disposal - Because of the possibility of raw fuel being flushed out with the water and because of the concern for disposal of the detergent, the effluent from the water wash process will be collected and passed to a holding tank which will be sized to hold all the collected effluent for later disposal by manual means. Provisions for this ultimate disposal are TBD. To protect the system from freezing weather it will be necessary to drain the clear water from the water wash system after the wash cycle has been performed in order to avoid freezing damage. For this purpose the connection to a conventional drain will be made.

B. Power Supply - Three phase 460 volts 60 Hz power is available for operating the system pump. Single phase 115 volts 60 Hz also is available to power solenoid valves and the sensors required in the system.

C. Water Supply - A supply of potable water at a minimum of 35 psi is available to be plumbed into the water holding tank filling control.

D. Fire Control System Interface - If the installation is such as to require the alcohol anti-freeze feature, provisions will be made to tie into the fire control system provided with the powerplant installation.

E. Detergent Tank Fill - Provisions will be made to plumb the potable water supply to the detergent tank fill controls which will be manually operated in the process of mixing the cleaning agent with the water required to form the detergent solution.

VII. CONTROLS AND INSTRUMENTATION

A. Local Automatic Controls

1. Water Addition/Level - The water wash system will be provided with an automatic water fill feature to maintain the proper water level in the heated water holding tank. The detergent holding tank will not be equipped with an automatic fill feature.

2. Pressure - Automatic control of pressure will not be required.

3. Temperature - Temperature of the water in the holding tank will be controlled automatically to an adjustable level from 180°F to 190°F ±15°F using the electrical water heating element.

4. Shutdown - Initiation of the water wash cycle will be under the control of the gas turbine controller (GTC) which will monitor engine temperature, water level, water temperature in the holding tank, inlet and effluent water conductivity, pump outlet pressure and other signals to impede the startup or interrupt the operation of the water wash system in the event of an abnormal operating condition.

5. Fire Alarm/Extinguisher - If the alcohol antifreeze provisions are required by the installation, the site fire alarm/extinguisher system will be invoked.

6. Overpressure/Overtemperature Protection - Pressure relief provisions will be incorporated to bypass the pump in the event of a line blockage. The relief set point will be a maximum of 150 psi. The water holding tank temperature control will be backed up to limit the water temperature to a maximum of 205°F.

7. Water/Detergent Mix Ratio - This ratio will not be automatically controlled.

B. Local Manual Controls

1. Startup/Shutdown - Provisions will be made locally to override the automatic control to permit manual startup or shutdown of the water wash procedure.

2. Temperature/Pressure Adjustment - Manual adjustment of water temperature will be provided; however, no adjustment of pressure will be required.

3. Manual Operation - Manual operation of the drain system will be provided to allow draining the clean water from the system in the event of freezing weather. Suitable manual shut off valves will be provided to permit replacement of components without having to drain holding tanks.

4. Fire Extinguishing - The fire extinguishing system in the vicinity of the water wash system will be capable of being triggered by manual means.

5. Detergent Addition - The addition of detergent to the holding tank will be done by manual operation of suitable valves.

6. Water/Detergent Mix Ratio - The setting of mix ratio of water to detergent and water to alcohol will be done by manual operation of suitable valves.

C. Remote Control - Since the operation of the water wash system is through the electronic controller, remote initiation of startup, shutdown, fire alarm and fire extingishing will all be possible.

D. Local And Remote Instrumentation - Gages will be mounted on the water wash system to indicate pump outlet pressure, water temperature in the holding tank and sight gages to indicate fluid levels in the various holding tanks. No indication of flow rate or mixing ratios is required.

VIII. ENVIRONMENTAL CONDITIONS

A. Normal Environment For Equipment

1. Location - The water wash system will be installed in an unheated enclosure.

2. Temperature - The environmental temperature for normal operation will range from TBD to TBD.

3. Pressure - Normal operation will be provided under environmental pressures associated with altitudes from sea level to 7000 feet.

4. Humidity - Humidity levels ranging from TBD to TBD must be accommodated.

5. Externally Induced Vibration - The engine mounted components and connecting plumbing must be designed so as to accept the vibration levels resulting from engine operation.

B. System Impact On Local Environment

1. System Leakage - The system will be designed to minimize the leakage of detergent and alcohol. Provisions will be made to collect any leakage which may occur and to dispose of the fluid in a manner to preclude the possibility of contaminating the installation area. Design practice consistent with EPA and OSHA requirements must be adhered to.

2. System Noise - Pump speed and noise will be within acceptable engineering practices and OSHA regulations.

3. Heat Loss - In order to minimize the amount of heat required to control the water temperature under cold weather conditions, the water holding tank will be insulated to the extent permitted by cost effective design considerations.

C. Abnormal Environment For Equipment

1. Temperature Extremes - The system will be designed to withstand environmental temperatures ranging from -65°F to 130°F without damage in a conditioned non-operating mode.

2. Earthquake - The system will be designed to withstand seismic loads of 0.336 (2.1G max) horizontal (ground motion) on structure no damping. Similarly it will withstand 0.22G (1.41G max) vertical (ground motion) on structure no damping. Interconnecting piping will be designed in accordance with local requirements and building foundation design criteria.

3. Fire - In the event of indication of fire by the Fire Protection System, provision will be made for immediate isolation of any flammable fluid such as the alcohol by activation of suitably located solenoid valves. The system design will be such as to minimize the probability of its feeding any fire in the area.

4. Fire Suppression - The design of the system and selection of materials will be such that no damage to any equipment, wiring, plumbing or connections will result from the release of the fire suppression agents.

5. Conditions During Maintenance - The system will be designed to be protected from accidental damage by maintenance procedures applied to this or any adjacent system. Workmen stepping on critical components, possible damage due to dropped tools or swinging crane hooks must be taken into account in the system design.

6. Lightning - Provisions must be made for proper system grounding to minimize the impact of lightning strikes.

7. Dust And Dirt - All containers will be closed except for necessary vents which will be protected by screens and filters from wind blown dust and dirt.

8. Vermin And Insects - All system air intakes, exhausts, vents and breathers will be protected with screens. All cables and tubing will be armor protected against vermin.

9. Liquid Leakages From External Sources - The system will be designed and materials selected such that no system damage or impairment to life will result from exposure to any fluids employed in the powerplant installation.

IX. STRUCTURAL/PACKAGING REQUIREMENTS

A. Packaging Objectives And Requirements - The system package will be designed for ease of maintenance, emphasizing minimum down time for repair or replacement of parts. The packaging will be such as to protect the system from the elements and the abnormal environments cited above with the minimum use of footprint area.

B. Factors Of Safety/Codes - Normal engineering practice will be employed in sizing components, plumbing and wiring. Where design analysis shows component selections to be marginal for the service, the next larger component suitable for the service will be employed. Where analysis shows that life or reliability can be enhanced by the use of devices having greater performance or capacity margins than normal practice would dictate, the greater margins will be employed. The design of all plumbing will be consistent with ANSI B31.1.

C. Handling/Shipping Loads - The system will be designed to withstand handling loads of up to 3 G's without damage. The system also will be capable of withstanding the maximum loads imposed as a result of shipment by truck or by rail transportation.

D. Interface Requirements And Loads - All interfaces with the powerplant and the facility will be designed to hold stresses to components, plumbing and cabling well within safe limits as a result of thermal expansion, and normal and abnormal environmental conditions.

E. Provisions For Lifting/Mounting - An overhead crane and fork lift trucks will be available for lifting and moving the system and its subassemblies. All parts weighing 50 pounds or more will have provisions for lifting and trunnioning. Eye bolts will be provided for removing covers and permanent lifting lugs suitable for slings will be provided on sub modules where possible. The bulk of the system hardware will be skid mounted.

F. Servicing Considerations - The system will be designed to be serviced by simple procedures and components will be easily accessible to minimize down time associated with repair or replacement.

G. Cleaning And Flushing Criteria - The cleanliness of fluids delivered by the system will be assured by the use of suitable filters. In the event severe contamination of fluids in the various holding tanks occurs for any reason, as indicated by clogging of filters in short periods of operation, it will be necessary to flush clean the system fluid loops involved. Appropriate valving will be provided to permit such flushing on an individual fluid loop basis.

H. Size Limits - The system will be designed to be truck transportable.

I. Weight Limits - Individual module weights will be consistent with limits permitted by truck transportation.

X. RELIABILITY AND MAINTAINABILITY CRITERIA AND GOALS

A. Reliability Goals - All components will be selected for maximum reliability. Redundancy will be employed in the system design where it can be shown to improve system reliability. An MTBF of TBD will be used as a goal in the design of this system.

B. Maintenance Philosophy - To the maximum extent consistent with personnel safety, the water wash system will be designed to be serviced by simple procedures while the powerplant is operating. Requirements for special tools and fixtures are to be avoided. The system will be subjected to a monthly "walk-around" visual inspection. Maintenance will be performed only on a when-needed basis.

C. Component Access And Replacement Times - Use of modular construction will be emphasized to minimize downtime for service and replacement of components. The design of the module interfaces will be such as to facilitate access to connections and allow replacement of the module by use of simple procedures. Downtime for repair should not exceed one working shift as a goal. All cabling, external accessories, and plumbing should be capable of being checked for security of mounting and evidence of leakage or chafing. Filters will be easily accessible for replacement without loss of operating time.

D. Scheduled Down Time Criteria - Scheduled periods of down time for routine service other than that specified for the engine water wash is acceptable.

E. Spare Parts Philosophy And Guidelines - In the selection of components, preference will be given to equipment which is readily available. If long lead time parts must be employed and system availability could be jeopardized consideration will be given to the use of installed spares. Otherwise a policy of stocking spare long lead time components will be followed.

F. Design Life - In general the system should be designed for 30 year life; however, as a minimum, the water wash system will be capable of 5000 hours of operation over a 30 year period without major maintenance.

G. Restrictions On Cascading Failures - The water wash system will be designed to guard against the possibility of cascading failures initiated by the malfunction of any system component.

XI. SPECIFIC DESIGN GUIDELINES

A. Codes To Be Observed - The system plumbing will be designed to be consistent with ANSI B31.1. All electrical design will be consistent with the National Electrical Code and NEMA standards. The design of the alcohol subsystem will be in accordance with NFPA NO. 30. OSHA regulations will apply to the entire system design.

B. Materials To Be Banned - The materials will be selected to be compatible with all fluids with which they come in contact. All materials and processes used will be of high industrial quality and will not contribute to personnel or environmental hazard.

C. Hardware Types Not Suitable - Electrical equipment not consistent with Class I Div. 2 Category of the NEC is prohibited.

D. Safety Provision - The system will be consistent with NFPA No. 30 and OSHA regulations.

E. Other Guidelines - Maximum use will be made of standard off-the-shelf fasteners, flanges, connections and seals. Self locking features will be employed on connectors and fasteners to the maximum extent possible.

DESIGN CRITERIA DOCUMENT
IGNITION SYSTEM

I. INTRODUCTION

The purpose of this document is to define the initial design philosophy and guidelines for the Ignition system which supports the gas turbine in a combined cycle powerplant. The design criteria gives reliability equal priority with functionality, performance and cost in keeping with the goals of the program, EPRI 1187 High Reliability Gas Turbine Combined Cycle Development.

This design criteria is subject to review and revisions as the requirements and design evolves.

II. GENERAL DESIGN OBJECTIVE

The function of the ignition system is to provide high level electrical energy and to deliver this energy to spark igniters. The gas turbine is designed to operate on a wide variety of fuels. This includes gaseous fuels with a heating value as low as 100 BTU/SCF, alcohols including methyl fuel and ethanol and hydrocarbon liquid fuels including naphtha, petroleum distillates and selected crudes, residual fuels, coal derived and shale oil derived fuels containing fuel bound nitrogen. The ignition system will be designed to provide sufficient energy to ignite these fuels directly, or indirectly by ignition of special starting fuels. Activation of the ignition system will be controlled by the gas turbine controller (GTC).

III. SCOPE

The ignition system consists of a set of exciters, spark igniters and interconnecting cables. Each exciter and spark igniter is connected with an armor jacketed cable. The ignition system will have a spark rate and energy level capable of igniting all proposed fuels.

IV. OPERATING MODES

The powerplant is started by rotating the gas generators's compressor with a starter to provide sufficient pressurized air to support combustion in the burners. The starter drives the compressor until the powerplant is self-accelerating. The ignition system operates for a TBD time period of the start cycle.

V. ENGINE IMPOSED REQUIREMENTS

A. Fuels To Be Used - The powerplant will operate on a variety of gaseous and liquid fuels. The ignition system must have sufficient energy to ignite these fuels or a special fuel used during the start cycle.

B. How Energized - The gas turbine controller will activate and de-activate the ignition system.

C. Number Of Ignition Systems - The number of ignition systems required will be dictated by combustor design and/or the use of cross over tubes.

D. Where Mounted - The ignition system components will be engine mounted. The exact location will be set by engine design.

E. Number Of Operations - The exciters will be designed to accommodate a minimum of TBD powerplant starts over the life of the unit.

F. Special Provisions - This powerplant will have a water wash cycle during which all igniters will be exposed to water and possibly a detergent. No special drying procedures will be required for igniters following a water wash cycle; and the igniters will not be damaged as a result of being energized when wet.

VI. AVAILABLE SERVICES

A. Electrical Power - The following electrical services will be available to power the ignition system:

460 VAC, 60Hz, 115 VAC, 60Hz, and 125 VDC. The ignition system will be tied to the same power source as that for the starter motor.

B. Cooling Air - Should ignition system components require cooling compressor bleed air will be available.

VII. CONTROLS AND INSTRUMENTATION

A. Each ignition system will be energized by applying power to the exciter and de-energized by removing power from the exciter through the operation of a contractor in the engine controller.

VIII. ENVIRONMENTAL CONDITIONS

A. Normal Environment For Equipment

Locations - The ignition system will be located in an unprotected area and all parts will be corrosion resistant or treated with a protective coating. All components of the ignition system will be mounted on the engine.

Temperature - The exciter and cables must operate over the temperature range TBD to TBD°F. The spark igniter will be subjected to temperatures up to TBD°F. Cooling of the ignition system components is permitted but not preferred.

Vibration - The ignition system must be designed to operate at the engine induced vibration amplitudes and frequencies. Mil-STD-810C Procedure I - Table 514.2-II should be used as a guide.

B. System Impact On Local Environment

System Electrical Noise - The ignition system will not generate radiated and conducted electromagnetic emissions. Mil-STD-461A should be used as a guideline. The ignition exciter will incorporate provisions for suppressing inductive turn-off transients.

C. Abnormal Environment For Equipment

Temperature Extremes - The ignition system will be designed to withstand temperature extremes of -30 to 130°F without damage.

Earthquake - The ignition system must survive seismic loads of 0.33G (2.16 max) horizontal on structure no damping; and 0.22G (1.416 max) vertical on structure no damping.

Lightning - The ignition system's enclosures and cable shields will be properly grounded to avoid damage due to lightning.

Vermin - All cabling will be vermin proof.

Liquid Leakage From External Services - The system design and selected materials will be such that no damage or life impairment will result from exposure to any fluids employed in the powerplant installation including the fire protection system agent.

