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Advanced Turbine Systems Study
System Scoping and Feasibility Study

Final Report

Work Performed Under Contract No.: DE-AC21-92MC29247

For
U.S. Department of Energy
Office of Fossil Energy
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Glossary of Terms

ATS - Advanced Turbine Study
CF - Capacity Factor
COE - Cost of Electricity
ETA - Efficiency (η)
HAT - Humid Air Turbine
HHV - Higher Heating Value
HPC - High Pressure Compressor
HPT - High Pressure Turbine
IGHAT - Integrated Gasifiation Humid Air Turbine
IPP - Independent Power Producer
ISO - International Standard Organization
kW - kilowatt
LPC - Low Pressure Compressor
LPT - Low Pressure Turbine
LHV - Lower Heating Value
MW - Megawatt
NGHAT - Natural Gas Humid Air Turbine
O&M - Operation & Maintenance
PW - Pratt & Whitney
TCA - Turbine Cooling Air
TOBI - Turbine On Board Injector

Executive Summary

United Technologies Research Center, Pratt & Whitney Commercial Engine Business, And Pratt & Whitney Government Engine and Space Propulsion has performed a preliminary analysis of an Advanced Turbine System (ATS) under Contract DE-AC21-92MC29247 with the Morgantown Energy Technology Center. The natural gas-fired reference system identified by the UTC team is the Humid Air Turbine (HAT) Cycle in which the gas turbine exhaust heat and heat rejected from the intercooler is used in a saturator to humidify the high pressure compressor discharge air. This results in a significant increase in flow through the turbine at no increase in compressor power. Using technology based on the PW FT4000, the industrial engine derivative of the PW4000, currently under development by PW, the system would have an output of approximately 209 MW and an efficiency of 55.3%. Through use of advanced cooling and materials technologies similar to those currently in the newest generation military aircraft engines, a growth version of this engine could attain approximately 295 MW output at an efficiency of 61.5%. There is the potential for even higher performance in the future as technology from aerospace R&D programs is adapted to aero-derivative industrial engines.

Estimates of emissions for the HAT Cycle, based on theoretical considerations, indicate that NOx, CO, and HC emissions would be <5 ppm. The emission of CO_2, a potential "Greenhouse" gas, would be low on a per kWhr basis because of the high efficiency. Estimates of capital costs using the EPRI guidelines show a total capital requirement of $511/kW and a 30-yr levelized cost of electricity (COE) of 40.75 mills/kWhr at a capacity factor of 0.8 and a fuel cost of $2.75 million/Btu (all costs in 1991 $'s) for the 295 MW, 61.5% plant. This is approximately 20% less than a 1994 technology heavy frame gas turbine combined cycle.

A six year schedule for the natural gas-fired HAT Cycle development was defined with development costs through prototype operation estimated to be $232 million. A four year program at an estimated $100 million additional would be required to demonstrate a coal gas-fired HAT. This system, using a Texaco-type gasifier and having the advanced technology of the advanced natural gas-fired HAT Cycle, would have a projected performance of approximately 48.7%, coal in to kW out.

ADVANCED TURBINE SYSTEMS STUDY
SYSTEM SCOPING AND FEASIBILITY STUDIES

1.0 - Introduction

Over the next several decades, the US utility industry will need to replace existing plants nearing the end of their useful life, as well as add new generating capacity. The potential additions for the US and Canada could approach 115,000 MW by 2001. These new additions and replacement units must meet increasingly stringent emission requirements, conserve fuel, thus reducing production of CO2 and other “Greenhouse” gases, while producing electricity at costs enough lower than current plants that their development can be justified.

Developments in the last decade in gas turbine technology have provided a basis for simple-cycle power systems having high efficiencies (>40%), lower capital cost, more rapid installation, and lower emissions. The projected availability of competitively priced natural gas has made possible the consideration of these gas turbine-based plants as high load factor plants. Combined-cycle systems based on large, heavy “frame” machines are being ordered on a regular basis for base-load utility use. Smaller machines, typically aero-derivative turbines are being widely used by Independent Power Producers (IPP) and private operators for industrial cogeneration.

The United Technologies Corporation has been in the forefront of providing the utility market with aero-derivative turbines. Since its first installation of a aero-derivative turbine for industrial use in 1960, over 1200 units have been marketed including over 22,000 MW in peaking and combined cycle use. Currently, UTC has adapted the JT8D, the most widely used aircraft engine (over 14,000 in airline operation) to industrial service, introducing the FT8 in 1989. Since then, over fifty of these nominal 25 MW machines operating at a pressure ratio of 18 and at firing temperatures of over 2300 F are installed or on order.

To meet the demands of the market place for a more efficient, larger machine, development has begun on industrializing the PW4000, (Fig. 1.0.1) a 60,000 to 90,000 lb thrust engine used in today's most modern wide-body aircraft. The FT4000 (Fig.1.0.2) is a nominal 47 MW engine with a pressure ratio over 31 and a rotor inlet temperature (RIT) over 2300 F. Unlike previous PW industrial engines, the low pressure spool of the two spool FT4000 operates at 3600 rpm, negating the need for a separate power turbine (for 60 Hz operation).

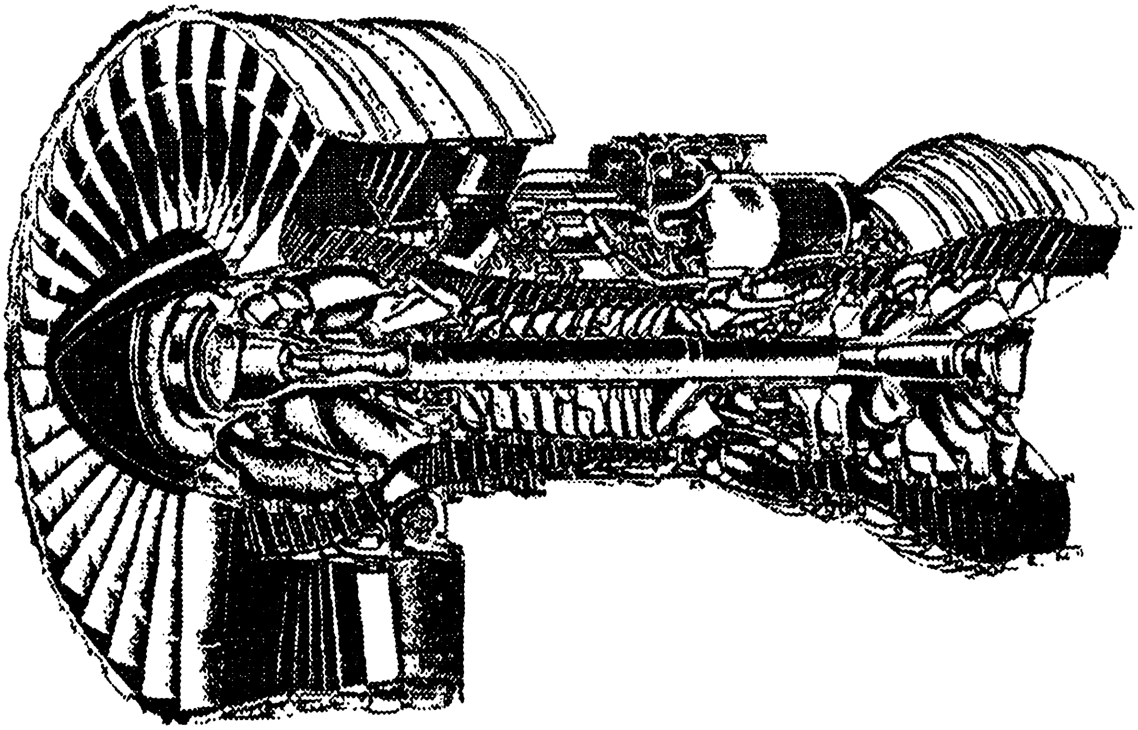

Fig. 1.0.1 - Cutaway of PW4000

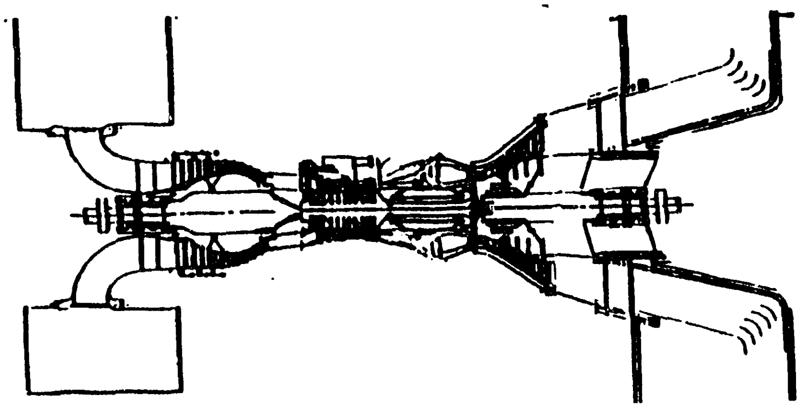

Fig. 1.0.2 - FT4000 Cross Section

The application of aircraft turbine technology to large-scale utility use can be accomplished in two ways; 1) the introduction of the advanced aerodynamics and turbine cooling techniques to large, heavy frame machines; and 2) the development of advanced power cycles taking direct advantage of the high pressure ratios and high turbine temperatures. The first method is being practiced by most aircraft turbine manufactures, including UTC, who is supplying State-of-the-Art technology to Siemans for use in their V-series of heavy frame machines. This technology transfer can be relatively slow and may not take full advantage of the advanced technology.