IX. STRUCTURAL/PACKAGING REQUIREMENTS

A. Packaging Objectives - Standard size enclosures and cables will be used where possible.

B. Factors of Safety - Normal aircraft engine design practice will be employed in sizing the components in the system. Components which are marginal for the service will not be used. Where life or reliability can be enhanced by the use of devices having greater performance margins than normal practice would dictate, the greater margins will be employed.

C. Handling And Shipping Loads - The ignition system will be shipped installed on the powerplant, and the installation design will be such that shipping loads experienced on the engine will not damage any component of the system.

D. Interfaces - All components of the system must interface through armored cables and connectors.

E. Mounting - The ignition system will be shock mounted on the engine in accordance with standard aircraft and industrial practice.

F. Servicing Considerations - In the design of the installation of the system, every consideration must be given to ease of servicing and replacement of parts by simple procedures.

G. Size Limits - The size limits for the components of the system will be governed by engine design and space available for installation. Cable lengths shall be kept to a minimum.

X. RELIABILITY AND MAINTAINABILITY CRITERIA AND GOALS

A. Reliability Goals - In the design of the ignition system, derating factors will be applied where necessary to provide reliable, long life operations.

Meantime between failure (MTBF) for the system and its components must be a minimum of TBD and/or consistent with a gas turbine starting reliability of 95%.

B. Maintenance Philosophy - The system design will be consistent with an "as-required maintenance" philosophy. There will be no periodic "walk around" visual inspections because the gas turbine will be operated continuous as a base load unit. In addition, the engine will have an acoustical cover that will make it difficult or impossible to inspect wiring on the igniters during operation; therefore, all cabling and external accessories will be designed with security of mounting and avoidance of chafing as important criteria.

C. Component Access And Replacement Time - All components of the ignition system will be installed so as to be readily accessible for minimum replacement time.

D. Scheduled Down Time - The ignition system will have no scheduled downtime. Spark igniters can be replaced when the engine undergoes hot section inspections at one year intervals.

E. Spare Parts Philosophy - Spares of long lead time items will be stocked in order to minimize engine downtime associated with component replacement.

F. Design Life - The complete ignition system, less the spark igniters, will be capable of operating on a powerplant for a minimum of TBD hours of electrical on-time while remaining within normal service operating limits.

G. Cascading Failures - Adequate protection will be provided to prevent cascading failures due to ignition system failures. Isolation from other electrical systems will be assured by proper fusing.

H. Drying Criteria - No special drying procedure will be required for the spark igniters because of the water wash cycle.

XI. SPECIFIC DESIGN GUIDELINES

A. Codes - The components of, and the assembled ignition system must conform to the latest issue of the National Electrical Code, with special attention to Articles 500 and 501 for Class I, Division 2 Groups B and D locations.

B. Materials - Materials must be compatible with all anticipated fuels; and materials which could produce galvanic effects must be avoided.

C. Hardware Types - All hardware used in the fabrication of the system must be of the best industrial quality and entirely suitable for the intended purpose. The exciters will be designed to maximize use of solid state electronic circuitry.

D. Safety Provisions - Positive locking features will be incorporated on connectors and fasteners to the maximum extent possible. Standard off-the-shelf fasteners will be used.

E. Electrical Cables and Connectors - The input electrical connector to the exciter will be self locking and will conform to the requirements of Mil-C-83723B; and the high-tension connectors will conform to the requirements of ARP-670-4.

DESIGN CRITERIA DOCUMENT
START SYSTEM

I. INTRODUCTION

The purpose of this document is to define the initial design philosophy and guidelines for the Start system which supports the gas turbine in a combined cycle powerplant. The design criteria gives reliability equal priority with functionality, performance and cost in keeping with the goals of the program, EPRI 1187 High Reliability Gas Turbine Combined Cycle Development.

This design criteria is subject to review and revisions as the requirements and design evolve.

II. GENERAL DESIGN OBJECTIVE

The function of the start system is to provide the required torque and speed to start the engine. In addition, the start system will be used to rotate the engine at the required speed during the wash cycles.

The start system will be designed as a highly reliable means to rotate the engine to light-off speed in the specified time period. In addition to normal start procedures, provisions will be made for aborted starts, instant restarts, water wash and shutdowns with continuous or cyclic rotation for uniform cooldown. Accommodations will also be made for reliable starting with the wide variety of liquid and gaseous fuels on which the engine is designed to operate. Activation of the start system will be controlled by the Gas Turbine Controller.

III. SCOPE

The start system consists of an AC induction motor and electrical accessories, input gear box, hydraulic torque converter, output speed reducing gearbox and an engine interfacing clutch. The system will also incorporate a turning gear box and barring motor for engine barring during startup and rotation of the powerplant during cooldown.

IV. OPERATING MODES

The start system rotates the gas turbine and connected electric generator at a speed to provide sufficient air at the required pressure to permit light off and sustain combustion in the burners. The start motor continues to drive the compressor until the engine is self-accelerating. Prior to starting, the powerplant will be barred for TBD minutes at a minimum of 50 (TBD) RPM to assure the formation of an oil film on all bearings. The gas turbine and generator will be barred for 20 hours after shutdown and for a period of TBD hours every two weeks if not in operation. For water washing, the start system interface clutch will engage the powerplant at TBD RPM during slow down and provide motor rotation as needed during the wash cycle. There is no requirement for black start or synchronous condenser operation.

V. ENGINE IMPOSED REQUIREMENTS

A. Light-Off Speed - The engine is designed to operate on a variety of fuels from low BTU gases to selected liquid crudes as well as a special starting fuel. The light-off speed is 1000 RPM TBD.

B. Engine Self-Sustaining Speed - The start system will have sufficient horsepower and speed to maintain the powerplant at TBD RPM until self-sustaining combustion and self-acceleration is achieved.

C. Starting-Time - The start system will be designed to provide a sustained start in 15 minutes.

D. Inertias And Breakaway Torque - The inertias of the gas turbine and breakaway torque and generator are:

	Generator	Combustion Turbine	Total
Rotor Weight	80,000 (TBD)	40,000 (TBD)	120,000 (TBD)
Inertia lb Ft^2	96,000 (TBD)	161,000 (TBD)	257,000 (TBD)
Breakaway Torque w/o Jacking lb Ft	10,600 (TBD)	6,700 (TBD)	17,300 (TBD)
Breakaway Torque with Jacking	2,800 (TBD)	1,800 (TBD)	4,600 (TBD)

E. Torque-Speed - Torque vs. speed requirements are TBD.

F. Abnormal Operating Events - The start system will be designed to permit the interface clutch to engage the shaft while the engine is rotating. This feature will allow abort starts to be restarted without waiting for the powerplant to come to a stand-still.

VI. AVAILABLE SERVICES

A. Electrical Power - The following electrical services will be available to power the start system components.

1 - 460 VAC, 60 Hz three phase
2 - 115 VAC, 60 Hz single phase
3 - 125 VDC emergency power

B. Cooling - Auxiliary cooling water at 95°F is available.

C. Fire Protection - A fire protection system is provided for the installation, however no special ties with this system are anticipated other than that necessary to abort an attempted start in the event of a fire.

D. Lubricators - A circulating lubricating oil system and a source of hydraulic control oil are available for use by the start system. A mineral base lubricating oil at TBD psi pressure and TBD temperature is available from the GT lube oil system.

VII. CONTROLS & INSTRUMENTATION

A. Automatic Control - The gas turbine controller will automatically control the following startup related functions:

1. Startup from stand-still involving the use of the barring system for bearing oil film formation, and acceleration to light off RPM with continuous drive to self sustaining combustion.

2. Shutdown in a normal deceleration mode with the barring system interfacing at TBD engine rpm for cool down.

3. Barring for 20 hours after shutdown.

4. Water wash of the gas generator will be done at TBD intervals at a TBD powerplant speed. The engine will be rotated with the start motor during the wash cycle.

5. Emergency shutdowns or abort starts will be initiated by the powerplant controller monitoring critical parameters.

B. Condition Monitoring - Instrumentation will be provided with sensing of motor current, motor temperature and torque converter pressure.

VIII. ENVIRONMENTAL CONSIDERATIONS

A. Normal Environment For Equipment

Location - The entire start system will be located inside an unheated building and protected fron normal adverse elements.

Temperature - The start system including barring motor must operate over the temperature range TBD & TBD.

Humidity - Performance of the start system will be unaffected by relative humidities from TBD & TBD % at TBD°F ambient temperature.

Vibration - The start system must be designed to operate at the engine induced vibration amplitudes and frequencies when coupled to the engine.

B. System Impact On Local Environment

Noise - The system induced acoustic noise level will be masked by that produced by the powerplant and should not be a problem.

System Induced Vibration - System induced vibration levels must not affect performance of the start system or powerplant.

C. Abnormal Environment For Equipment

Temperature Extremes - The start system will be designed to withstand temperature extremes of -65 to 130°F without damage.

Fire Protection - The system design and material selections will be such that no damage to any equipment, wiring, plumbing or connections will result from exposure to the fire suppression agent.

Lightning - Proper grounding techniques will be used to avoid damage due to lightning.

Dust And Dirt - All housings must be fully enclosed. Any vents must be protected from dust and dirt with screens and filters.

Earthquakes - The start system must be designed to withstand seismic loads of 0.33G (2.16 max) horizontal on structure no damping; and 0.22G (1.416 max) vertical on structure no damping.

Vermin And Insects - All cables and tubing will be armor protected against vermin. All vents and breathers will be protected against insects by suitable screens.

IX. STRUCTURAL/PACKAGING REQUIREMENTS

A. Packaging Objectives - The system package will be designed for ease of maintenance, emphasizing minimum downtime for repair or replacement of parts.

B. Factors Of Safety - Normal engineering practice will be employed in sizing components, plumbing and wiring. Where design analysis shows component selections to be marginal for the service, the next larger component suitable for the service will be employed. Where analysis shows that life or reliability can be enhanced by the use of devices having greater performance or capacity margins than normal practice would dictate, the greater margins will be employed. The design of all plumbing will be consistent with ANSI B31.1.

C. Handling And Shipping Loads - The system will be designed to withstand handling loads of up to 3 G's without damage. The system will also be capable of withstanding the maximum loads imposed as a result of shipment by truck or rail.

D. Interface Requirements - All interfaces with the powerplant and the facility will be designed to hold stresses to components, plumbing and cabling well within safe limits as a result of thermal expansion, and normal and abnormal environmental conditions.

E. Lifting And Mounting Provisions - A portable overhead crane and fork lift trucks will be available for lifting and moving the system. All parts weighing 50 pounds or more will have provisions for lifting and trunnioning.

F. Servicing Considerations - In the design of the system, every consideration must be given to ease of servicing and replacement of parts by simple procedures to minimize downtime. Clearance around the components will be such as to make them easily accessible for repair or replacement. All electrical interfaces will be raised to a central easily accessible terminal box.

H. Size Limits - The size limits on the assembled system will be governed by the space available in the enclosure.

I. Weight Limits - The system weight is not critical, but it will be consistent with the requirement for truck transportability.

X. RELIABILITY AND MAINTAINABILITY CRITERIA AND GOALS

A. Reliability Goals - All components will be selected for maximum reliability. Derating factors will be used where feasible to provide reliable long life operation. A starting reliability of 99% will be used as a goal in the design of this system.

B. Maintenance Philosophy - The start system will be designed to be serviced by simple procedures. Requirements for special tools and fixtures are to be avoided. The system will be subjected to a monthly "walk-around" visual inspection. Maintenance will be performed on a when-needed basis.

C. Component Access And Replacement Times - Use of modular construction will be emphasized to minimize downtime for service and replacement of components. The design of interfaces will be such as to facilitate access to connections and allow component replacement by simple procedures. Downtime for repair should not exceed one working shift as a goal.

D. Scheduled Down Time - There will be no scheduled periods of downtime for routine service other than that specified for the engine which is no more frequent than every 9000 hours.

E. Spare Parts Philosophy - In the selection of components, preference will be given to equipment which is readily available. Spares of long lead time items will be stocked to minimize downtime.

F. Design Life - In general the system should be designed for 30 year life. As a minimum, the start system will be capable of provided a 100 TBD starts per year.

G. Restrictions On Cascading Failures - The start system will be designed to guard against the possibility of cascading failures initiated by the malfunction of any system componeents.

XI. SPECIFIC DESIGN GUIDELINES

A. Codes - All electrical design will be consistent with NFPA-70 for Class I Division 2 Groups B and D and NEMA standards. The system plumbing will be designed to be consistent with ANSI B31.1. OSHA regulations will apply to the entire system design.

B. Materials - The materials selected in the design of the system will be compatible with all fluids with which they come in contact. All materials and processes used will be of high industrial quality and will not contribute to personnel or environmental hazard.

C. Hardware Types - Electrical equipment not consistent with Class 1 Div. 2 Groups B and D category of the NEC is prohibited.

D. Safety - System safety will be consistent with NFPA No. 30, NFPA No. 70 and OSHA relations.

E. Other Guidelines - Maximum use will be made of standard off the shelf hardware such as fasteners, piping, flanges, connectors and seals. All materials and hardware will be of the best commercial quality and entirely suited for the intended usage. Self locking features will be employed on connectors and fasteners to the maximum extent possible.

DESIGN CRITERIA DOCUMENT
FIRE PROTECTION SYSTEM

I. INTRODUCTION

The purpose of this document is to define the initial design philosophy and guidelines for the Fire Protection system which supports the gas turbine in a combined cycle powerplant. The design criteria gives reliability equal priority with functionality, performance and cost in keeping with the goals of the program, EPRI 1187 High Reliability Gas Turbine Combined Cycle Development.

This design criteria is subject to review and revisions as the requirements and design evolve.

II. GENERAL DESIGN OBJECTIVES

The function of the fire protection system for the High Reliability Gas Turbine is to minimize damage and downtime due to a fire and/or the method used to extinguish the fire.

The fire protection system will be designed for the automatic detection of the on-set of fire. It will sound an alarm and extinguish the fire, keeping damage to a minimum. The system will employ HALON 1301 for the extinguishing agent. Typical combustible materials in the site are liquid fuels, gaseous fuels, lube oil, hydraulic fluid, and alcohol used in the water wash system. The system must also be designed to extinguish Class "C" electrical fires. Fire detectors will be rate compensated thermal types located in all enclosures. When the detected temperature rise is sufficient to predict that a fire is occurring, the detection system will automatically actuate the fire extinguishing system. Provision for manual actuation will be included in the design. Combustible gas detector heads will be located in all enclosures, but will be used only for alarm and gas fuel shutoff functions. Signals from the combustible gas detectors will not be used to activate the fire suppression system. If smoke detectors are installed in the enclosures and control room, their alarm signals will not be used to activate the fire suppression system.

III. SCOPE

The fire protection system consists of two sub-systems: detection and extinguishing. The fire detection sub-system will employ a TBD number of rate compensated thermal detectors located to survey critical areas in each enclosure. The fire extinguishing subsystem will be of the total flooding type and employ Halon 1301 as the extinguishing agent. Activation of the extinguishing subsystem will be automatically triggered by a signal from any one of the thermal detectors or manually by switches in the control house and outside the gas turbine enclosure.