Under normal development conditions, the manufacturers of gas turbines will make only small, incremental advances in turbine technology over the next several years. While these turbines can be used in power systems having efficiencies approaching 55% (LHV), further advances can be made only by significant cycle and turbine modification. This is because machines appearing in the mid-1990 decade will have essentially reached the plateau of current turbine cooling technology based on extraction air. The attainment of higher efficiencies combined with lower emissions and comparable cost of electricity will require cycle and machinery changes including, among others, intercooling, improved turbine cooling, flow augmentation by steam or water vapor, and combustor and turbine material improvements. At current internal funding levels, such changes will appear gradually over the next 10 or 15 years.

To assure that these advances are developed in a timely manner, the Department of Energy/Morgantown Energy Technology Center has initiated the Advanced Turbine Systems Program. The goal of the ATS program is the development of an ultra-high efficiency gas turbine system that will be both environmentally superior and cost competitive in base-load utility applications, industrial cogeneration applications, and for use by IPP's. These advanced systems are to be commercially available by 2002. Specifically, these goals require the ATS demonstrate:

- 60% or greater cycle efficiency (natural gas lower heating value (LHV)),
- emissions of NOx, CO, and HC's of less than 5 ppm, and
- cost of electricity (COE) comparable to, or less than, current state-of-the-art power systems.

The definition and preliminary engineering of an advanced power cycle based on FT4000 technology was undertaken by a team of UTC, Fluor Daniel, Inc., Texaco, and EPRI using privates funds in 1991 (Ref. 1). The Humid Air Turbine (HAT) cycle is patented by Flour Daniel

(Ref. 2). This cycle, shown simplistically in Fig. 1.0.3, uses exhaust heat from the turbine in a recuperator and heat from within the cycle in the saturator to supply heated, water-vapor laden air to the turbine. Both the increased mass flow of the humid air and the higher specific work function of the mixture provide additional power but require no additional compression work from the turbine, resulting in both efficiency and power output gains. The Ref. 1 design study was based on the use of coal-derived gaseous fuel produced by the Texaco entrained flow gasifier. Considerable advantage can be made of the heat in the gasifier effluent by using it to humidify the compressor discharge air. To date, only cursory attention has been paid to a natural gas-fired HAT cycle.

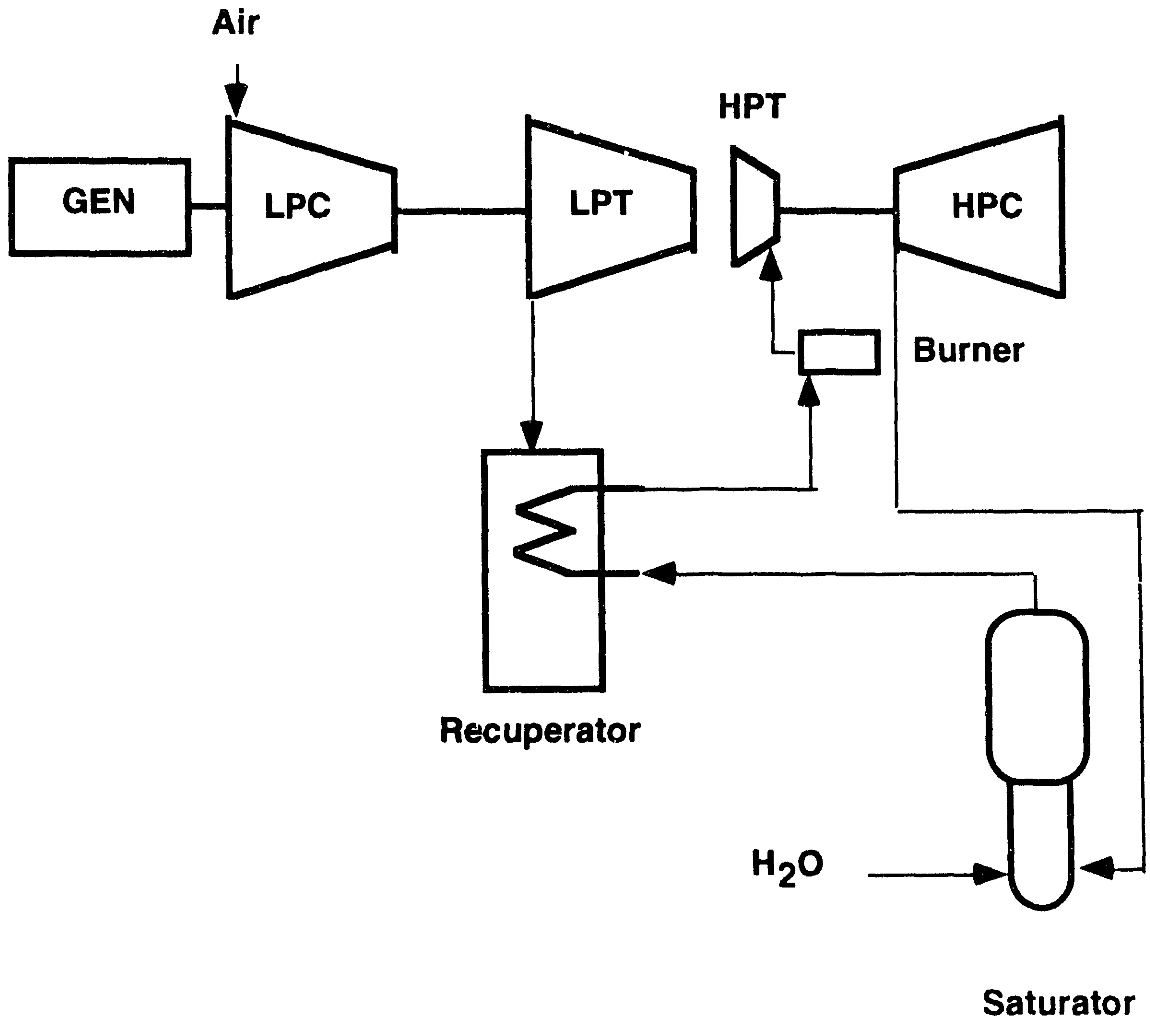

Fig. 1.0.3 Simplified HAT Cycle Schematic

1.1 - Selection of Natural Gas-Fired Reference System

The initial phases of a development program on the nominal 45 MW aero-derivative FT4000 are underway at UTC. When this engine is introduced in the mid-1990's, it will have performance features at the forefront of its class. The major characteristics of this engine are given in Table 1.1.1. Because of its high level of performance and leading edge technology, the FT4000 has been selected as the basis for the ATS. A number of internal studies at UTC have shown that one of the most attractive applications of the FT4000 technology would be in the HAT-cycle configuration.

Table 1.1.1 - Major Characteristics of the FT4000
(60 Cycle - Natural Gas Fuel)

Inlet Flow - lb/sec	330
Overall Pressure Ratio	31.2
Rotor Inlet Temperature - F	2300+
Exhaust Temperature - F	796
Nominal Power Output - MW	47
Efficiency - %	41.7

HAT Cycle Configuration

A schematic of the HAT cycle is given in Fig. 1.1.1. It can be seen that the cycle makes use of intercooling to reduce compressor work and recovers the heat from the intercooler and the turbine exhaust to heat water for use in humidifying the compressor discharge air. The humidified air is further heated by the turbine exhaust before entering the combustor. The use of an intercooler introduces an opportunity to change the physical layout of the FT4000. It was shown in Fig. 1.0.2 that the FT4000 is a twin shaft machine having the high compressor and high turbine on one shaft located concentrically with the second shaft carrying the low compressor and low turbine.

The need to remove air between the low and high compressors to send to the intercooler would require significant changes to the shafting and bearings for the FT4000. Similarly, the ducting for the recuperated air would change the high turbine location. A new arrangement of the gas turbine would retain the two shaft design, but would use two cases, a high pressure case and a low pressure case to house the engine (Fig. 1.1.2.) This arrangement allows extraction/re-introduction of air with minimum duct losses and gives some advantages in engine bearing location. It has the additional advantage of not requiring the large, low pressure rotor shaft to pass through the high pressure rotor.