IV. OPERATING MODES

The fire protection system will be in operation at all times regardless of powerplant actvity or shutdown. When the detected temperature rise within an enclosure is sufficient to predict that a fire is occurring, the detection system will automatically actuate the fire extinguishing system, cause the fire valves and fire dampers to close, signal the emergency shutdown relays to shut down the gas turbine and generator and alarm and annunciate this condition. The alarm will be sounded as soon as a fire trip occurs, but there will be a five-second delay before discharge of the extinguished agent. There will be no delay when the system is actuated manually. An explosive will be used to rupture the container valve diaphragm which instantly releases the agent into the discharge system.

V. INSTALLATION IMPOSED REQUIREMENTS

A. Nature Of Combustible Materials - The fire protection system will be capable of extinguishing Class "A" fires involving the following combustibles:

1. Gaseous fuels such as natural gas and coal derived low and medium BTU gases.

2. Liquid fuels including alcohols, light hydrocarbons such as naphtha, petroleum distillates and heavy crudes.

3. Turbine lube oil.

4. Hydraulic fluids.

5. Plastics and synthetic rubber.

6. Materials involved in Class "C" electrical fires.

B. Fire Suppression Agents - Halon 1301 (Bromotrifluoromethane) will be the fire extinguishing agent.

C. Types Of Sensors - The sensors sused will be rate-compensated thermal detectors. The number of detectors and their locations in each enclosure are TBD.

D. Areas To Be Protected - The areas to be protected are enclosed, and include the enclosures for the gas turbine and the generator and start system.

E. Time To Discharge System - The system will be designed to allow complete discharge of the agent in less than 10 secs. Design parameters to be considered include: plumbing size and length of runs, type and number of discharge nozzles, the system pressure and number and location of the agent storage containers.

F. Ability To Test System - Provisions will be made for continuous monitoring of the electrical continuity of the thermal detection and extinguishing system. Monitoring will be accomplished by a continuity check light located in the plant control room.

G. Installed Reserve Storage Of Agent - There will be at least two separate agent storage tanks with suitable transfer valves arranged to provide full availability of agent while one tank is being serviced. The agent reserve storage container will be designed to hold Halon 1301 in liquified form at ambient temperature. They will not be charged to a filling density greater than 70 pounds per cubic foot, and will be super-pressurized with dry nitrogen to 360 psig total pressure at 70°F. The storage tanks will meet other requirements as outlined in paragraph 1550 of NFPA #12A. The assembly will be designed for a maximum ambient temeprature of 130°F as specified in paragraph 1550 of NFPA 12A. The capacity of each storage tank will be great enough to attain a minimum agent concentration of 6% in the enclosure in 10 seconds to meet NFPA requirments. The leakage of agent will be zero.

VI. AVAILABLE SERVICES

A. Agent Storage And Supply Data - A weighing system will be available to check quantity of agent in storage at TBD time intervals. A supply of dry nitrogen will be available for super-pressurizing the storage container.

B. Power Supply - Three-phase 460 volt 60 Hz power is available if required. Single phase 110 volt 60 Hz is available to power solenoid valves, sensors and relays. An emergency power supply of 128 volts DC is also available.

C. Site Interface - Enclosure vents will be available to dissipate the Halon after use.

VII. CONTROL AND INSTRUMENTATION

A. Automatic Controls

System Activation - The system will be activated when the temperature in the enclosure reaches the thermostat setting of any detector. Closing of the normally open switch element will energize the shutdown/alarm relays.

Delayed Discharge - In the automatic control mode, a five-second delay between alarm and agent discharge will be incorporated into the electric circuit to permit personnel to vacate the enclosure.

Alarm - An audible alarm will be energized at the instant of system activation. The alarm will have an outstanding sound level and distinctive tone to alert personnel to the fire and the imminent discharge of the fire suppression agent.

Engine Shutdown - The signal from the detector will energize relays to automatically shutdown the powerplant. The signal will also close fuel supply fire valves, shut off vent fans and close fire dampers in the enclosure. NOTE: In some installations, the fire dampers are activated by pressure taps in the discharge piping. This option will be considered in the reliability study.

B. Manual Controls

Manual Activation - The fire suppression system will be capable of being manually activated by conveniently located switches in the control room and outside the gas turbine enclosure. There will be no time delay between alarm and agent discharge when the system is activated manually.

Alarm - A manual acknowledge will be provided by means of a switch which will shut off the audible alarm. A manual reset switch will be provided to restore the system to a shutdown condition including the visual alarms.

Shut Off - Agent discharge valves will be closed by manual operation of the reset switch in preparation for refilling of the system.

C. Remote Controls

Manual Activation - Activation of the fire suppression system will be possible by means of switches in the Control Center and outside of each enclosure.

Delayed Discharge - There will be no time delay between the alarm and agent discharge when the system is manually triggered.

Alarm - The alarm will be acknowledged by the procedure previously described.

D. Local And Remote Instrumentation - The agent storage pressure will be measured and monitored by a gage at the storage location and in the Control Center.

VIII. ENVIRONMENTAL CONDITIONS

A. Normal Environment For Equipment

Location - The storage containers will be located outside the powerplant enclosures.

Temperature - The ambient temperature for the agent will be within the range -20 to 130°F. The temperature range in the enclosures at the location of detectors and nozzles will be from TBD to TBD.

Pressure - Normal operation will be provided under environmental pressures associated with altitudes from sea level to 7000 feet.

Humidity - Humidity levels ranging from TBD to TBD will be accommodated.

Externally Induced Vibration - The system will be designed to withstand vibration levels resulting from engine operation within the facility.

B. System Impact On Local Environment - The agent is slightly toxic and, therefore, provisions will be made to check for leakage. The system will be designed for zero leakage.

C. Abnormal Environment For Equipment

Temperature Extremes - The system will be designed to withstand environmental temperatures ranging from -65°F to 130°F without damage.

Earthquake - The system will be designed to withstand seismic loads of 0.336G (2.16 G max) horizontal (ground motion) on structure no damping. Similarly, it should withstand 0.22G (1.41 G max) vertical (ground motion) on structure no damping. Inter-connecting piping will be designed in accordance with local requirements and building foundation design criteria.

Conditions During Maintenance - The system will be designed to be protected from accidental damage by maintenance procedures applied to this or any adjacent system. Workmen stepping on critical components, possible damage due to dropped tools or swinging crane hooks must be taken into account in the system design.

Lightning - Provisions must be made for proper grounding to minimize the impact of lightning strikes.

Dust And Dirt - Provisions will be made to protect all components and plumbing from dust and dirt prior to and during installation and at any time it is unsealed for maintenance.

Vermin And Insects - All cables and tubing will be armor protected against vermin.

Liquid Leakage From External Sources - The system will be designed and materials selected such that no system damage or impairment to life will result from exposure to any fluids employed in the powerplant installation.

IX. STRUCTURAL/PACKAGING REQUIREMENTS

A. Packaging Objectives And Requirements - The system package will be designed for ease of inspection and maintenance, emphasizing minimum downtime for recharging the system and for repair or replacement of parts. The packaging will be such as to protect the system from the elements and the abnormal environments cited above.

B. Factor Of Safety/Codes - Normal engineering practice will be employed in sizing components, plumbing and wiring. Where design analysis shows component

selection to be marginal for the service, the next larger component suitable for the service will be employed. Where analysis shows that life or reliability can be enchanced by the use of devices having greater performance or capacity margins than normal practice would dictate, the greater margins will be employed.

C. Handling/Shipping Loads - The system will be designed to withstand handling loads of up to 3 G's without damage. The system will also be capable of withstanding the maximum loads imposed as a result of shipment by truck or rail.

D. Interface Requirements And Loads - All interfaces with the powerplant enclosure and the facility will be designed to hold stresses to components, plumbing and cabling well within safe limits as a result of thermal expansion, and normal and abnormal environmental conditions.

E. Servicing Considerations - The system will be designed to be serviced by simple procedures and components will be easily accessible to minimize downtime associated with repair or replacement.

F. Size Limits - The system will be designed to be truck-transportable.

G. Weight Limits - The weight of the agent container will be consistent with limits permitted by truck transportation.

X. RELIABILITY AND MAINTAINABILITY CRITERIA AND GOALS

A. Reliability Goals (MTBF) - All components will be selected for maximum reliability. Redundancy will be employed in the system design where it can be shown to improve system reliability. An MTBF of TBD will be used as a goal in the design of this system.

B. Availability Goals - This system must be designed for 100% availability except immediately following a fire.

C. Maintenance Philosophy - To the maximum extent consistent with personnel safety, the fire suppression system will be designed to be serviced by simple procedures while the powerplant is operating. Requirements for special tools and fixtures are to be avoided. The system can have periodic "walk-around" visual inspection. Maintenance will be performed only on an as-required basis. There

will be a continuous monitoring of electrical continuity to the detectors and the pressure in the agent storage system.

D. Component Access And Replacement Times - All components of the system will be easily accessible. The design of component interfaces will be such as to facilitate access to and replacement of the components using simple procedures. All cabling, external accessories, and plumbing will be capable of being checked for security of mounting and evidence of leakage or chafing.

E. Scheduled Downtime Criteria - There will be no scheduled periods of downtime for routine service.

F. Spare Parts Philosophy And Guidelines - In the selection of components, preference will be given to equipment which is readily available. If long lead time parts must be employed and system availability could be jeopardized, consideraton will be given to the use of installed spares. Otherwise, a policy of stocking spare long lead time components will be followed.

G. Design Life - In general, the system should be designed for infinite life; however, as a minimum the system should be capable of standby operation over a 30-year period without major maintenance.

XI. SPECIFIC DESIGN GUIDELINES

A. Codes To Be Observed - The overall system will be designed consistent with NFPA #12. The system plumbing will be designed in accordance with NASI.B31.3. All electrical design will be consistent with the National Electric Code and NEMA standards. OSHA regulations will apply to the entire system design.

B. Materials To Be Banned - The materials will be selected to be compatible with all fluids with which they come in contact. All materials and processes used will be of high industrial quality and will not contribute to personnel or environmental hazard.

C. Hardware Types Not Suitable - Electrical equipment not consistent with Class I Div. 2 Groups B and D Category of the NEC is prohibited.

D. Safety Provisions - The system will be consistent with NFPA No. 12A and OSHA regulations.

E. Other Guidelines - Maximum use will be made of standard off-the-shelf fasteners, flanges, connections and seals. Self-locking features will be employed on connectors and fasteners to the maximum extent possible.

DESIGN CRITERIA DOCUMENT
ENGINE HEATER SYSTEM

I. INTRODUCTION

The purpose of this document is to define the initial design philosophy and guidelines for the Engine Heater system which supports the gas turbine in a combined cycle powerplant. The design criteria gives reliability equal priority with functionality, performance and cost in keeping with the goals of the program, EPRI 1187 High Reliability Gas Turbine Combined Cycle Development.

This design criteria is subject to review and revisions as the requirements and design evolve.

II. GENERAL DESIGN OBJECTIVES

The function of the engine heater system is to provide a supply of heated air to maintain the temperature of the engine above the dewpoint of the ambient air in the enclosure. The heater system is required to prevent moisture from condensing inside the engine resulting in the corrosion of engine components.

III. SCOPE

The major components making up the engine heater systems are a blower, an electric heater, flexible ducting, pressure switch, temperature control, check valves, and an air filter. The heated air is transferred to the engine by means of the flexible ducting which is connected to cooling tube flanges mounted on the gas generator at the inter compressor stage. The heated air is discharged through the exhaust and inlet of the engine.

IV. OPERATING MODES

This system will be put into operation when the powerplant is scheduled for an extended shutdown period of time. Sufficient heated air will be forced into the

engine to maintain the temperature of the internal components above the ambient dew point. Startup and shutdown of the Engine Heater System is controlled automatically by the unit sequencer.

V. ENGINE IMPOSED REQUIREMENTS

A. Heater Size - The heater will be sized to maintain the required air flow at a temperature approximately 40°F above ambient air temperature.

B. Air Flow Rates - The air flow rate will be established by heat transfer analysis and will be at a level which will result in a 3°F engine temperature rise with heater system exhaust air temperature 40°F above the ambient temperature.

C. Pressure - The system outlet pressure will be established by analysis of engine pressure drop characteristics.

D. Filtration - The blower inlet filter will be capable of filtering particles of 5 micron size and larger with a minimum of 99% efficiency. The pressure drop across the inlet filter will not exceed TBD inches of water column at rated blower flow rate.

E. Temperature - The heater will provide output air at a temperature of 40°F ± 5° above ambient temperature.

VI. AVAILABLE SERVICES

A. Power Supply - Three phase 460 volts 60 Hz power is available for operating the blower motor and air heaters. Single phase 115 volts 60 Hz also is available to power control components and sensors.

VII. CONTROLS AND INSTRUMENTATION

A. Automatic Controls

Temperature - The air temperature will be automatically controlled within the system by means of a thermostat sensing outlet air temperature and controlling power to a heater in on/off fashion.

System Activation - The system will be activated by a control signal from the sequencer. The blower and heater will start simultaneously.

B. Manual Controls

System Activation - Manual activation of the system will be possible by a switch located in the control center.

Temperature Adjust - It will be possible to adjust the setting of the temperature control by manual means in the control room.

C. Local And Remote Instrumentation For Condition Monitoring And Diagnostics

Temperature - A thermocouple(s) will be provided to monitor the temperature of the heated air with visual readout in the control room.

Presence Of Flow - Verification of air flow through the gas generator will be measured and monitored by air flow switches.

VIII. ENVIRONMENTAL CONDITIONS

A. Normal Environment For Equipment

Location - The engine heater sytem will be designed for operation inside an unheated enclosure.

Temperature - The environmental temperature for normal operation will range from TBD to TBD.

Pressure - Normal operation will be provided under environmental pressures associated with altitudes from sea level to 7000 feet.

Humidity - Ambient air humidity levels ranging from TBD to TBD must be accomodated.

Externally Induced Vibration - The system must be designed to operate at the vibration levels induced by the operation of any other powerplant in the enclosure and any other ancillary equipment.

B. System Impact On Local Environment

System Leakage - The system will be designed to minimize leakage of heated air. Design practice consistent wth OSHA regulations must be adhered to.

System Noise - Noise produced by the blower will be within the limits prescribed by OSHA regualtions.

Heat Loss - In order to minimize the amount of electrical power used to heat the air to the required temperature, the ducting system will be insulated to the extent permitted by cost effective design considerations.

System Induced Vibration - An analysis will be performed to assess the impact of vibrations generated by the blower and its motor and to establish the need for isolation to preclude the possibility of stresses to components in this and any other installed system.

C. Abnormal Environment For Equipment

Temperature Extremes - The system will be designed to withstand environmental temperatures ranging from -65°F to 130°F without damage in a non-operating mode.

Earthquake - The system will be designed to withstand seismic loads of 0.33G (2.1 G max) horizontal (ground motion) on structure no damping; and 0.22 G (1.41 G max) vertical (ground motion) on structure no damping.

Fire - In the event of indication of fire by the Fire Protection System, provision will be made for automatic shutdown of blower motor and air heaters.

Fire Suppression - The design of the system and selection of materials will be such that no damage to any equipment, wiring, ducting or connections will result from the release of the fire suppression agents.