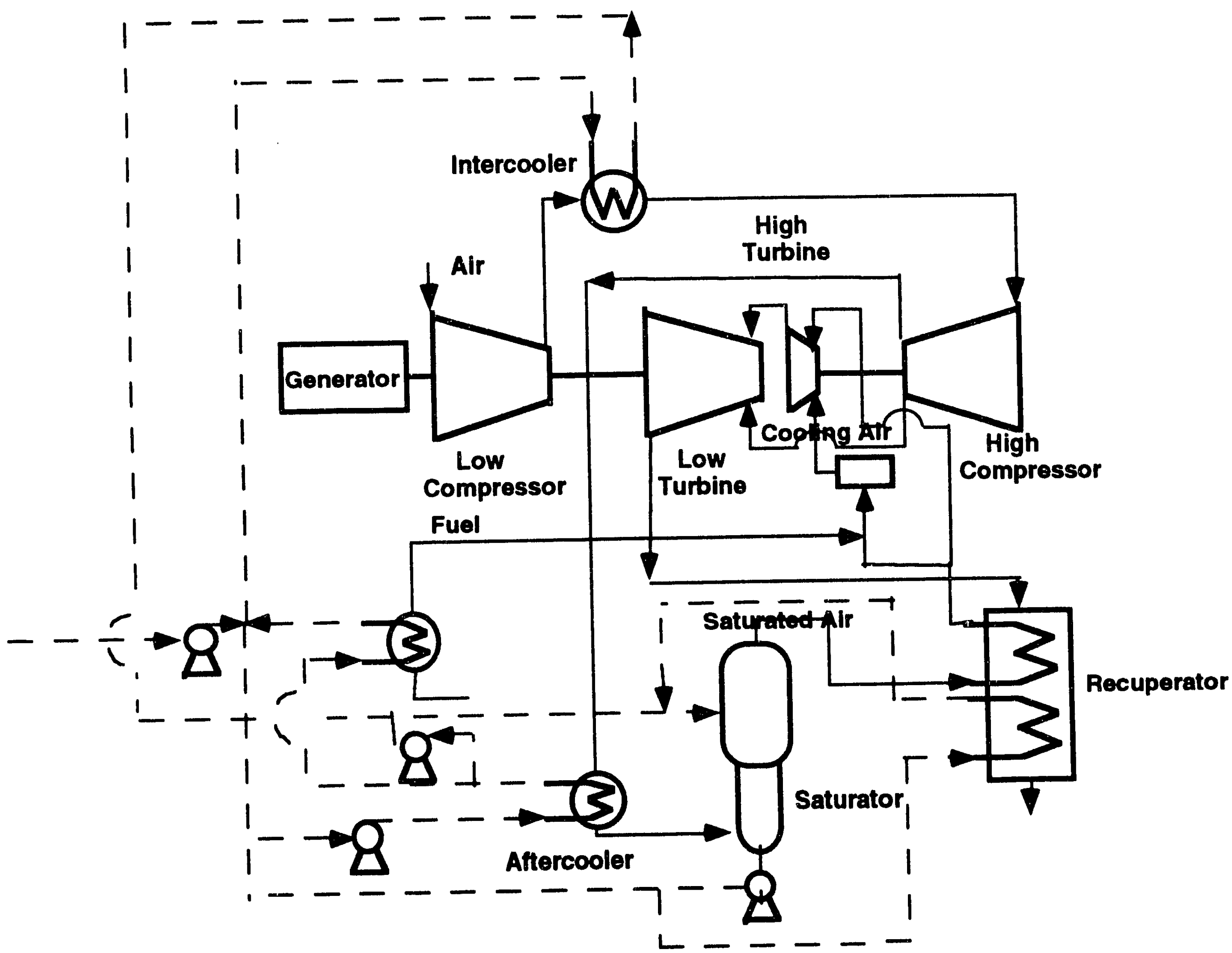

Fig. 1.1.1 - Natural Gas-Fired HAT Cycle

One of the results of the ATS Scoping and Feasibility Studies is to be the identification of the changes required to go from natural-gas fired systems to coal-derived fuel gas-based systems. The UTC approach is unique in that considerable effort has been already expended on the design of an advanced integrated coal gas-based HAT cycle power system (IGHAT). Since these design studies have been based on the use of PW FT4000 technology, the major consideration in identifying a natural gas-fired HAT (NGHAT) will be to maintain technology commonalty among the NGHAT, the IGHAT, and simple-cycle FT4000 as much as possible without compromising performance. This will assure the most economic development of advanced systems capable of burning a wide variety of fuels.

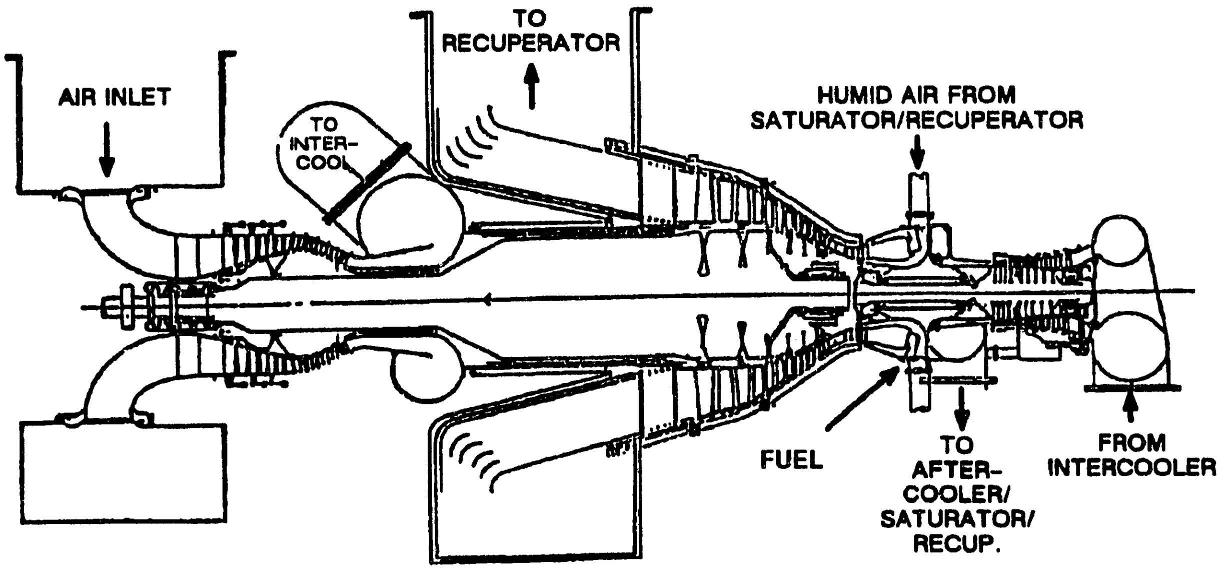

Fig. 1.1.2 - FT4000 HAT Cycle Configuration

1.1.1 - ATS Performance

A unique characteristic of the HAT cycle is the large difference between gas flow rate through the turbine and the air flow rate through the compressor of the HAT cycle turbine. This flow difference depends on the quantity of water vapor added to the compressed air which in turn is a function of the heat availability for evaporating water into the air. Therefore, the machine specific power output, i.e., the net power output per unit of air flow through the compressor would be significantly more in the IGHAT than in the NGHAT. This is because in a NGHAT cycle plant, the only heat available for evaporating the water is the heat from within the cycle. When the HAT cycle is integrated with a coal gasifier such as the Texaco total quench gasification plant, large quantities of heat are available as thermal energy from the gasifier reaction products. Thus, for a given size air compressor and corresponding air flow, the gas flow through the turbine could be widely different depending on whether the HAT cycle application is in natural gas fired plants or integrated with a gasification plant.

1.1.1.1 Design Approach

Two different commonalty approaches are available for NGHAT and IGHAT, a "common compressor" approach or a "common turbine" approach.

"Common Compressor" Approach

One approach in developing a NGHAT machine is to maintain maximum commonalty with the IGHAT machine by keeping the compressor (high and low pressure) common for both applications and make design changes in the turbine to account for the difference in fuel and moisture mass flows. Some commonalty would be maintained in the two turbine designs, such as the turbine cases and rotor systems (including disks and bearings) and in the combustion system design. Fuel nozzles and turbine air foil and flow paths were to be changed in converting the natural gas fired machine to syngas applications.

"Common Turbine" Approach

A second approach, which generates higher power at essentially the same efficiency, is also available. This approach consists of maintaining the IGHAT turbine hardware design and tailoring the compressor system to maximize the HAT system power output with natural gas fuel. This approach, which minimizes the overall NGHAT/IGHAT development costs, was selected for the ATS study.