Conditions During Maintenance - The system will be designed to be protected from accidental damage by maintenance procedures applied to this or any adjacent system. Workmen stepping on critical components, possible damage due to dropped tools or swinging crane hooks must be taken into account in the system design.

Lightning - Provisions must be made for proper system grounding to minimize the impact of lightning strikes.

Dust And Dirt - The blower inlet will be equipped with an air filter capable of filtering particles of 5 micron size. In addition, provisions will be made to protect all components and duct work from dust and dirt prior to and during installation and at any time it is disconnected from the powerplant.

Vermin And Insects - The system air intake will be protected from insect ingestion by the inlet air filter. All cables and tubing will be armor protected against vermin.

Liquid Leakage From External Sources - The system will be designed and materials selected such that no system damage or impairment to system life will result from exposure to any fluids employed in the powerplant installation.

Foreign Object Ingestion - A "last chance" screen will be installed downstream of the heater to protect the engine from system component failure.

IX. STRUCTURAL/PACKAGING REQUIREMENTS

A. Packaging Objectives And Requirements - The system package will be designed for ease of inspection and maintenance, emphasizing minimum downtime for repair or replacement of parts. The packaging will be such as to protect the system from the normal and abnormal environments cited above with the minimum use of footprint area.

B. Factors Of Safety/Codes - Normal engineering practice will be employed in sizing components, ducting and wiring. Where design analysis shows component selection to be marginal for the service, the next larger component suitable for the service will be employed. Where analysis shows that life or reliability can be enhanced by the use of devices having greater performance or capacity margins than normal practice would dictate, the greater margins will be employed. Blower inlet design will be consistent with OSHA regulations.

C. Handling, Shipping Loads - The system will be designed to withstand handling loads of up to 3 G's without damage. The system will also be capable of withstanding the maximum loads imposed as a result of shipment by truck or rail transportation.

D. Interface Requirements And Loads - All interfaces with the powerplant and the facility will be designed to hold stresses to components, ducting and cabling well within safe limits as a result of thermal expansion, and normal and abornmal environmental conditions.

E. Provisions For Lifting/Mounting - An overhead crane and fork lift trucks will be available for lifting and moving the system and its subassemblies. All parts weighing 50 pounds or more will have provisions for lifting and trunnioning. Eye bolts will be provided for removing covers and permanent lifting lugs suitable for slings will be provided on submodules where possible. The bulk of the system hardware will be skid mounted.

F. Servicing Considerations - The system will be designed to be serviced by simple procedures and components will be easily accessible to minimize downtime associated with repair or replacement.

G. Size Limits - The system will be designed to be truck transportable.

H. Weight Limits - Individual module weights will be consistent with limits permitted by truck transportation.

X. RELIABILITY AND MAINTAINABILITY CRITERIA AND GOALS

A. Reliability Goals - All components will be selected for maximum reliability. Redundancy will be employed in the system design where it can be shown to improve system reliability. An MTBF of TBD will be used as a goal in the design of this system.

B. Maintenance Philosophy - To the maximum extent possible consistent with personnel safety, the engine heater system will be designed to be serviced by simple procedures while the powerplant is operating. Requirements for special tools and fixtures are to be avoided. During system operation, the system will be subjected to periodic "walk-around" visual inspection. Maintenance will be performed only on a when-needed basis.

C. Component Access And Replacement Times - All components of the system will be easily accessible. The design of component interfaces will be such as to facilitate replacement of the components using simple procedures. All cabling, external accessories and ducting will be capable of being checked for security of mounting and evidence of leakage or chafing.

D. Scheduled Down Time Criteria - There will be no scheduled period of downtime for routine service.

E. Spare Parts Philosophy And Guidelines - In the selection of components, preference will be given to equipment which is readily available. If long lead time parts must be employed and system availability could be jeopardized, a policy of stocking spare long lead time components will be followed.

F. Design Life - In general the system should be designed for infinite life, as a minimum, the engine heater system should be capable of TBD hours of operation over a 30 year period without major maintenance.

G. Restrictions On Cascading Failures - The engine heater system will be designed to guard against the possibility of cascading failures initiated by the malfunction of any system component.

XI. SPECIFIC DESIGN GUIDELINES

A. Codes To Be Observed - All electrical design will be consistent with the National Electric Code NEMA standards. OSHA regulations will apply to the entire system design.

B. Materials To Be Banned - The materials will be selected to be compatible with all fluids with which they come in contact. All materials and processes used will be of high industrial quality and will not contribute to personnel or environmental hazard.

C. Hardware Types Not Suitable - Electrical equipment not consistent with Class I Division 2 Groups B and D of the National Electric Code is prohibited.

D. Safety Provisions - The system design will include interlocks to insure that the heaters cannot be put into operation without airflow. Personnel safety considerations will be consistent with OSHA regulations.

E. Other Guidelines - Maximum use will be made of standard off-the-shelf fasteners, flanges, connections and seals. Self locking features will be employed on connectors and fasteners to the maximum extent possible.

DESIGN CRITERIA DOCUMENT
HYDRAULIC SERVO SUPPLY SYSTEM

I. INTRODUCTION

The purpose of this document is to define the initial design philosophy and guidelines for the Hydraulic Servo system which supports the gas turbine in a combined cycle powerplant. The design criteria gives reliability equal priority with functionality, performance and cost in keeping with the goals of the program, EPRI 1187 High Reliability Gas Turbine Combined Cycle Development.

This design criteria is subject to review and revisions as the requirements and design evolve.

II. GENERAL DESIGN OBJECTIVES

The function of the hydraulic servo supply system is to act as a source of control oil at the proper pressure, temperature and cleanliness for use with long life close tolerance components such as valves, actuators, and pumps. Since proper operation of these various powerplant components is critical to the continued operation of the powerplant the reliability of the hydraulic servo supply system is critical to the reliability and availability of the power generation system. Special care will be taken in the system design to assure uninterrupted operation during any operating mode. The powerplant availability requirements make special demands. The hydraulic servo supply system design will allow maintenance and component replacement during operation. The design of the system will not preclude potential future gas turbine performance up-rating.

III. SCOPE

The system will consist of an oil reservoir which is capable of being electrically heated in cold weather, redundant oil pumps, a water cooled heat exchanger, a filter system with associated transfer valves, and accumulator, pressure regulators,

multidirectional control valves, check valves, shutoff valves (both manual and solenoid), relief valves, control valves, sensors and gages. The system will be plumbed and wired to interface with the engine control requiring the hydraulic fluid, the powerplant automatic controller, alarm annunciators and site a-c and d-c power sources. Operation of the hydraulic servo supply system will be fully automatic in terms of control of pressure and temperature and in the sequencing of system components during startup, normal and emergency running, shutdown and other modes of operation that might require the source of hydraulic power.

IV. OPERATING MODES

The hydraulic servo supply system will be required to operate continuously under all powerplant modes in which fuel is supplied to the gas turbine. It will be fully operational prior to a startup, to permit proper functioning of the fuel controls during that phase of operation. Since the fuel control system will be designed as a fail-safe system it will be possible for the powerplant to be shutdown in an emergency mode without the availability of hydraulic fluid; however, in order to achieve a normal orderly shutdown the hydraulic servo supply system will be functional through the shutdown period. Since the hydraulic system is available for use in other ancillary systems in the powerplant installation it may be necessary that the hydraulic system be operational for other operating modes such as in the barring mode. Definition of this requirement must await firming up of the powerplant design and its ancillaries.

V. ENGINE IMPOSED REQUIREMENTS

A. Components Requiring Hydraulic Drive Muscle - As a minimum the following components will require hydraulic fluid for purposes of actuation: fuel modulating valve, fuel shutoff valve, fuel dump valve, quick fill bypass valve for fuel, fuel pump startup bypass valve, fuel divider bypass valve and the inlet guide vane control. Other components in the installation may require the services of the hydraulic supply and these will be identified as the powerplant design is better defined.

B. Flow Rates - The maximum flow rate capability of the system will be sized to accommodate the simultaneous demands of all components tied to the system with a margin of 20%. The actual maximum flow rate is TBD. Since some of the components requiring hydraulic fluid are engine mounted and will be subject to elevated temperature, the system will be designed to provide a small circulating flow in normally

dead ended control loops to avoid possible coking of fluid due to the high temperature. The minimum flow required for this purpose is TBD. The oil reservoir will be sized to provide enough oil for three minutes of flow at the maximum flow rate. It will have volume margin sufficient to accommodate all of the fluid contained in the component and plumbing in the event the system must be drained for the purpose of maintenance.

C. Pressure - The maximum system pressure will be established by the size and the force requirements of the control components to be actuated by the fluid. It is estimated that the maximum supply pressure will be about 1,000 psi. Suitable pressure regulators will be provided to control the pressure to a level of TBD plus or minus 5% (estimated at 500 ± 25 psi) for most of the actuators. A pressure relief bypass around the pump, set for 1,000 TBD psi, will be provided to limit the maximum pressure. A gas charged bladder type accumulator will be provided to absorb pressure transients in the system.

D. Filtration - A full flow duplex filter system will be provided on the supply line feeding the high tolerance control components. The filter will be rated for 5 microns absolute particle service. Manually operated transfer valving will be provided to allow for filter cartridge replacement without interruption of service. A differential pressure switch across the filter will be provided to alarm the need for change of the filter cartridge. A pump suction line strainer will be installed outside the reservoir with bypass and isolation valves so as to permit its being cleaned without interruption of service.

E. Special Transient Requirements - The system will be designed so as to maintain regulated pressure within 10% of the nominal control value for a 75% of maximum flow step change. The selection of relief valve types and pressure settings will be such as to avoid the excessive pressure surges and component chatter.

F. Temperature - The temperature of the fluid in the reservoir will be controlled to TBD plus or minus 5°F by means of heat rejection via a water cooled heat exchanger with a bypass control valve on the hydraulic fluid side. The water cooled heat exchanger will be sized to limit the temperature of the fluid to 60° above the ambient temperature. In the case of cold weather operation electric heaters controlled by suitably located thermostats will be provided in the reservoir to maintain the temperature above the minimum allowable value established by the low temperature viscosity characteristics of the fluid. The heaters will be sized so that a thermostat failure will not result in fluid coking.

G. Abnormal Operating Events - In the event of any abnormal operating condition such as loss of ac power, the hydraulic servo supply system will shut down to a condition that will permit normal startup following resumption of power without special conditioning o or readjustment of the system.

VI. AVAILABLE SERVICES

A. Working Fluid Storage - The hydraulic servo supply system is a completely self contained system with its own reservoir of fluid. Storage of hydraulic fluid will be in drums located in the facility. There will be no permanent interface with this storage for automatic fluid replenishment.

B. Electrical Power - Electrical power for the hydraulic pumps and heaters will be available from the 60 Hz 460 volt three phase power bus system. Single phase 115 volt 60 Hz power also is available for the solenoid valves and sensors.

C. Fire Protection - The installation fire protection system will accommodate sensors signals indicating a fire condition in the area of the hydraulic servo supply system. Suitable alarms will be provided and release of a halon fire suppression agent will be triggered in the event of a fire. The fire protection system will provide a signal which will shut down the hydraulic supply system when a fire is indicated.

D. Drains - No provision will be made for site oil drains. Portable drain tanks with associated valves and pumps will be used to drain the fluid from the system or components for ultimate fluid disposal off-site in accordance with local regulations. Similarly, in the event of a cleaning operation, any flushing effluent will be drained into a portable system for off site disposal.

VII. CONTROLS AND INSTRUMENTATION

A. Automatic Controls - The startup and shutdown sequences associated with the hydraulic servo supply system will be controlled by the gas turbine controller (GTC) which will also accept pressure, temperature and liquid level signals and respond with suitable alarms. The automatic control of temperature and pressure within the hydraulic system will be self contained and not require the intervention of the powerplant controller except to initiate startup and shutdown. The hydraulic servo supply system will be interfaced with the fire protection system to

produce alarm indications of fire and to respond to the instruction from the fire protection system to shutdown. By self contained controls the system will start the backup hydraulic supply pump in the event of abnormally low fluid pressure.

B. Manual Controls - The hydraulic servo supply system will be capable of being started and shutdown by manual means locally at the hydraulic system assembly.

Fluid addition to maintain proper reservoir level will be done by manual means. Control of temperature and pressure will have manual overrides available and set-points for the controls will be manually adjustable. Draining of hydraulic fluid from the system will be accomplished by use of manually operated valves. Since both local and remote triggering of the fire protection system are provided as part of that system no special provision for this is needed in the hydraulic servo supply system.

C. Remote Control - Remote manual means for startup and shutdown of the hydraulic servo supply system will be provided either in a manual or automatic mode. Adjustment of control settings for temperatures and pressures will not be available at the remote location. However, manual override of alarms and shutdown signals will be provided at the plant control room.

D. Instrumentation For Condition Monitoring - Local gages will be provided to indicate reservoir tank pressure, pump outlet pressure, filter pressure drop and oil level in the reservoir. Sensors with electrical output signals will be provided for tank outlet temperature, cooler inlet and outlet temperatures, filter pressure drop, pump outlet pressure, and regulated pressure. The signals from these sensors will be sent to the UTC controller for use in alarm and shutdown functions and to the on-line condition monitoring and diagnostic system. In the case of signals which can cause powerplant shutdown such as loss of pump pressure or oil temperature multiple sensors will be employed where possible using signals voting requiring agreement of at least two signals or the use of 3 sensors with median select provision to initiate shutdown. The alarm and shutdown trip levels are all TBD subject to system design analysis and will have provision for override by the plant operator.

VIII. ENVIRONMENTAL CONDITIONS

A. Normal Environment

Location - The hydraulic servo system will be installed in an unheated enclosure.

Temperature - The environmental temperature for normal operation will range from TBD to TBD °F.

Pressure - Normal operation will be provided under environmental pressures associated with altitudes from sea level to 7,000 feet.

Humidity - Humidity levels ranging from TBD to TBD will be accommodated.

Externally Induced Vibration - The engine mounted components and connecting plumbing will be designed so as to accept the vibration levels resulting from engine operation.

B. SYSTEM IMPACT ON LOCAL ENVIRONMENT

System Leakage - Since severe fluid leakage can result in powerplant shutdown and adversely effect availability, the Hydraulic Servo Supply system will be designed with the goal of being leak-free. The number of plumbing joints will be kept to a minimum and maximum use will be made of welded connections. In the case of pipe threaded connections, adequate space will be provided to assure proper tightening during assembly and all piping will be securely fastened to minimize the impact of engine vibration and stresses arising from shipping loads. Fittings will have positive locking provisions. The plumbing design will incorporate provisions for thermal expansion to minimize resulting stresses on plumbing fittings and component housings. Where possible connections to components will be by means of straight threaded fittings using "O" ring seals. All fittings and seals will be selected to withstand hydraulic pressure spikes of at least 300% of normal pressure without losing system seal.

System Noise - Pump and blower speeds and generated noise will be within acceptable engineering practice and OSHA regulations.

System Induced Vibration - The system pumps will produce vibrations which can be transmitted to structures and to associated plumbing. A design analysis will be performed to assess the impact of the generated vibrations and to establish the need for isolation to preclude the possibility of stresses to components and plumbing in this and any other installed system.