The new compression system handles higher airflow to make up for the reduced water vapor production (vis' a vis the IGHAT) and thereby supply the same high mass flow to the combustor and turbine section. The larger turbine designed for the IGHAT application can be used. A comparison of the common compressor (CC) design and the common turbine (CT) design is given in Table 1.1.2. Note that in Table 1.1.2, the technology level is the same as the FT4000 in its initial offering in simple-cycle form.

In order to have a common turbine for both IGHAT and NGHAT designs, it is necessary to hold the same overall pressure ratio while increasing the air flow by 28% to approximately 627 lb/sec. The compressor work split had to be changed to match the low and high turbine work split. The LPC and HPC pressure ratios are respectively increased from 4.9 to 6.9 and reduced from 8.4 to 6.1. This is accomplished by adding stages to the front of the LPC in order to increase the air flow and delete stages from the rear of the HPC in order to retain the same overall pressure ratio and corrected inlet flow to the turbine for both designs.

The RIT of the NGHAT is approximately 100F higher than the base FT4000. This is because: 1) the intercooler reduces the temperature of the compressor discharge (< 650 F); and 2) humid air is used to cool the first vane.

Table 1.1.2- Performance Summary for NGHAT*

	CC	CT
Inlet Flow, lb/sec.	464	591
LPC Pressure Ratio	4.94	6.92
LPC Corrected Flow, lb/sec.	483	614
HPC Pressure Ratio	8.40	6.12
HPC Corrected Flow, lb/sec.	107	97
Rotor Inlet Temperature, F	2408	2408
Overall Pressure Ratio	38.1	38.9
Water/Air Ratio in Saturator Exit, lb/lb	0.23	0.24
Fuel Flow, lb/sec.	12.5	15.9
HPT Area, sq. in.	107.0	133.0
LPT Area, sq. in.	228.0	249.4
Gas Flow (Turb exit), lb/sec.	557	713
Gross Output at Generator Terminals, MW	157	200
Net Plant Output, MW	156	199
Net Plant Heat Rate, Btu/kWhr (LHV)	6154	6186
Net Plant Efficiency, % (LHV)	55.5	55.3

* At ISO plus 31 F. From Ref. 1.

1.1.1.2 Cycle Parametrics

The performance of the NGHAT using initial FT4000 technology is better than, or comparable to, that projected for the best combined cycles using heavy frame machines (55.3% vs. 52.6% (Ref. 1) to 54% (Ref. 3)). However, it is slightly over 9% (5.7 pts) less than the ATS goal of 60%. A series of parametric analyses were performed to identify the conditions needed to achieve 60%. These analyses were performed with the UTC State-of-the-Art Performance Program (SOAPP) briefly described in Appendix A. The parameters varied were component efficiency, turbine inlet temperature, and turbine metal hot-spot temperature. In all analyses, the compression system was held constant (constant airflow and pressure ratio).

As part of the Ref. 1 study, a number of cooling alternatives were assessed. The scheme selected uses humid air from the recuperator diffuser inlet to cool the first vane and extraction air from matching compressor bleeds to cool all other areas. This approach resulted in maximum cycle efficiency while minimizing risk to the engine from piping external to the engine control which could preclude positive turbine protection under a number of failure moods. For example, cooler humid air could be taken from the line leading from the saturator to the recuperator. However, flow in this line could be interrupted by a failure in the water system, leading to catastrophic turbine failure.

Appendix B discusses advances projected for aircraft turbine cooling and materials. These advances will appear in commercial flight engines over the next decade and will be used in later generation aero-derivative industrial turbines. With accelerated development resulting from programs such as the ATS, they could appear in the 1996 - 2002 time frame.

Component Efficiency

The levels of turbomachinery component efficiency and the potential improvement are shown in Table 1.1.3.

Table 1.1.3 - Component Efficiencies

Component	Representative Base Level	Potential Level	Cycle Efficiency Effect
LPC	90% Poly*	92% Poly	+.25 pts
HPC	90% Poly	90.5% Poly	+.05 pts
HPT	89.7% Ad**	89.7% Ad	————
LPT	92% Ad	94% Ad	+.85 pts
			+1.15 pts

* Polytropic
** Adiabatic

At the base rotor inlet temperature (RIT), the cycle efficiency with "best" components would be 55.3% + 1.15% = 56.5%. The component efficiency is affected by turbine inlet temperature as shown in Fig. 1.1.3. The effects of component efficiencies on cycle efficiency and on cycle power output are shown in Figs. 1.1.4 and 1.1.5, respectively, as a function of turbine inlet temperature.

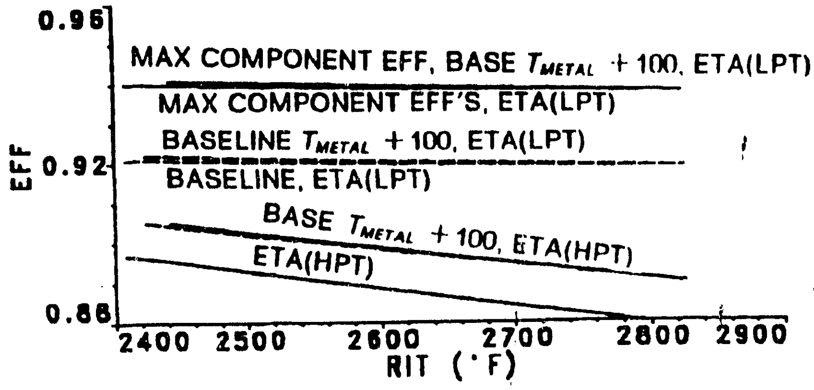

Fig. 1.1.3 - Effect of RIT on Component Efficiency

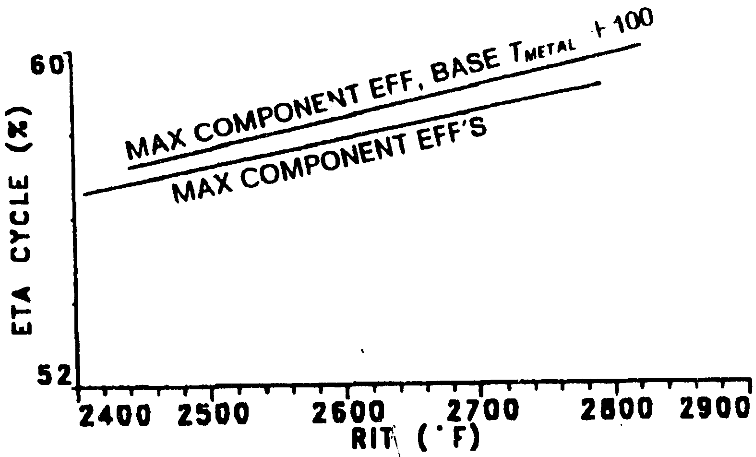

Fig. 1.1.4 - Effect of Component Efficiecy on Cycle Efcieincy

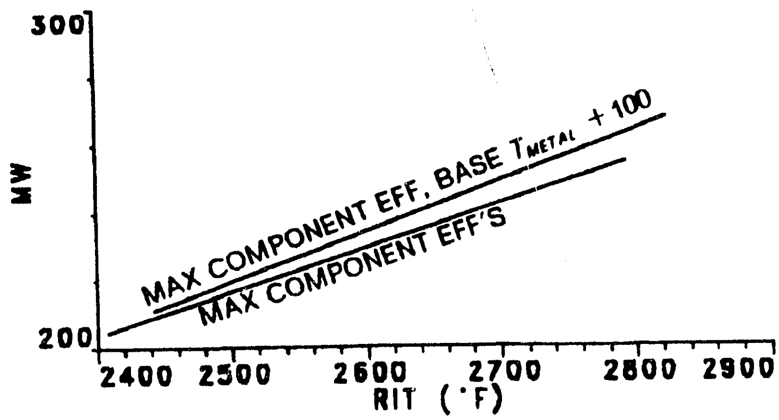

Fig. 1.1.5 - Effect of Component Efficiency on Power

Turbine Inlet Temperature

The turbine inlet temperature was varied by 500 F from 2400 F to 2900 F RIT. The effects of this variation, at cooling effectiveness levels typical of industrial applications such as the FT4000, are also shown in Figs. 1.1.4 and 1.1.5. The cooling flows for the LPT and HPT are shown in Fig. 1.1.6.

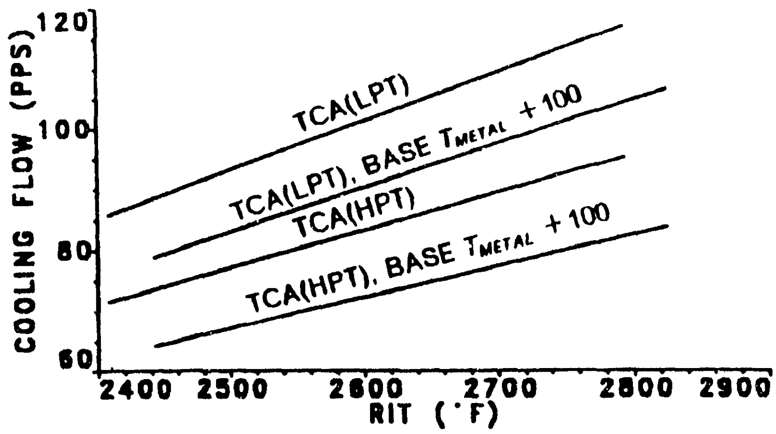

Fig. 1.1.6 - Effect of Hot Spot Temperature on Cooling Flow

Hot Spot Metal Temperature

Since the driver in industrial turbine part lifetime is erosion/ corrosion, and, to some extent, local creep, it is hot spot temperature, not bulk metal temperature that is the key turbine metal temperature. Using lifetime criteria similar to those of heavy frame machines, e.g., 100,000 hr part life with allowable refurbishment at 25,000 hr intervals, there is a potential for a 100 F increase in metal temperature during the time frame of the ATS program. The effect of this increase is shown in Fig. 1.1.7. Cycle efficiencies of nearly 60% were estimated at a RIT of approximately 2870 F at the 100F increase.

Advanced Turbine Cooling

Analysis was made of a closed-loop water system to cool the vane in the HPT and first vane in the LPT. The heat was recovered in the humidification circuit. It was anticipated that the reduction in bleed air would significantly improve performance. While turbine output did increase, cycle efficiency decreased. The analysis showed that because the water cooling removes energy from the flow path at the highest pressure portion of the cycle and re-introduces it in the saturator at a

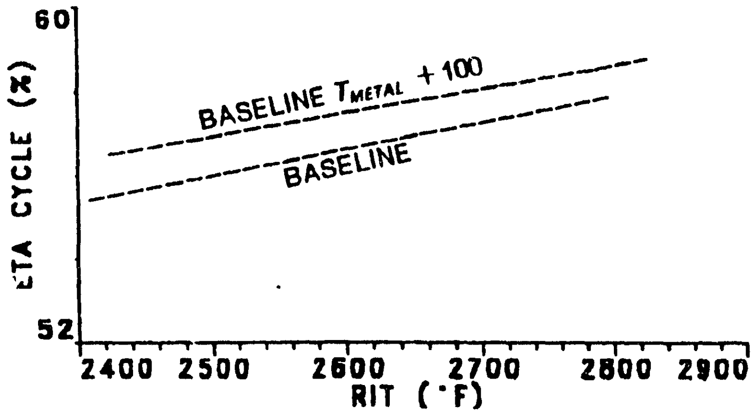

Fig. 1.1.7 - Effect of Metal Hot Spot Temperature on Cycle Efficiency

lower cycle temperature, the average temperature of heat addition is reduced. No increase in turbine temperature was investigated. (The air cooling scheme is open loop meaning that the heated cooling air is introduced into the cycle at a somewhat reduced pressure and expanded through the remainder of the turbine. No consideration was given to an open loop water cooling scheme at this time.)

An advanced air cooling scheme using components with higher cooling effectiveness was considered (see Appendix B). This cooling technology is currently used in PW's newest aircraft engines. With the advanced cooling, the ATS goal of 60% efficiency was achieved at a rotor inlet temperature of approximately 2700 F (Fig. 1.1.8). With the 100 F higher metal temperature, the efficiency is nearly 61%. Using the full 300 F temperature increment allowed by the advanced cooling techniques, an efficiency of approximately 61.5% would be attainable at the 100 F higher metal temperature. Power reached an estimated 295 MW (Fig. 1.1.9).

Cycle Modification

The basic HAT cycle was modified by the addition of a reheat combustor (Fig. 1.1.10). At the baseline RIT condition (2408 F), with reheat to the same temperature, the cycle efficiency was increased by 1.1 pts (2%) to 56.4% and the power output increased by over 41 MW to 241.5 MW. As turbine inlet temperature increases, the efficiency gains through reheat decrease because of the requirement for additional turbine cooling in the LPT. Using the advanced cooling scheme, the incremental efficiency was only 0.4 pts (<1%). Power was increased significantly, over 50 MW. From an efficiency viewpoint, the value of

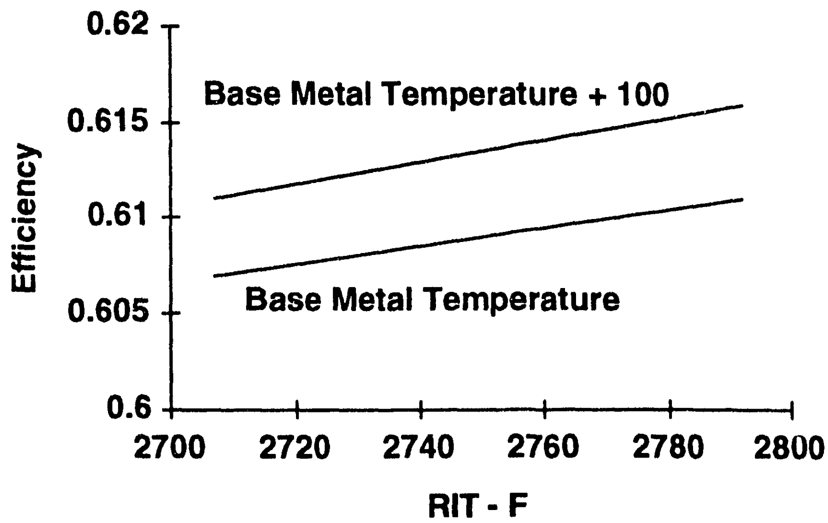

Fig. 1.1.8 - Effect of Hot Spot Metal Temperature on Efficiency

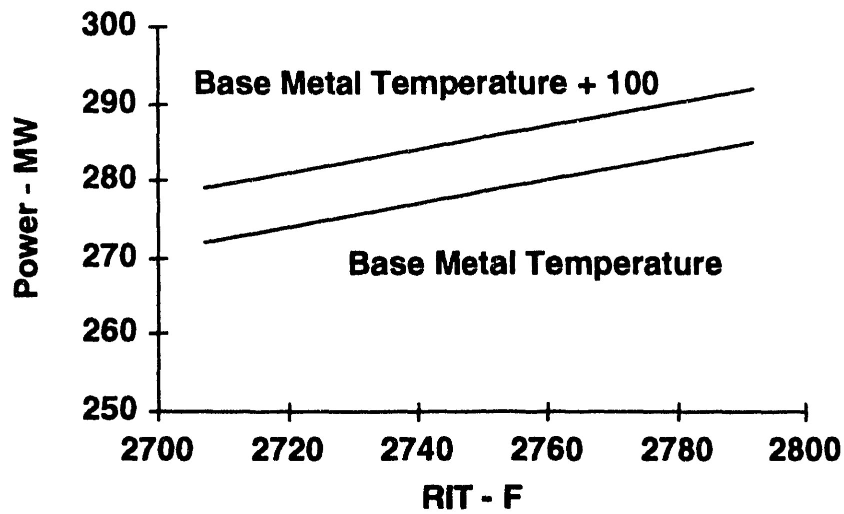

Fig. 1.1.9 - Effect of Hot Spot Metal Temperature on Power

reheat at these high turbine inlet temperatures is marginal. A new LPT design, a new reheat combustor, and high temperature ducting would have to be developed, resulting in appreciably higher program costs. While cycle modifications such as reheat remain an option, advances in materials and cooling techniques should provide less costly efficiency increments.