C. Abnormal Environment For Equipment

Temperature Extremes - The system will be designed to withstand environmental temperatures ranging from -65°F to 130°F without damage in a non-operating mode.

Earthquakes - The system will be designed to withstand seismic loads of 0.33G (2.1G max) horizontal (ground motion) on structure no damping. Similarly, it will withstand 0.22G (1.41G max) vertical (ground motion) on structure no damping. Interconnecting piping will be designed in accordance with local requirements and building foundation design criteria.

Fire - In the event of indication of fire by the Fire Protection System, provision will be made for isolation of the hydraulic fluid supply by activation of suitably located solenoid valves. The desire is to minimize the probability of the fluid feeding any fire in the area.

Fire Suppression - The design of the system and selection of materials employed will be such that no damage to any equipment, wiring, plumbing or connections will result from exposure to the fire suppression agent.

Conditions During Maintenance - The system will be designed to be protected from accidental damage by maintenance procedures applied to this or any adjacent system. Workmen stepping on critical components, possible damage due to dropped tools, or swinging crane hooks must be taken into account in the design of the system.

Lightning - Provisions must be made for proper system grounding to minimize the impact of lightning strikes.

Dust And Dirt - All containers will be closed except for necessary vents which will be protected by screens and filters from dust and dirt.

Vermin And Insects - All system vents and breathers communicating with the system exterior will be protected with screens. All cables and tubing will be armor protected against vermin.

Liquid Leakage From External Sources - The system will be designed and materials selected such that no system damage or impairment to system life will result from exposure to any fluids employed in the powerplant installation.

IX. STRUCTURAL/PACKAGING REQUIREMENTS

A. Packaging Objectives - The system will be designed as a packaged assembly with a suitable structural base plate. It will be capable of being factory assembled, shipped as a unit and installed with a minimum number of connections. The pumps will be mounted external to and close to the reservoir tank and at a level such that the pump inlets are gravity fed from the reservoir. If components are mounted on the tank top, it will be rigid enough to preclude sagging and propagation of vibrations. Pads will be used for mountings to prevent any hole penetrations into the tank. The structural base plate will have a drip rim with suitable drain connections to allow collection of fluid leakage or spill from a fill operation.

B. Factors Of Safety/Codes - Normal engineering practice will be employed in sizing components, plumbing and wiring. Where design analysis shows component selections to be marginal for the service, the next larger component suitable for the service will be employed. Where analysis shows that life or reliability can be enhanced by the use of devices having greater performance or capacity margins than normal practice would dictate, the greater margins will be employed. The design of all plumbing will be consistent with ANSI B31.1.

C. Handling/Shipping Loads - The system will be designed to withstand handling loads of up to 3 G's without damage. The system also will be capable of withstanding the maximum loads imposed as a result of shipment by truck or by rail transportation.

D. Interface Requirements And Loads - All interfaces with the powerplant and the facility will be designed to hold stresses to components, plumbing and cabling well within safe limits as a result of thermal expansion, and normal and abnormal environmental conditions.

E. Provisions For Lifting/Mounting - An overhead crane and fork lift truck will be available for lifting and moving the system and its subassemblies. All parts weighing 50 pounds or more will have provisions for lifting and trunnioning. Eye bolts will be provided for removing covers and permanent lifting lugs suitable for slings will be provided on sub-modules where possible.

F. Servicing Considerations - The system will be designed to be serviced by simple procedures and clearance around components will be such as to make them easily accessible to allow repair or replacement while the system is operating. All adjustments will be capable of being made using standard industrial tools. All electrical interfaces will be wired to a central easily accessible terminal junction box.

G. Cleaning And Flushing Criteria - The cleanliness of the fluid delivered by the system will be assured by the use of suitable filters in a clean system. The components and plumbing must be flushed repeatedly and new filters installed prior to the initial startup. In the event subsequent severe contamination of the oil occurs for any reason as indicated by clogging of filters in short periods of operation, it will be necessary to flush clean the system fluid loop. Appropriate valving will be provided to permit such flushing and collection of the effluent from this operation. To facilitate a cleaning operation, inspection plates and cleanout ports will be provided in the reservoir.

H. Size Limits - The system will be designed to be truck transportable.

I. Weight Limits - The system weight will be consistent with limits permitted by truck transportation.

X. RELIABILITY AND MAINTAINABILITY CRITERIA AND GOALS

A. Reliability Goals - All components will be selected for maximum reliability. Redundancy will be employed in the system design where it can be shown to improve system reliability. An MTBF of TBD will be used as a goal in the design of this system.

B. Availability Goals - Since the Hydraulic Servo Supply System is critical to the operation of the powerplant, special attention is required in its design in order to avoid down time for any reason. The use of redundant components and

installed spares and provisions for servicing while operating will be necessary to maximize the hydraulic servo supply system availability which will be TBD as a goal.

C. Maintenance Philosophy - To the maximum extent consistent with personnel safety, the Hydraulic Servo Supply System will be designed to be serviced by simple procedures while the powerplant is operating. Requirements for special tools and fixtures are to be avoided. The system can have routine "walk-around" visual inspection while the plant is operating. Maintenance which requires shutting down the system will be performed only on an as-required basis.

D. Component Access And Replacement Times - Use of modular construction will be emphasized to minimize down time for service and replacement of components. The design of the module interfaces will be such as to facilitate access to connections and allow replacement of components by use of simple procedures. Down time for repair should not exceed one working shift as a goal. All cabling, external accessories, and plumbing should be capable of being checked for security of mounting and evidence of leakage or chafing. Filters will be easily accessible for replacement without loss of operating time.

E. Scheduled Down Time Criteria - There will be no scheduled periods of down time for routine service other than that specified for the engine which is 50 hours every 9,000 hours and 100 hours every 18,000 hours.

F. Spare Parts Philosophy And Guidelines - In the selection of components, preference will be given to equipment which is readily available. If long lead time parts must be employed and system availability could be jeopardized, consideration will be given to the use of installed spares. Otherwise, a policy of stocking spare long lead time components will be followed.

G. Design Life - In general the system should be designed for a 30 year life; however, as a minimum, the system will be capable of 9,000 hours of operation without major maintenance.

H. Restrictions On Cascading Failures - The system will be designed to guard against the possibility of cascading failures initiated by the malfunction of any system component.

XI. SPECIFIC DESIGN GUIDELINES

A. Codes To Be Observed - The system plumbing will be designed to be consistent with ANSI b31.1. The vessels will conform to the ASME code for unfired pressure vessels, Section VIII and all electrical design will be consistent with NFPA-70. All electrical equipment will be explosion proof rated for Class I, Division 2, Groups B and D conditions per NFPA-70. The system design will be in general agreement with API Standard 614. OSHA regulations will apply to the entire system design.

B. Materials To Be Banned - Cast iron or copper will not be used in any fluid wetted portions of the system. The materials will be selected to be compatible with all fluids with which they come in contact. Teflon pipe thread tape will be prohibited. All materials and processes used will be of high industrial quality and will not contribute to personnel or environmental hazard.

C. Hardware Types Not Suitable - Electrical equipment not consistent with Class I, Group D, Division 1, Groups B and D of the NEC is prohibited. Multi-set point thermal switches will not be used in any control function. All piping will be of seamless stainless steel.

D. Safety Provision - The system will be consistent with NFPA No. 30, NFPA No. 70 and OSHA regulations.

E. Other Guidelines - Maximum use will be made of standard off-the-shelf fasteners, flanges, connections and seals. Self-locking features will be employed on connectors and fasteners to the maximum extent possible. The hydraulic fluid will be selected for maximum component life potential and be fire resistant if possible consistent with the low temperature viscosity requirements. The reservoir will be a baffled tank to allow settling of contaminants and provide isolation between the pump suction and the high velocity fluid return flow. The reservoir to pump interface will be such as to assure laminar flow conditions at the pump inlet. The pump will prime immediately on starting.

RELIABILITY PROGRAM PLAN

Contract RP1187-1

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INTRODUCTION

The Reliability Program Plan establishesd procedures and lines of organizational responsibility which will ensure that reliability becomes an integral part of product development. Formulation of and adherence to a Reliability Program Plan has become standard practice in the Propulsion Group of United Technologies whenever there is a new gas turbine program. The organizational structure within the Propulsion Group provides direct staff responsibility for product integrity at all levels. This serves to provide a common gas turbine reliability discipline which can be shared among the divisions. Thus, Power Systems Division's organizational structure and procedures parallel those of other UTC divisions.

The Reliability Program Plan associated with a particular engine project takes the pre-existing organization and standard methods for product integrity and tailors them to that project. The material provided in this document pertaining to the High Reliability Gas Turbine development program was formulated in that manner. Normally an initial plan as a separate document is prepared early in the program, then expanded, modified and refined throughout the course of the program to keep current with the development status as it evolves. As such, the document can be a flexible, working guide. By developing a reliability plan at this early stage of the HRGT program, it can maintain that flexibility as the program evolves through subsequent phases.

The complete Reliability Plan usually includes three basic elements:

- Assignment of organizational responsibility
- Standardized control procedures during design and procurement phases
- Test development plan

The material covered in this document, which was developed as part of the Phase I conceptual design of the EPRI High Reliability Gas Turbine Program, Includes the first two elements. Details of the third element, Test Development Plan, will be worked out at a later date as the program nears its hardware development phase. At that time the product integrity testing can be made an integral part of overall Test Development Plan. That plan will be established to best suit the total program requirements in general, the reliability goals in particular, and the available resources to meet those requirements. The Reliability Program Plan will be updated accordingly.

A. ADMINISTRATION

1. Organization

The PSD organization as shown in Figure 9-1 has been established with sufficient authority to ensure the successful implementation of the reliability program plan. Both the Quality Assurance Manager and the Product Assurance group report through Administration, independent of other departments.

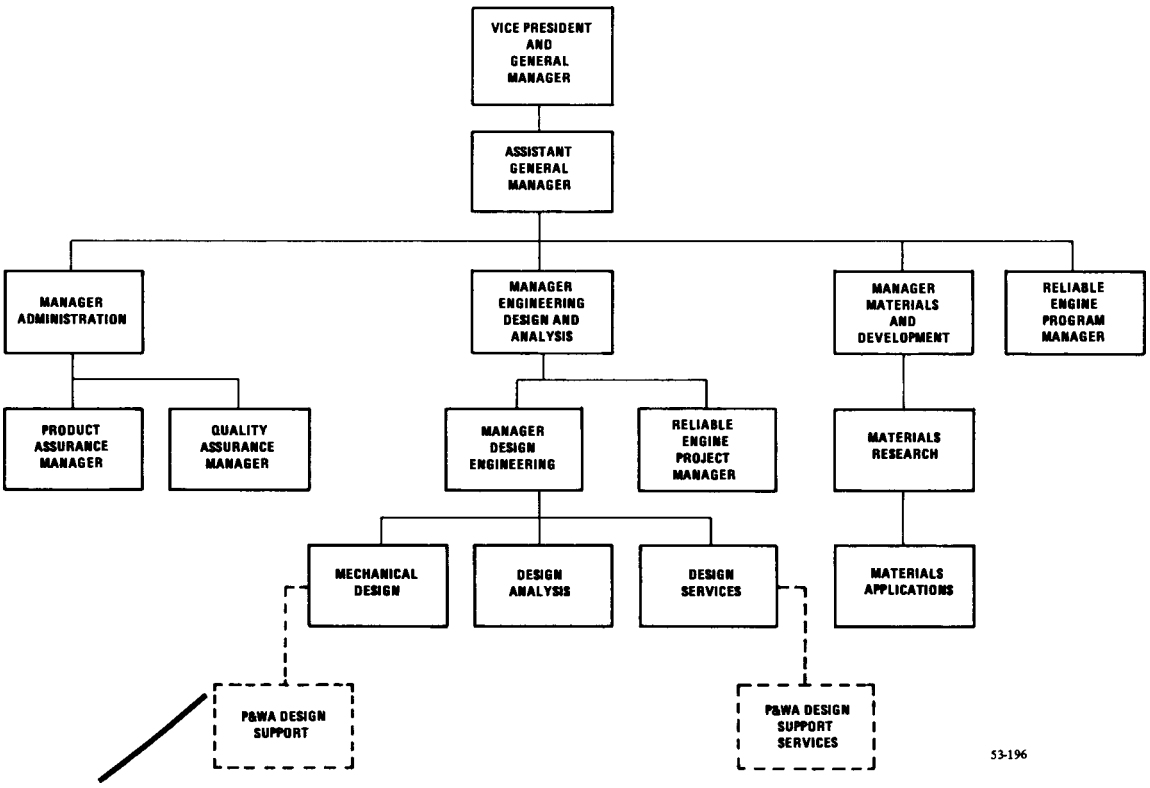

Figure 9-1. PSD Organization

The functional responsibilities and activities of the Quality Assurance organization is shown in Figure 9-2. The quality assurance system beginning with the customer's specification and follow through, to the delivery of material to the customer is outlined schematically in Figure 9-3.

Product Integrity responsibilities at PSD provide assurance that the powerplant is designed properly to meet application requirement and manufactured to meet those design requirements. The Quality Assurance and Product Assurance sections are responsible for product integrity.

The Product Assurance section conducts design reviews and does Reliability Analysis, Failure Mode and Effect Analysis (FMEA), Fault Tree Analysis, Safety and Hazards Analysis and Failure Analysis. They recommend and negotiate design and operational changes with Project Management and Design. They are afforded a direct route to management on items of concern which do not engender reasonable response.

The Quality Assurance section provides Quality Engineering design support, quality methods analysis, supplier quality control, receiving, assembly and test inspection, materials control, destructive and non-destructive testing, and gage standards and calibration.

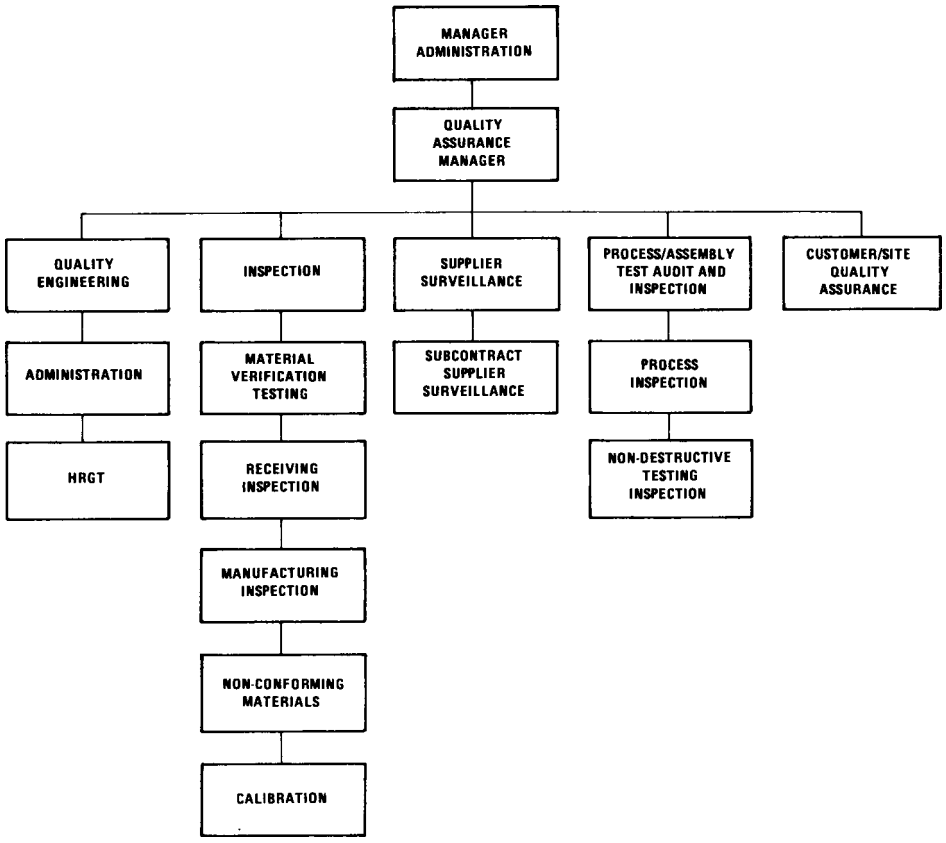

Figure 9-2. Quality Assurance Organization

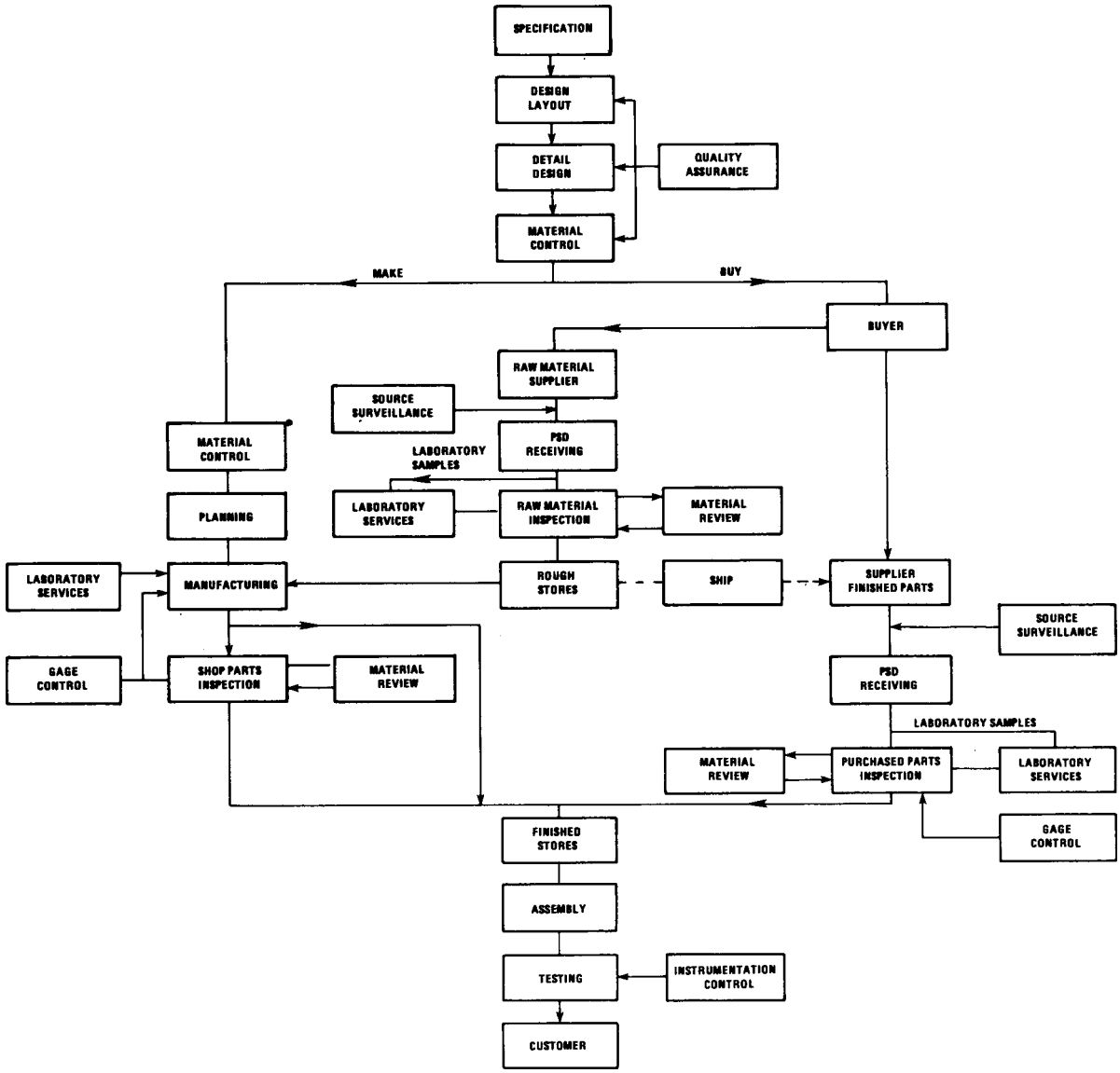

Figure 9-3. Quality Assurance System

2. Applicable Documents

The following documents are considered applicable to the High Reliability Gas Turbine program.

a. United Technologies Corporation

- PSD Standard Procedures Manual

- PSD Quality Assurance Procedure
- PSD Product Assurance Procedure
- P&WA Standard Procedures Manual
- P&WA Design Manual
- P&WA Drafting Room Manual

b. ANSI Standards

N45.2.9	Requirements for Collection, Storage, and Maintenance of Quality Assurance Records
N45.2.10	Quality Assurance Definition

B. DESIGN CONTROL

1. Product Assurance Engineering

Product Assurance Engineering assists in the establishment of the following requirements during all stages of a products life cycle from conceptual design through development and production.

- Reliability and life characteristics
- Maintainability
- Availability
- Codes and Standards
- System safety
- Statistical analysis

Specific responsibilities to meet these requirements include:

a) Participation with Engineering in the interpretation of customer's requirements for all Product Assurance functions.

b) Preparation of reliability, maintainability, safety, and code requirements for incorporation into the Component Design Requirements (CDR) and specification.

c) Conduct reliability/redundancy trade studies, system Failure Mode and Effect Analysis (FMEA) to identify critical failure, modes and the magnitude of inherent hazards. Results of these analysis are used as inputs for design reviews.

d) Conduct maintainabilty study prior to completion of packaging design.

e) Participation in the testing program as required to provide input for designed experiments, reliability and life testing and safety incidents.

2. Quality Engineering

Quality Engineering assists in the establishment of quality requirements during the design process and with the implementation of those requirements during the procurement, manufacturing, assembly and test processes. Specific responsibilities include:

a) Design reviews to ensure quality requirements are established uniformly throughout the design. Quality Assurance Data (QAD) sheets are used to document these requirements. They include establishment of pressure, leak and non-destructive testing requirements, parts inspectablity, and considerations to ensure design requirements are met.

b) Participation in Engineering change review and approval to establish quality requirements, assess impact on existing inspection procedures, and determine ability to incorporate changes to inspection procedures consistent with schedule requirements.

c) Preparation of quality specifications as required to establish system quality requirements.

d) Preparation of Quality Methods and Standards as required to interpret Engineering Drawing and Specification requirements.

e) Approval of Engineering Specification to ensure requirements can be verified.

f) Participation in source selection activity including membership on source selection committee to ensure suppliers have a quality system capable of satisfying design requirements.

g) Establishment of destructive and non-destructive testing techniques as required to verify material integrity in conjunction with Design, Project and Materials activity.

h) Reviews of drawings and specifications with representatives of Design, Project, and Manufacturing to establish levels of receiving manufacturing and source inspection activity as documented in Master Inspection Plans.

i) Participation in problem investigation and resolution to enable corrective action to be taken which will prevent a recurrence of the problem.

j) Performance of an audit of Engineering activity to ensure program requirements are being satisfied.

Consideration will be given to end item functional and assembly requirements when establishing quality requirements.

Quality Engineering also supports the inspection activity to ensure design and drawing requirements are properly verified. Specific responsibilities include:

a) Establishment of inspection techniques as required to verify drawing requirements when requested by the Inspection Department.

b) Design, development and implementation of inspection tooling and fixturing required to support inspection techniques.

c) Review of manufacturing operation sheets and job orders to establish in-process or final inspection requirements for new material or material being reopened.

3. Design Engineering

Design Engineering is responsible for developing the design and producing the drawings necessary to enable manufacture of products that will meet the contractual requirements. Specific responsibilities include:

a) Preparation of Design Table that defines the system performance characteristics.

b) Participation in the preparation of Component Design Requirements (CDR) which summarize the design requirements for each particular component.

c) Substantiation of the mechanical design by detailed analyses in the area of structures, fluid flow, heat transfer, etc. All work directly in support of formally released drawings and specifications will be documented in a design calculation notebook.

d) Initiation and control of changes to all released design documents to ensure that all changes are properly approved and incorporated into the appropriate documents (drawings, specifications, parts list, etc).

e) Preparation and processing of release approval packages which contain the Engineering Release Form and the prints of new and/or revised parts and all attendant documents.

f) Distribution of all approved drawings and their documents to both PSD's engineering department and to the required customer contacts.

4. Documentation

Documentation control ensures that results of inspection and testing are controlled for reference and investigation purposes.

C. PROCUREMENT CONTROL

Supplier Surveillance

Supplier surveillance activities ensure that items purchased from suppliers are being produced in accordance with design and specification requirements. All suppliers will be approved by Quality Assurance prior to placement of orders. Specific responsibilities include:

a) Review of suppliers QA systems prior to initiating source surveillance activity.

b) Identify nonconforming material.

c) Control processing and disposition of acceptable and nonconforming material.

d) Control documentation associated with manufacturing inspection and inspection activities.

D. PROCESS CONTROL - NONDESTRUCTIVE TESTING

Process Control

Process Control ensures that processing requirements as defined in process specifications have been satisfied. Processes are controlled by fixed and random audits of manufacturing process procedures. Nondestructive testing as specified on the Quality Assurance Data (QAD) sheet is performed to ensure that the product is free of defects and meets all product integrity requirements.

E. ASSEMBLY - TEST INSPECTION

Assembly and Test

Assembly and Test QA activity ensures design requirements are satisfied during powerplant assembly and test operations. Specific responsibilities include:

a) Review of assembly and test specifications and operation sheets to establish the level and sequence of inspection required.

b) Qualification of operation sequences and personnel performing specific assembly and test operations.

c) Inspection of assembly and test operations.

F. NONCONFORMANCES

Nonconforming Material

Material which has been identified as nonconforming to drawing or specification requirements must be controlled and evaluated to determine acceptability for power plant use. Specific controls and evaluations include:

a) Documentation of nonconformances.

b) Investigation of nonconformances in conjunction with Engineering personnel to determine acceptability.

c) Release of material following documented acceptance or rejection of nonconformances.

G. METROLOGY - CALIBRATION

Metrology

All dimensional, pressure, and electrical measuring equipment used for manufacturing inspection, and testing is calibrated with the calibration standards traceable to the National Bureau of Standards. Quality Assurance, Gage Standards Department, is responsible for calibration of dimensional measuring equipment. Pressure and electrical measuring equipment calibration is the responsibility of the Electronic and Pressure Test Laboratories. A Metrology Control Group comprised of representatives of the various user and calibration groups is responsible for establishing all calibration policies.

H. DOCUMENT CONTROL

Document Control

The control of the documents affecting quality is audited by Quality Assurance. This activity ensures that new/or revised documents are reviewed for adequacy and approved for release by authorized personnel. Also, their distribution is controlled to ensure that the documents are available to and used at the location where the prescribed activities are reviewed and approved by the same groups that performed the original review and approval.

Documentary evidence that processes, materials, and equipment conforms to the specified requirements is maintained and traceable. This evidence is made up of design calculation notebooks, design review records; procurement records; inspection and tests records; qualification of personnel; procedures and equipment records; and nonconformances and corrective actions records.

The transmittal of documents to the customer will be controlled using the Document Transmittal Form. This form presents a complete discription of the document and provides a confirmation of receipt.

I. PACKING - SHIPPING

Packing and Shipping

Packing and shipping activities ensure that material has been properly protected to prevent damage from corrosion, deterioration, and physical damage during handling and shipping.

J. FORMS

1. Engineering Calculation Sheet

   Documents the design criteria and calculations.

2. Design Change Request

   Initiates design change activities.

3. Drafting Job Requests - Engineering

Initiates drafting activities.

4. Design Information

Documents results of design reviews.

5. Quality Assurance Data (QAD) Sheet

Defines quality and inspection requirements.

6. Engineering Release (2 Pages)

Defines task function to be accomplished for new/or reviewed configuration.

7. Engineering Part Lists and Drawing Worksheet

Documents configuration of parts, assembly and deliverable hardware.

8. Document Transmittal

Controls the issuance of documents.

9. Failure Mode and Effect Analysis (FMEA)

Identifies critical failure modes.

10. Inspection Material and Test Record (IMTR)

Documents results of inspection for all shipments.

11. Master Inspection Plan (MIP)

Documents inspection requirements.

12. Inspection Plan and Status Record (IPSR)

Documents results of inspection on individual shipments.

13. Receiving Report (RR)

Identifies shipment and documents nonconformances.

14. Inspection Job Order

Defines unique inspection and gage calibration requirements.

15. Job Order

Identifies job requirements.

16. X-Ray Job Order

Define X-ray inspection requirements and results of testing.

17. Nonconformance Report

Documents nonconforming material disposition.

18. Operation Sheet

Defines manufacturing operations.

19. Assembly Sheet

Defines assembly operations.

UNITED TECHNOLOGIES
POWER SYSTEMS DIVISION

ENGINEERING CALCULATION SHEET

BY		TITLE	YEAR	SERIES	JOB NUMBER
DATE					
MODEL			PAGE		

FCO FORM NO. 0216 8/02/77

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1. Engineering Calculation Sheet

UNITED TECHNOLOGIES
POWER SYSTEMS DIVISION

DESIGN CHANGE REQUEST

PROPOSED BY	DEPT.	PHONE	LOCATION	YEAR	SERIES	JOB NO.
WOS	POWERPLANT(S) AFFECTED					

PROBLEM

ACTION DESIRED

REASON

PARTS AFFECTED

INCORP. REQUIREMENTS & DISPOSITION OF MATERIAL. (THIS CHANGE APPLIES TO POWERPLANTS & SPARES UNLESS OTHERWISE NOTED BELOW) (INDICATE APPLICABLE INCORPORATION REQUIREMENTS WITH "X")

☐ THIS CHANGE MUST BE INCORPORATED IN ALL _____ POWERPLANTS

☐ THIS CHANGE MUST BE INCORPORATED IN POWERPLANT SER. NO. _____ AND SUBSEQUENT POWERPLANTS

☐ USE PARTS AND MATERIAL ON HAND AND IN PROCESS

☐ PARTS MADE/IDENTIFIED IN ACCORDANCE WITH THIS CHANGE MAY BE MIXED WITH THOSE OF THE PRIOR CONFIGURATION/IDENTIFICATION

☐ PARTS, MATERIAL AND TOOLS ARE NOT AFFECTED (PER

(✓) ITEMS AFFECTED	YES	NO		YES			YES	NO
CANCEL PART FROM DELIVERY			ASSY PARTS LIST			TOOLING		
OBSOLETION OF PART REQ'D			ASSEMBLY DRAWING			INTERCHANGEABILITY		
INSTALLATION			INSTALLATION DRAWING			FUEL CELL COMPONENT SPEC		
RETROFIT RECOMMENDED			MODEL SPEC. AFFECTED			RED FLAG NOTICE REQ'D		

TEST DATA – LIST ALL RUNS

☐ CONSIDERED UNNECESSARY

☐ WILL FOLLOW

ENDORSEMENTS (SIGNATURE & DATE)	ENGINEERING RECORDS	PROJ ENGINEER	MANUFACTURING	PURCHASING	PRODUCT ASSURANCE
	MATERIALS		FINANCE	QUALITY ASSURANCE	MATERIAL DEPT
	ASST PROJ ENG		PRODUCT SUPPORT	PROG /PROJ MGR	

REMARKS

FCO FORM NO. 0220 REV 3/16/78

2. Design Change Request

DRAFTING JOB REQUEST — ENGINEERING

TO _____

YEAR	SERIES	JOB NO.