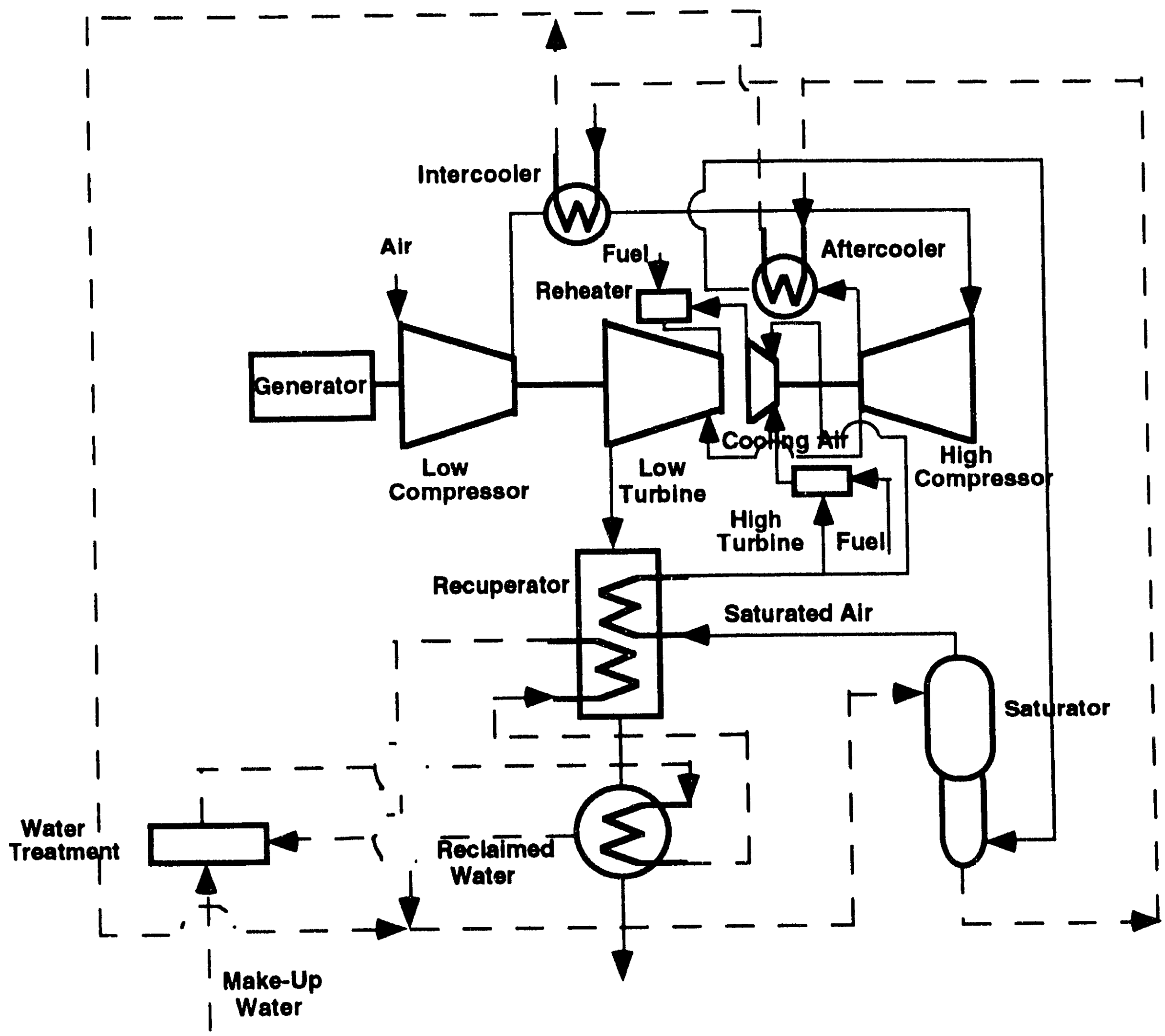

Fig. 1.1.10 -Natural Gas-Fired HAT Cycle with R/H

1.1.2 - Base Load Emissions

UTC is currently developing dry-NOx combustor technology for the FT8 which will meet current and projected NOx regulations. This technology will be available for use in FT4000 and HAT applications. A discussion of the NOx and other emission problems and our approach to solving them is given in Appendix C.

Analytic estimates of the emissions from a NGHAT system were developed as part of the Ref. 1 study. They are presented in Table 1.1.4. As discussed in Appendix C, application of combustor advances including new cooling techniques and the use of high amounts of water vapor should allow comparable emission levels for the NGHAT using advanced turbine cooling technology.

The emissions of CO and HC are somewhat more difficult to estimate. Very preliminary analysis (Ref. 4) has indicated that the residence time needed for complete carbon burnout (mainly CO) in a gas stream containing water vapor fractions of 30% to 45% could be as much as 3 or more times that encountered in normal combustors. The necessary space to attain complete burnout is available in the NGHAT configuration. Therefore, it is estimated that CO and HC goals can be attained, but empirical data will have to be obtained to verify this supposition.

Table 1.1.4 - Estimated Emissions for NGHAT System*

NOx ppm(v) (dry @ 15% O_2)	< 5
CO_2, lb/kWhr	0.724
CO ppm(v)	<5
HC ppm(v)	<5
Raw Water Intake, gal/kWhr	
Cooling tower makeup	0.26
Remainder of plant	0.278

* Emissions must be verified by experiment.

1.1.3 - Cost of Electricity

The cost of electricity (COE) estimates are based on the methodology outlined in the EPRI document "Economic Premises for Electric Power Generating Plants" extracted from the Technical Assessment Guide, Vol. 1 Rev. 6, 1989 (EPRI Report No. P-6587).

1.1.3.1 - Capital Costs

The capital costs are made up of a number of cost components and are summed to define the total plant investment:

- direct field material costs
- direct field labor costs
- indirect field costs
- home office engineering
- sales tax
- contingency
- initial chemicals

The cost base for a NGHAT plant is detailed in Ref. 1. Each of the foregoing is described including the rationale for determining contingency. The computer simulation program used in Ref. 1 is based on experience constructing various chemical and power plants to determine a sampling distribution of for project mean costs. At the desired confidence level of a 50% probability of project underrun, the contingency for the NGHAT was 11% (In comparison, the NG combined cycle in Ref. 1 had a contingency of 9%.) Total plant costs for the NGHAT are given in Table 1.1.5. These costs are in Jan. 1991 dollars for the 61.5%, 295 MW system (ISO Rating).

Table 1.1.5 - NGHAT Capital Costs
(Thousand 1991 $'s)

Machinery and Equipment	
Gas Turbine Generator	$54102
Air Saturator	4156
Heat Recovery Unit	9727
Balance of Plant	7081
Total Equipment	$75066
Bulk Materials	12041
Direct Field Labor	5095
Total Direct Field Cost	$92202
Indirect Field Cost, Home Office, Engineering & Sales Tax	$20658
Total field Cost	$112860
Contingency (11%)	12415
Total Plant Cost	$125275
$/kW	$425

Using the EPRI methodology, the total plant investment (TPI), based on a three year construction time, is found to be TPC x 1.062, or $133,042,000.

In addition to the above total plant TPI, there are start-up costs and the requirement for working capital. The start-up costs are comprised of one month of fixed operating and maintenance costs, two months of consumables excluding fuel, one month of fuel at 125%, and two per cent of TPC. The working capital is the sum of two months of consumables excluding fuel, two months of fuel at full capacity, three months labor, two months O&M costs, spare parts inventory at 0.5% TPC, a contingency of 25% of the last four items. The cost of land is site specific and is not included in the costs. These costs are added to the TPI to determine the total capital requirement (TCR) used in the calculation of the thirty year levelized COE.

1.1.3.2 - Operating and Maintenance Costs

The operating and maintenance (O&M) costs are made up of fixed and variable costs. The fixed costs are operating labor, fixed maintenance costs, and overhead charges. The variable operating costs are those for raw water, chemicals and other consumables, and variable maintenance and materials. The estimate for labor in Ref. 1 indicates 11 men/shift at a rate of $22.10/hr (1991 $'s). Maintenance costs are estimated as a percentage of the installed cost on a system by system basis. The values used are: gas turbine generator, 1.5%; saturator 2.%; heat recovery, 1.5%; and general facilities, 1.5%. Having identified a maintenance cost, it is divided into fixed costs (65%) and variable costs (35%). The overhead charge is that for administration and support labor, assumed to be 30% of O&M labor. These costs for the NGHAT are given in Table 1.1.6.

Table 1.1.6 - Annual Operating and Maintenance Costs
(Thousand 1991 $'s)

Fixed Operating Costs	
Operating Labor	$2129.34
Maintenance Labor 376.60	
Maintenance Materials	862.45
Overhead Charges	812.69
Total	$4181.09
Fixed O&M $/kW-yr	14.67

Variable Operating Costs

Raw Water	$470.65
Chemicals	761.65
Maintenance Labor 202.78	
Maintenance Materials	464.40
Total	$1899.48
Variable O&M mills/kWhr	0.761

The estimation of the COE is shown in Table 1.1.7. The results are shown in Fig. 1.1.10 as a function of capacity factor (CF) and fuel cost. At the design CF of 80% and a 1991 fuel cost of $2.75/million Btu, the COE is 40.75 mills/kWhr . A similar analysis of the cost given for the combined cycle system in Ref. 1 results in a COE of 48.80 mills/kWhr.