ENG. RELEASE NO. _____ SUPP _____

RELEASE NO. _____ SUPP _____

PRIORITY		TARGET DATE	REPORT PAGE & DATE
DESIGN	DRAFTING		

REFERENCE MEMO & DATE OR WOS _____

LAYOUT TITLE OR REF. P/N & TITLE _____

_____ DCB NO. _____

FIELD SURVEY NO. _____ LAYOUT & SH. NO. _____ CHG. _____

DETAIL ☐ REVISE ☐ TIME CHARGE NO. _____

RELEASE TO:

☐ RECORD - ACTIVE 2 YRS. ☐ EXP.
☐ RECORD - ACTIVE 6 MOS. ☐ DELIVERY
☐ RECORD - OBSOLETE

MODELS AFFECTED { _____ _____ _____

NATURE _____

REASON _____

INCORPORATION:

☐ MUST CONFORM
☐ DATED
☐ UNDATED
☐ OTHER _____

MATL. DISPOSITION: *(Indicate P/N & special instructions if necessary)*

☐ DIVERT _____
☐ REOPERATE _____
☐ SCRAP _____
☐ USE _____
PARTS MATL. & TOOL NOT AFFECTED PER: _____ DATE: _____
OTHER _____

ERP JUSTIFICATION:

☐ RECORD ☐ PROD. STOP
☐ INTERFACE ☐ COST RED
☐ COMPATIBILITY ☐ SAFETY
☐ CORRECTION ☐ VALUE ENGR.
☐ OPNL/LOG

YES	NO	
☐	☐	PERFORMANCE AFFECTED
☐	☐	INTERCHANGEABILITY AFFECTED
☐	☐	INSTALLATION (INTERFACE OR CG) AFFECTED
☐	☐	INSTALLATION DRAWING AFFECTED
☐	☐	ALTER SIMILAR PARTS
☐	☐	NEW PART NO.

SIGNATURES

DESIGN ENGR.	DATE	PHONE		DESIGN PROJ. ENGR.	DATE
MATERIAL DEPT.	DATE	MATERIALS*	DATE	DETAILER	DATE
PROJECT ENGR.	DATE	Q.A.E.*	DATE	DESIGN ANALYST	DATE
CHIEF DESIGN ENGR.	DATE	SOURCE APP.*	DATE	CHIEF DFTSMN.	DATE

* REQUIRED ONLY IF NO LAYOUT

SEE REVERSE SIDE FOR ADDITIONAL INFORMATION — YES ☐ NO ☐

FCO-0087 Rev. 6/13/77

3. Drafting Job Requests - Engineering

DESIGN INFORMATION

ENGINEER	DATE	ENGINE AREA CODE	ENGINEERING FILE NO. YEAR / SERIES / JOB NO.
GROUP			

ENGINE MODELS AFFECTED	EO NO.	SUPPL	ENGRG CHG NO.	SUPPL

LAYOUT NO.	SHEET	CHG	LOC	COMPL DATE MO	YR	ENGINEER / AUTHOR	SECURITY CLASS.	DCB NO.	VOL

DSM
☐ YES ☐ NO

OTHER DOCUMENT TYPE & NO.

JOB PURPOSE
☐ STUDY
☐ TEST RIG
☐ TEST FACILITY
☐ DEMONSTRATOR
☐ ENGINE

ENGINE ITEMS AFFECTED

DESCRIPTIVE JOB TITLE (CIRCLE APPROPRIATE SUBJECTS)

DESIGN APPROVAL

RELIABILITY FILE NO. APPROVED	STRUCTURES FILE NO. APPROVED	PRODUCT COST ANALYSIS FILE NO. APPROVED
MAINTAINABILITY FILE NO. APPROVED	VIBRATION FILE NO. APPROVED	MANUFACTURING FILE NO. APPROVED
VALUE ENGINEERING FILE NO. APPROVED	HEAT TRANSFER FILE NO. APPROVED	FILE NO. APPROVED
WEIGHT ENGINEERING FILE NO. APPROVED	MECHANICAL COMPONENTS FILE NO. APPROVED	FILE NO. APPROVED
GEAR DESIGN FILE NO. APPROVED	AERO / THERMO FILE NO. APPROVED	FILE NO. APPROVED

REMARKS

PWA FORM 3989 (FRONT) REV. 9-76

4. Design Information (2 PAGES)

SUBJECTS APPLICABLE (Check Appropriate Blocks)

- [ ] AIRFRAME PROVISION
- [ ] ASSEMBLY METHODS
- [ ] COMMONALITY STUDY
- [ ] COMPUTER PROGRAMS
- [ ] CONFIGURATION STUDY
- [ ] CONFORMANCE ANALYSIS
- [ ] CONTRACT REQUIREMENTS
- [ ] COST ANALYSIS
- [ ] COST COMPARISON
- [ ] COST ESTIMATES
- [ ] COST REDUCTION
- [ ] CRITICAL SPEED ANALYSIS
- [ ] CUSTOMER LIAISON
- [ ] DAMAGE ANALYSIS
- [ ] DAMAGE SURVEY
- [ ] DATA ANALYSIS
- [ ] DESIGN
- [ ] DESIGN APPROVAL
- [ ] DESIGN COMPARISON
- [ ] DESIGN CRITERIA
- [ ] DESIGN CURVES
- [ ] DESIGN DATA
- [ ] DESIGN ESTIMATES
- [ ] DESIGN MARGIN
- [ ] DESIGN METHODS
- [ ] DESIGN PHILOSOPHY
- [ ] DESIGN REPORT
- [ ] DESIGN REVIEW
- [ ] DESIGN STUDY
- [ ] DESIGN SUBSTANTIATION
- [ ] DESIGN SUPPORT
- [ ] DESIGN TABLES
- [ ] DEVELOPMENT DATA REPORT
- [ ] DURABILITY IMPROVEMENT
- [ ] EMISSIONS REDUCTION
- [ ] EMPLOYEE SUGGESTION IMPROVEMENT
- [ ] ENGINE WEIGHT STATUS
- [ ] ENGINEERING CHANGE PROPOSAL
- [ ] FAULT TREE ANALYSIS
- [ ] FEASIBILITY STUDY
- [ ] FIELD EXPERIENCE ANALYSIS
- [ ] FLUID FLOW ANALYSIS
- [ ] FMEA INPUT
- [ ] HAZARD ANALYSIS
- [ ] HEAT TRANSFER
- [ ] INITIAL DESIGN
- [ ] INSPECTION METHODS
- [ ] INSTALLATION
- [ ] INSTALLATION WEIGHT STUDY
- [ ] LABORATORY QUALITY REVIEW ORDERS
- [ ] LAYOUT WEIGHT ESTIMATES
- [ ] LOW CYCLE FATIGUE
- [ ] MAINTAINABILITY
- [ ] MAINTENANCE COST
- [ ] MAINTENANCE TASK LISTS
- [ ] MANUFACTURING SUPPORT
- [ ] MASS PROPERTIES
- [ ] MATERIALS STUDY
- [ ] MATH MODELS
- [ ] MOCK-UPS
- [ ] NOISE REDUCTION
- [ ] OPERATIONS PLANNING
- [ ] PARAMETRIC STUDY
- [ ] PERFORMANCE IMPROVEMENT
- [ ] PRELIMINARY DESIGN
- [ ] PROBLEM STUDY
- [ ] PROJECT SUPPORT
- [ ] PROPOSAL SUPPORT
- [ ] QUALITY REVIEW
- [ ] REDESIGN
- [ ] SAFETY
- [ ] SALES SUPPORT
- [ ] SERVICE DATA REPORT
- [ ] SERVICE FIX
- [ ] SERVICE SUPPORT
- [ ] SERVICE TESTS
- [ ] SPECIFICATION WEIGHT ESTIMATES
- [ ] STRUCTURAL ANALYSIS
- [ ] SURVIVABILITY
- [ ] TARGET WEIGHT
- [ ] TDM WEIGHT ESTIMATES
- [ ] TEST PROGRAM
- [ ] TOLERANCES
- [ ] TRADE STUDY
- [ ] TRAINING PROGRAM
- [ ] TRIP REPORT
- [ ] UPRATED ENGINE ESTIMATES
- [ ] VIBRATION ANALYSIS
- [ ] VULNERABILITY
- [ ] WEAR LIMITS
- [ ] WEIGHED WEIGHT ANALYSIS
- [ ] WEIGHT COMPARISON
- [ ] WEIGHT REDUCTION
- [ ] WEIGHT SCALING STUDY

OTHER APPROPRIATE SUBJECTS

ABSTRACT

PWA FORM 3989 (PACK) REV. 9-76

QUALITY ASSURANCE DATA – (WORKSHEET)

PRATT & WHITNEY AIRCRAFT GROUP

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REF. INFO. (Not Keypunched)	04	QAD 59	REV. 77
	07	REVISION DATE 54	CODE IDENT. NO. 77445
KEY: # - CARD NO. # - COLUMN NO	11	FURNISHED UNDER CONTRACT 54	

13	PART NAME 4			
	4 REQ'T	16 METHODS	39 STANDARDS	54 INSPECT
17				
18				
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33				

34	PREPARED BY	NAME 16	DATE 32	DSGN. ENGRG.	NAME 52	DATE 72
35	QUAL. ENGRG.	NAME 16	DATE 32	PROJ. ENGRG.	NAME 52	DATE 72
36	MATLS. ENGRG.	NAME 16	DATE 32			

38	ADDITIONAL REQUIREMENTS: INFORMATION FOR PWA INSPECTION CONTROL	QAD CONTROL NO. 77	1
39	4	REVISION NOTICE NO: 72	
40	4	PRIOR REVISION NOTICE NOS 62	
41	4	62	
42	4	43	4
44	4	45	4
46	4	PWA FINAL ACCEPTANCE SYMBOL MAY BE WAIVED (DS&C) 64 ☐ YES 73 ☐ NO	47 NOT USED

ADDITIONAL REQUIREMENTS FOR PURCHASED MATERIAL

48	4 KEYPUNCH FORTY X'S TO CROSS-OUT 'NDT MUST BE PERFORMED BY PWA APPROVED VENDOR'	PWA-QA 62	49	4 KEYPUNCH PWA QUALIFICATION OF VENDOR NOT (REQUIRED) (WAIVED)	50	51 NOT USED
52	CONTROL OF HIDDEN REQ'TS PER PWA-QA 6071	7 ☐ YES	13 ☐ NO		53	54 NOT USED
55	PWA TO SUPPLY SPECIAL GAGING EQUIPMENT	7 ☐ YES	13 ☐ NO	56 TYPE 26		57 NOT USED
58	PURCHASE PERFORMANCE SPECIFICATION	7 ☐ YES	13 ☐ NO	PPS 26	59	60 NOT USED

	OTHER REQUIREMENTS FOR PURCHASED MATERIAL PWA 300 APPLIES		
61	4	62	4
63	4	64	4

PWA FORM 4420A REV. 7-76

ENGINEERING SERVICES (ORIGINAL)
QUALITY ASSURANCE ENGINEERING (COPY)

5. Quality Assurance Data (QAD) Sheet

CODE _____
WORK ORDER NO. _____
WORK ORDER SUPP. NO. _____
MEMO REF. _____
JOB NO. _____
COPIES TO: LOCATION

UNITED TECHNOLOGIES
POWER SYSTEMS DIVISION
SOUTH WINDSOR, CONN.

ENGINEERING RELEASE
☐ EXPERIMENTAL
☐ DELIVERY

NO. _____ SUPP. _____
PAGE _____ OF _____
DATE ISSUED _____
MODEL(S) _____

NATURE, REASON, DISPOSITION

INCORP. REQUIREMENTS & DISPOSITION OF MATERIAL. (THIS CHANGE APPLIES TO POWERPLANTS & SPARES UNLESS OTHERWISE NOTED BELOW)

(INDICATE APPLICABLE INCORPORATION REQUIREMENTS WITH "X")

☐ THIS CHANGE MUST BE INCORPORATED IN ALL _____ POWERPLANTS.
☐ USE PARTS AND MATERIAL ON HAND AND IN PROCESS
☐ PARTS MADE/IDENTIFIED IN ACCORDANCE WITH THIS CHANGE MAY BE MIXED WITH THOSE OF THE PRIOR CONFIGURATION/IDENTIFICATION
☐ PARTS, MATERIAL AND TOOLS ARE NOT AFFECTED (PER
☐ THIS CHANGE MUST BE INCORPORATED IN POWERPLANT SER. NO. _____ AND SUBSEQUENT POWERPLANTS

LEGEND (SEE DISPOSITION OF MATERIAL COLUMN)

N_____ PARTS, MATERIAL AND TOOLS ARE NOT AFFECTED.
U_____ USE PARTS AND MATERIAL ON HAND AND IN PROCESS.
D_____ WHEN NEW PARTS ARE AVAILABLE, DIVERT SUPERSEDED PARTS AND MATERIAL NOT NEEDED FOR IMMEDIATE REQUIREMENTS, TO OTHER REQUIREMENTS OR DELIVER TO DORMANT STORES IN ANTICIPATION OF FUTURE REQUIREMENTS.
S_____ WHEN PARTS INCORPORATING THIS CHANGE ARE AVAILABLE, SCRAP ANY PARTS PRIOR TO THIS CHANGE WHICH ARE NOT NEEDED FOR IMMEDIATE REQUIREMENTS AND DIVERT MATERIAL TO OTHER REQUIREMENTS OR DELIVER TO DORMANT STORES IN ANTICIPATION OF FUTURE REQUIREMENTS.
R_____ REOPERATE PARTS PRIOR TO THIS CHANGE, ON HAND AND IN PROCESS NOT NEEDED FOR IMMEDIATE REQUIREMENTS, AND REIDENTIFY TO THEIR RESPECTIVE NEW PART NUMBERS/TO THIS CHANGE.
W_____ OTHER WRITTEN DISPOSITION OF MATERIAL.

SUPERSEDED PARTS RETENTION
LEGEND (SEE COLUMN AT RIGHT OF CANCELED PART NUMBERS)

Y_____ RETAIN FOR SPARES FOR THIS APPLICATION.
O_____ RETAIN FOR SPARES FOR OTHER APPLICATIONS ONLY.
N_____ DO NOT RETAIN FOR SPARES.