Table 1.1.7 - 30 Year Levelized Cost of Electricity

$$30 \text{ yr. Levelized COE} = \frac{(LCC + LFOM) \times ((1000 \text{ mills/\$})}{CF \times 8760 \text{ hr/yr.}} + LVOM + LFU$$

Where

LCC = Levelized Carrying Charges	(0.165 x TCR)	$/kW-yr
LFOM = Levelized Fixed O&M	(1.613 x FOM)	$/kW-yr
CF = Capacity Factor		
LVOM = Levelized Variable O&M*	(1.613 x VOM)	mills/kWhr
LFU = Levelized Fuel	(1.613 x W_f)	mills/kWhr
W_f = First Year Fuel Cost		

TCR = TPI + Start up + Initial Inventory + Initial Chemicals
= 133,042 +6850 + 10605 +205 = 150,747
= $511/kW

* Includes Consumables

$$30 \text{ yr. Levelized COE} = \frac{(LCC + LFOM) \times ((1000 \text{ mills/\$})}{CF \times 8760 \text{ hr/yr.}} + LVOM + LFU$$

$$= \frac{(84.31 + 18.93)}{CF \times 8.760} + 1.23 + 24.61$$

= 40.57 mills/kWhr @ $2.75/million Btu fuel and .8 CF

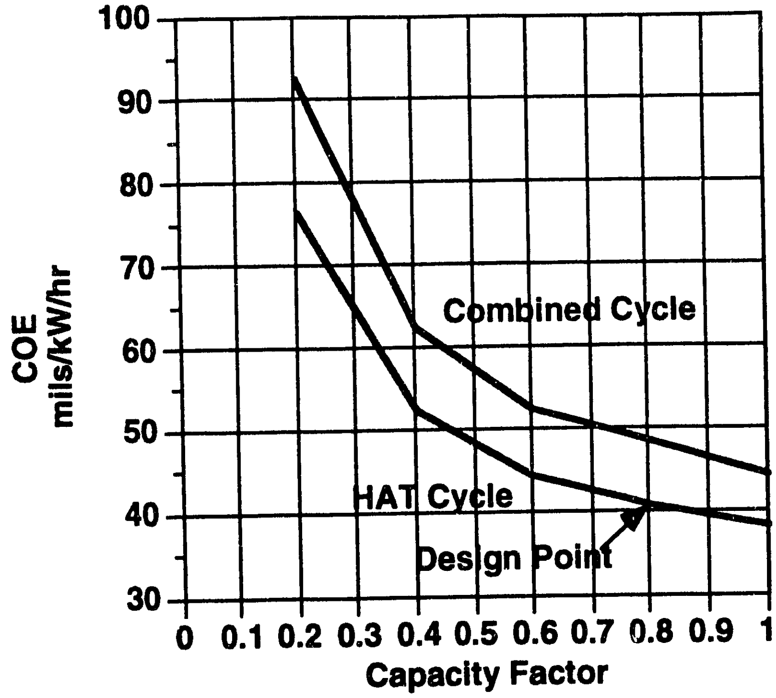

Fig. 1.1.11 - Cost of Electricity vs. Capacity Factor

1.1.4 - Commonalty of NGHAT and IGHAT Equipment

An aerodynamic analysis showed that the gas turbine compressor, combustion and control systems could be derived from existing hardware with the following adaptations:

- LPC - The first (8) set of stages of a large heavy frame combustion turbine; compressor clipped if necessary to provide approximately 625 lb/sec corrected flow, or, add 1 or 2 stages with case to the front of the FT4000 aeroderivative IGHAT low pressure compressor to increase the air flow to 625 lb/sec. These stages could be cantilevered over the bearings pending structural review to have a common bearing/rotor system and removed for the syngas engines.

- HPC - Removal of the last two stages of the IGHAT HPC to provide the proper airflow and pressure ratio for this cycle. "Dummy" disks without blades on the last two stages of the compressor may provide this characteristic while preserving the rotor system.

- HPC Discharge Diffuser/Collector System - Removal of the last 2 stages of the PW4000 compressor increases HPC exit flow parameter by over 25% which is necessary to match the turbine inlet flow conditions of the IGHAT design. The HPC diffuser/collector system which delivers flow to the aftercooler/ saturator would be opened up to prevent excessive losses. This same system would be used for both designs which would lower the IGHAT system losses.

• Combustor System - The fuel nozzles would be redesigned for the change in natural gas and syngas fuel flow and air flow to balance the ratio in the lean burn dry low-NOx nozzle. The combustor case and annular burner would be common with the possibility of dilution cooling air changes. The external plumbing and case penetration would differ. Cases can still be common by using oversized bosses.

• Turbine Cooling - Common turbine air foil castings and possibly common finished airfoils are expected for both IGHAT and natural gas designs. The heat transfer effectiveness of the gas path fluid decreases with less moisture which requires further detailed analysis to determine the extent of cooling commonalty.

• Control System - The electronic fuel control for the gas turbine would be common for the IGHAT and natural gas design. Programmable logic software changes would be required.

• Fuel System - A fuel plate is normally designed which contains the fuel system valves and hardware for engine control. The plate would be changed out to handle the higher volume syngas fuel in converting from natural gas fuel.

This second (common turbine) approach provides the maximum output for a NGHAT cycle. Compressor system and part of the air-collection/delivery system are significantly modified while expander components are common for the natural gas fueled and the syngas based HAT machines. Conversion from natural gas to IGHAT application would require removal of some of the forward LPC stages, addition of the last two stages in the HPC, and changes in the fuel nozzle and fuel system.

1.2 - Description of Natural Gas-Fired Reference System

The following sections describe the major features of the NGHAT turbine and system. Also included are a summary of the performance and cost of electricity, a discussion of maintainability, a preliminary program plan and schedule with estimated costs, identification of major enabling technologies, discussion of a commercialization plan, and an identification of the potential market.

1.2.1 - Description of NGHAT System

For convenience, the HAT cycle schematic Fig.1.1.1 is repeated here. The system operation is as follows. Filtered air is first compressed in the low-pressure compressor and then enters an intercooler where the majority of the compression energy is recovered for air saturation by the air saturator circulating water and makeup water. The cooled compressed air is further compressed in the high pressure compressor, cooled in an aftercooler with air saturator circulating water and then sent to the air saturator. The air is contacted over packing with water heated by the various heat sources in the cycle. The humid air leaving the saturator is preheated in the gas turbine exhaust and fed to the combustor. Natural gas is preheated with hot water before being sent to the combustor. The hot gas expands through the high pressure turbine which drives the high pressure compressor on the same shaft. The gas leaving the high pressure turbine is further expanded to near atmospheric pressure in the low pressure turbine which drives the low pressure compressor and the electric generator connected on a separate shaft. The turbine unit, therefore, consists of a high pressure shaft and a low pressure shaft contained in separate cases.

With the exception of the advanced gas turbine, all other equipment is either commercially available or, in the case of the recuperator, consist of elements that are commercial available. The recuperator would resemble the heat recovery steam generator (HRSG) portion of a combined cycle as the duties are similar. The HAT operating conditions are well within the demonstrated operating conditions of HRSG's. The first two rows of tubes would be 1.25% chrome steel with the remaining tube banks, carbon steel. The advanced gas turbine is described below.

The design options for major turbine flowpath elements are discussed, as are the mechanical design and material selection alternatives. Because the technology base for the HAT Cycle System is continually being reviewed and incremental changes made, the discussions are a "snap shot" of the current status. Other major NGHAT system components are also briefly discussed.