*DENOTES S.A. NOTE ITEM ₵ DENOTES MFRR'S NAME & ADDRESS ARE GIVEN – SEE ACCOMPANYING LIST FCO-0089

**DENOTES NEW DRAWINGS NOT ISSUED WITH THIS RELEASE # DENOTES NEW DRAWING

TOOLING AFFECTED _____ DOES THIS COMPLETE THIS DESIGN _____
INCORPORATE IN ASSY. SPEC. MODELS _____ WEIGHT CHANGE _____
EXPERIMENTAL REQUIREMENTS _____ COST CHANGE _____
DELIVERY REQUIREMENTS _____

MOCKUP AFFECTED _____
SPECIAL PRINT REQ.: LOCATION _____

CHECKER	LOC.	CHIEF DRAFTSMAN	LOC.	PROJECT ENG'R.	LOC.	RECORDS
DRAFT						
TRACE						

FCO-0080 Rev. 6/13/77

6. Engineering Release (2 Pages)

RELEASE NO. SUPP.

2 OF

SUMMARY OF SUBSTANTIATING DATA

CONTRACT NUMBER

CUSTOMER APPROVAL DOCUMENT

REFERENCE DOCUMENTS

EFFECT OF CHANGE ON	MODEL SPEC	TARGET COST OF (PRPLTS) (SPARES)	RETROFIT RECOM
	PERFORMANCE	INSTALLATION	SERV. RECOM
	CONTRACT PERFORMANCE	WEIGHT	RECORD OF PRPLT NOS. REQ'D
	STRENGTH OR DURABILITY	CONTRACT WEIGHT	
	INTERCHANGEABILITY	CENTER OF GRAVITY	
ITEMS AFFECTED BY CHANGE	ASSEMBLY PARTS LIST	TOOLING	
	TABLE OF LIMITS	UNDERSIZE – O'SIZE OR CLASS. PLTS	
	INSTALLATION DRAWINGS	CLEARANCE INDEX CHARTS	
	KITS, SPARES		

ASSEMBLIES INVOLVED REQUIRING NO DRAWING CHANGE	NEW PART NOS. REQ'D.	REASON

ENDORSEMENTS (Initials & Date)					
	ENGINEERING RECORDS	PROJ. ENG.	MANUFACTURING	PURCHASING	PRODUCT ASSURANCE
	MATERIALS		FINANCE	QUALITY ASSURANCE	MATERIAL DEPT.
	ASST. PROJ. ENG.		PRODUCT SUPPORT	PROG. PROJ. MGR.	

REMARKS

FCO-0080 Rev. 6/13/77

(DOCUMENT TYPE & RELEASE NO.) H 1 — 1 4 5 6 15

DISPO 16 — SPR 17

PRINTS 18 — NO 19

MODEL 20 26 — CL 28

ENGINEERING PARTS LIST AND DRAWING WORKSHEET

HEADER NO. 29 31

NO. OF HEADERS 32 34

DATE 35 40

COL. 41 - CORRECTION CODES

- ☐ A-REMOVE ENTIRE JOB
- ☐ B-REPLACE HEADER RECORD
- ☐ C-LINE ITEM CORRECTIONS
- ☐ L-REPROCESS & PRINT

PROJ INFO GR 42

STARTING PAGE NO. 44

SHEET NO. ___ NO. OF SHEETS ___

AREA CODE 45 54

JOB NO. 55 66

SB CODE 67 68

WOWOS 69 79

1	3	5	6	7	8	12	23	24	25	26	30	31	32	33	35	36	37	48	49	54	57	58	59	62	63	74	75	76	80
LINE NO	INSERT NO	TYPE CODE			REF LTR	CHANGE OR ADD P/N		STYLE	TYPE	BOD		SYM	SA	PPS & SUFX			0 1 2 3 4 5	TITLE	ITEM	NO			UNITS PRPLT	CANCEL P/N		$	DISP	SPR	OPEN

FCO-0078 Rev. 6/2/76

7. Engineering Part Lists and Drawing Worksheet

FCO FORM NO. 0357 9/24/79

DOCUMENT TRANSMITTAL

TO:

PAGE ____ OF ____

LETTER NO.

DATE:

PROJECT NO.

SUBJECT:

TRANSMTR: DATE:

DOCUMENT NUMBER	ISSUE	DOCUMENT TITLE/DESCRIPTION	NO. OF SHTS	DOE		GA		PROJECT STATUS	MICRO FICHE	PAPER COPY	MICRO CARD	PRINTS	REPRO PRINT
				CL	UC	P	N-P						

PLEASE CONFIRM RECEIPT OF SHIPMENT BY SIGNING & RETURNING A COPY OF THIS TRANSMITTAL FORM. SIGNED:

DO NOT CONFIRM RECEIPT OF SHIPMENT. DATE:

FROM: _____ SIG: _____

UNITED TECHNOLOGIES
POWER SYSTEMS DIVISION

REMARKS:

REVIEWED _____

8. Document Transmittal

SYSTEM: _____

PAGE _____ OF _____

DATE _____

PREPARED BY _____

APPROVED BY _____

FAILURE MODE AND EFFECT ANALYSIS

SUBSYSTEM: _____

COMPONENT/FUNCTIONAL ASSEMBLY	FAILURE MODE	FAILURE EFFECT	COMPENSATING FEATURE OR CONTROL	REMARKS

9. Failure Mode and Effect Analysis (FMEA)

POWER SYSTEMS DIVISION

UNITED TECHNOLOGIES

INSPECTION AND MATERIAL TEST RECORD

PART OR SPEC NUMBER	PART NAME OR DESCRIPTION	ESA REQUIRED ☐ YES ☐ NO ESA SUPPLIER	TEST FREQUENCY

REQ'D TESTS	SUPPLIER TEST REPORT REQ'D. ☐	QAD REQ'T	DWG /SPEC REQ'TS	S.A.
SPECTRO [ ]				☐
MICRO [ ]				☐
[ ]				☐
[ ]				☐
[ ]				☐
[ ]				☐
[ ]				☐
[ ]				☐

ASSY NO TEST REQ'D [ ]	SAMPLE	REQ'TS ESTABLISHED	REV

REC. REPORT OR W/O NO.	CHG LTR	SUPPLIER	PCS RECD	PCS REJ.	INS BY	MRB/ERB MDOW#	LAST TEST	DATE LAB RELEASED	DATE TO STORES	REMARKS	MISC. SEE BACK
											☐
											☐
											☐
											☐
											☐
											☐
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											☐
											☐

FCO-0012-1 REV. 9/7/76

10. Inspection Material and Test Record (IMTR)

CONTROL NO.	MASTER INSPECTION PLAN	UNITED TECHNOLOGIES POWER SYSTEMS DIVISION	SHEET 1 OF _

PART NUMBER	CHG LTR	PART NAME	QAD	REV LTR	PROGRAM	PREPARED BY
				CNTL NO.	REF P/N	ISSUE DATE

	SEQ	ITEM	SEQ	ITEM		SEQ	ITEM		SEQ	ITEM	REV DATE CNTL No.	REV DATE CNTL No.
DOCUMENTATION		PACKING SLIP		PARTS LIST	MVT		LAB RELEASE	INSPECTION		QAD REQ'TS		
		MATERIAL REPORT		PARTS LIST/ EEE LOT NO'S								
		ACCEPTANCE TEST DATA										
		PWA F3631										

INSPECTION

SEQ	DRAWING CHARACTERISTIC	AUTHORIZED ACCEPT LIMITS (AAL)		TOOLING/FIXTURING	COMMENTS

FCO FORM NUMBER 0198A R10/06/78

11. Master Inspection Plan (MIP)

UNITED TECHNOLOGIES
POWER SYSTEMS DIVISION

INSPECTION PLAN AND STATUS RECORD

PRIORITY

PART NO.	CHG LTR	REC. REPORT/JO NO.	DATE REC'D

PART NAME	QTY PARTS	☐ PROD. DEL ☐ EXPERIMENTAL
SUPPLIER/PSD DEPT.	TYPE SUPPLIER ☐ PP ☐ SI ☐ DIST ☐ OTHER ☐ LCS ☐ COM ☐ ENG	

MATERIAL VERIFICATION AND TEST
☐ LAB TEST REQ'D ☐ MDOW NO.
☐ LAB RELEASED ☐ WAIVED (9)

INSPECTOR ASSIGNED

QUALITY REQUIREMENTS
☐ MIP REV _____ ☐ SAMPLING PLAN _____
☐ VISUAL ☐ SEE REMARKS
☐ DIMENSIONAL ☐ SL NO. _____
☐ QAD C/L & CTL NO. _____ ☐ PL NO. _____

DOCUMENTATION MISSING — TYPE
NOTIFIED _____ DATE _____ RECEIVED DATE _____

DISPOSITION ☐ MRB/ERB NO. _____
☐ ACCEPT ☐ ACCEPT
☐ SEIZURE NO. _____ ☐ RTV-QRO NO. _____
☐ TEMP REL NO. _____ ☐ SCRAP NO. _____

INSPECTION COMPLETED
BY _____ DATE _____

REMARKS _____

FCO FORM NO. 0037A 6-22-77 (FRONT)

SEQ	INSP	SEQ	INSP	SEQ	INSP	SEQ	INSP	SEQ	INSP	SEQ	INSP

FCO FORM NO. 0037A 6-22-77 (BACK)

12. Inspection Plan and Test Record (IPTR)

RECEIVING REPORT

ISSUE DATE	STA	NO	PACKING SLIP	REC'D DATE	BUYER - VENDOR	VENDOR NAME	RPO	CONTAINER	LOCATION	REC	CHG NO

ACCOUNT NO	CONTRACTS APPRO W.O. JOB ORDERS EO EOS (USER DATA)	CHECKED	INSP	PWR SPECS	HEAT	DELIVER TO

SPECIFICATIONS OR REMARKS

ITEM NO.	PART NUMBER	PART NAME OR DESCRIPTION	U	F	P/S QTY.	QTY. REC'D.	✓	K	CHANGE OR GROUP	OVERSHIPPED QUANTITY

13. Receiving Report (RR)

INSPECTION JOB ORDER

SERIAL NO.	PART OR TOOL NUMBER	PART NAME	QUAN.	PROJ CODE	DATE OF MEMO
6					

1–2 3–9 23–24 37–38 41–42 44

E.O.	E.O.S	EXPED. 1ST	CD 2ND	PMC	APE	TARGET DATE	PRIORITY CD	AUTH BY	CENTER	EST. HOURS	ERB/MRB NO.

45 50–51 54–55 58–59 61–62 64–65 69–70 71 73–74 77

ENGINEER	PHONE	REQ'D DATE	REMARKS

ITEM	INSPECTION REQUIREMENTS	Qual Assur. By & Date
	Report of work done:	

ECO 0045 10-24-75

14. Inspection Job Order

JOB ORDER

TC	PART OR TOOL NUMBER	SERIAL NO.	E.O.	E.O.S.	ENG/RIG	BLD	CHG
4 8							
2 3	4 18	19 25	26 31	32 35	36 41	42 44	48 50

PROJ. CODE	PART NAME	ENGINEER	PHONE	REQ'D DATE	ISSUED BY
51 53	54 67				

TC	REF. PART OR TOOL NUMBER	SERIAL NO.	PMC	ORDER	QUAN. REQ'D	ORDER DATE	TARGET DATE
4 9							
2 3	4 18	19 25	26 28	29 32	33 36	37 41	42 46

MAT'L DATE	OP. SH. DATE	EXP. CD.	P C C	R C	MAKE IN DEPT (S)	DELIVER TO
					☐ 963 ☐ 954 ☐ 80963 ☐ 80955 ☐ OTHER	
47	51 52	56 57	60 61	66 71		

TC	SEQUENCE	SERIAL NO.	HU	APE	PLAN	TOOL REQ'D DATES	REQUISITION NO.	ENG. MODEL	SHOP LOC.
5 0									
2 3	4 8	19 25	41 42	48	50 53	54			

☐ QAD SHEETS	☐ MIN. TOL. ±.010	☐ OPER SHEET REQ'D	INSP REQUIREMENTS	PROJECT	AUTHORITY	DATE

REV. PART NUMBER	CHG.	E.C. OR ECM NO.	DATE	BY

REV QTY.	DATE	NO. PCS.	FOR	BY

RT.	LOC	HR	IN	OUT	DATE COMP.	RT.	LOC.	HRS.	IN	OUT	DATE COMP.	RT	LOC.	HRS.	IN	OUT	DATE COMP.
010						020						030					
040						050						060					
070						080						090					

REMARKS:

FCO-0001
4/1/75

-1-

15. Job Order

UNITED TECHNOLOGIES
POWER SYSTEMS DIVISION

X-RAY JOB ORDER

SERIAL NO.	PART OR TOOL NUMBER	PART NAME	QUAN.	X-RAY NO.

E.O.	E.O.S.	ENGINEER	DEPT.	PHONE	REQ'D DATE	DATE

SEE REVERSE SIDE FOR X-RAY TECHNIQUE

ITEM	INSPECTION REQUIREMENTS

Report of work done:

	FOR X-RAY USE ONLY

FCO FORM 0193 3-14-77 (FRONT)

16. X-Ray Job Order

UNITED TECHNOLOGIES POWER SYSTEMS DIVISION

NONCONFORMANCE REPORT

No. 4207

ITEM			SYSTEM		
PN-CL	SN		PN-CL	SN	
NAME	QTY		NAME	QTY	
SOURCE	DATE		SOURCE		
PS NO.	RCVR		PO NO.		

PRGM	EXP	COM	GOV'T	G/MRB	TYPE	DESIGN	MAT'L	IP	SF	F	LOC	SUPL	RCVG	MFG	ASSY	TEST	PKG
	E	C	G	G/MRB		D	M	I	SF	F		S	R	MF	A	T	P

MODEL	AUTH	QA	ENG'R	PROJECT	PURCH

NONCONFORMANCE

NAME/DATE

EVALUATION/CORRECTIVE ACTION

	DISPOSITION	MARKING
	ACCEPT	
	REPAIR/RESUB	
	REWORK/REINSP	
	RTV/SCRAP	
	OTHER	

NAME/DATE NAME/DATE

APPROVALS

D M E C G G/MRB

DESIGN	MATERIALS	ENG'R	PROJECT	QA	GOV'T

FCO FORM NO. 0295 R7/27/78

1

17. Nonconformance Report

OPERATION SHEET

PART NAME	STATION	OPER. NO.	MODEL CODE	PART NO.	CHG.
OPERATION	DEPT. NO.	W/O SERIAL NO	ENG.OR RIG NO.	SHEET OF	

TOOLS

ISSUE DATE	PREPARED BY
REVISION DATE	APPROVED BY

18. Operation Sheet

MODEL CODE _____

REF. PART NO. _____

CHG. LETTER _____

ASSEMBLY SHEET

PART NAME _____

PAS _____

SHT NO ____ OF _____

ENG OR RIG NO _____

OPER. NO.	OPERATION	TOOLS	OPERATOR CLOCK NO	DATE	WITNESS BY:	SKETCH

PREPARED BY _____ DATE _____

REVISED BY _____ DATE _____

19. Assembly Sheet