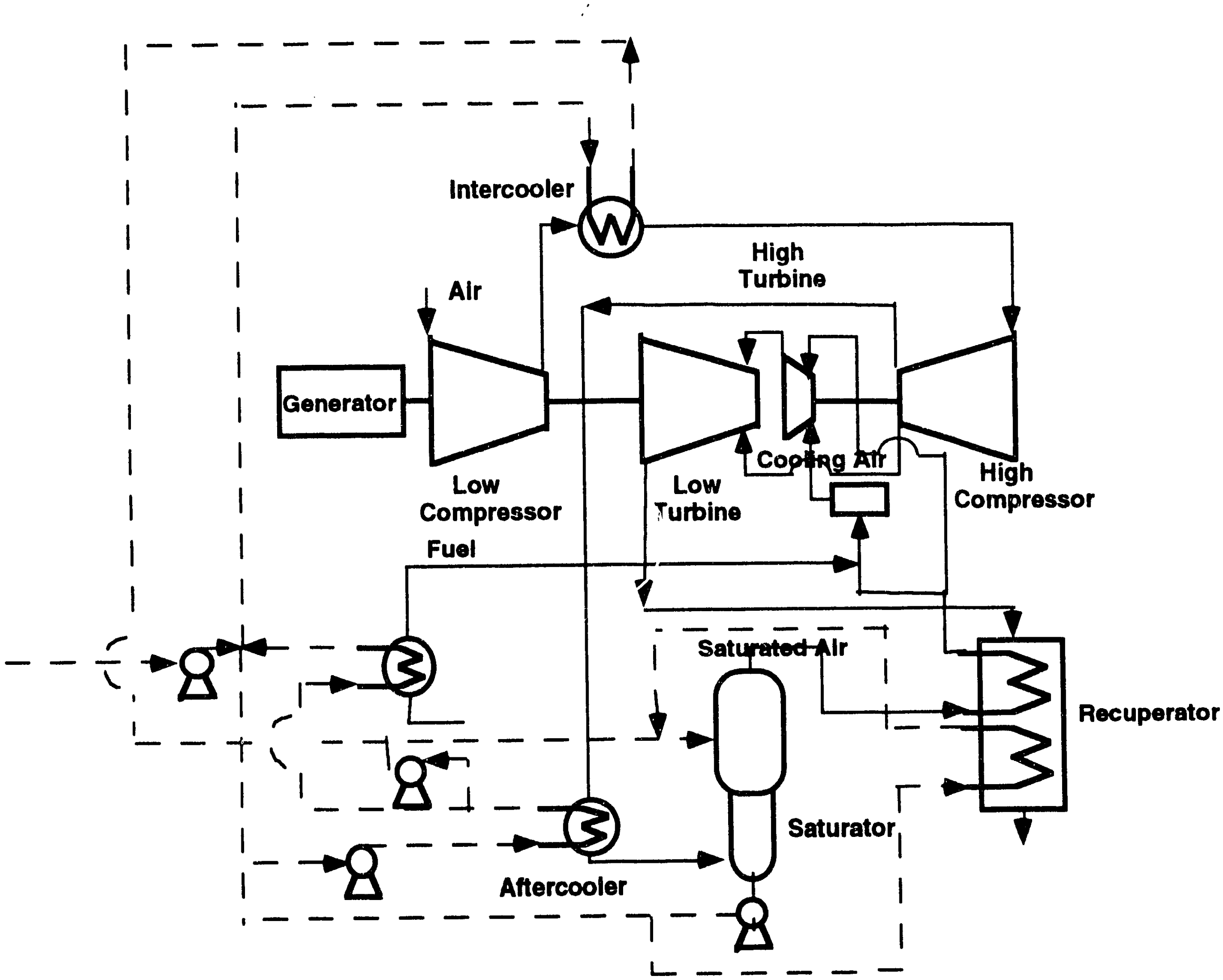

Fig. 1.1.1 - Natural Gas-Fired HAT Cycle

1.2.1.1 - Gas Turbine Flow Path

Low Pressure Compressor

Effort on the low pressure compressor concentrated on identifying a 60 Hz ,"low cost" design. Three options were available:

- Maximize commonalty with PW4000;
- Clean sheet design;
- Adapt industrial compressor.

One option was to retain hardware from the PW4000 LPC in an effort to keep development costs down. The 5-stage 4000 LPC was selected as a base. The design ground rules were to add to this compressor such that it ran at or near its design point but at a physical speed of 3600 RPM. This would give good performance while using Bill of Material parts. Based on the LPC design corrected speed and flow of 3131 RPM and 229 lb/sec respectively, an inlet flow and additional pressure ratio were determined. The result for the common compressor scenario at ISO plus 31 F (the Ref. 1 design point for HAT applications), was a compression system flowing 477 lb/sec with a pressure ratio of 4.94 at a physical speed of 3600 RPM, while for the common turbine the flow would be 591 lb/sec with a pressure ratio of 6.92.

The pressure ratio required by the 'new' front end part is 2.42 for the common compressor case and 3.39 for the common turbine. It is desirable to do this job in as few stages as possible while maintaining a stall margin of at least 15%. It was also desired to maintain a 'goal' efficiency of at least 87.3%, representative of aero-derivative industrial engines. Designs with 3 or 4 stages were chosen with relatively low aspect ratio vanes and blades (average AR = 1.3) with a flow path that went slightly out from the 'inlet' of the existing hardware.

For the same compressor do both 50 and 60 Hz generation, four variable stages would be required. Further studies, however, concluded that using the same hardware for both of these requirements would not be feasible. Therefore, 4 stages of variable geometry are not needed for the compressor to run only 60 Hz.

Another option was to design a 'clean sheet' compressor. To achieve the same flow and pressure ratio as the previous case while meeting the 'low cost' criteria, a constant inner diameter (CID)compressor was considered. The CID allows the same disk forging to be used more than once in the compressor. To meet the stall margin goal of at least 15%, a 4-stage, relatively low aspect ratio design was chosen. The compressor was also designed such that the axial distance between stages 2 and 3 is the same as that between stages 3 and 4 allowing the same spacer to be used here to reduce cost. This option may have favorable secondary effects. For example, it will allow a shorter, larger diameter LP shaft to be used, leading to more desirable critical speed characteristics. Also, a shorter diffuser can be used (with the same pressure loss) due to the lower exit Mach number.

The last option under consideration would use the first six stages of an industrial turbine compressor such as the V84.4 compressor, 'clipping' them to meet either the flow of 477 lb/sec or of 591 lb/sec (ISO + 31

F). The clipping results in a pressure ratio of 4.86. This is a very complex analysis; while preliminary results indicate this is possible, it is difficult to assess the effects on efficiency and stall margin without significantly more detailed analysis.

Also requiring more analysis are the various commonalty issues for the HAT LPC which include commonalty between 50 and 60 Hz as well as commonalty between the PW4000 LPC, the FT4000 HAT LPC, and the FT4000 Simple Cycle LPC. Studies continue to be conducted to assess the pros and cons of invoking various degrees of commonalty.

High Pressure Compressor

The HAT Cycle was configured such that the HPC, a high-cost development item, would be as close to an 'off-the-shelf' component as possible. At this time, the HAT Cycle will start with the same compressor to be used in a growth version of the PW4000 engine family. Its design point will be at a corrected flow of 107.1 lb/sec (97 lb/sec constant turbine) at a pressure ratio of 8.40. An audit level surge line analysis indicates the compressor has a surge margin of 26% at this operating condition. Some modification may be necessary to the secondary flow system associated with the HPC due to the relatively large amounts of cooling air required by the turbines.

Combustor

Early work on the combustor indicated that the silo type burner would be unsuitable because of durability reasons. Thus, an annular burner was selected. More efficient air delivery systems are being considered such as an integral manifold/combustor shell which could be a relatively cheap design as well as a lower pressure loss design than the system currently designated for the FT4000 simple cycle. Because of the large amount of water vapor, the combustion time will be longer than normal to assure CO and HC burnout. This will require a larger combustor than usual and may lead to a tilted combustor to minimize impact on shaft length.

High Pressure Turbine

The High Pressure Turbine (HPT) is a 'clean sheet' design. The flow path was chosen considering allowable rim speed, stage loading, and flow path integration with the LPT. The fact that the air is intercooled before it enters the HPC combined with the flexibility of a 'clean sheet' design (allowing the flow path to be pushed out as much as required) allows the use of a one-stage HPT than the two-stage HPT required to drive the PW4000 HPC . This one-stage design has a rim speed of 1254 ft/sec with rim velocity ratio of 0.58. Preliminary cooling flows estimates are based on using humid air from the regenerator diffuser outlet to cool the vane and compressor extraction air, at a low temperature because of the intercooler, to cool the blade. With 15% turbine cooling air (TCA) the meanline predicted efficiency is 89.7%. Cooling and leakage air for the HPT is supplied from the HPC exit.

Low Pressure Turbine

The LPT is also a 'clean sheet' design. To keep downstream losses low, it is desirable to keep the exit Mach number below .40 and the exit swirl less than I5°. The configuration arrived at was a 6-stage LPT, with an average rim velocity ratio of .52 with a relatively low stress. LPT leakages are based on values typical of large frame-type machines and airfoil cooling levels were estimated based on cooling air extraction from the compressor. With 19.7% TCA the meanline predicted efficiency is 93.4%. However, a value of 92.1% was used for the purposes of calculating preliminary performance, reflecting some degree of uncertainty in predicting the performance of a component of this nature.

The flow path moves outward as air moves aft in the LPT. This allows the front of the LPT to be integrated with the HPT, while giving more work capability to the rear stages. A 'hot strut' is used between the HPT and LPT as it was considered impractical to have a close coupled turbine system.

The first blade was assumed to be unshrouded because it is cooled. The next three were assumed to be shrouded. The last two blades were assumed to be unshrouded due to high AN^2, a practice used in the latest high efficiency frame machines. The blade tips are conical to provide the highest efficiency turbine for the desired exit conditions. More detailed trade studies are required to consider turbine efficiency, clearance control effectiveness, and cost to determine the viability of the conical tip flow path. A "stair step" flow path will result in an efficiency reduction, but there is margin included in the turbine efficiency. The inner diameter of the last two stages is constant, allowing a common forging for both disks.

1.2.1.2 - Gas Turbine Mechanical Design

General Arrangement

The turbine is arranged in 2 modules (see Fig. 1.1.1): a 4 stage (or 8 stage, if the PW4000 commonalty is retained) low pressure compressor

(LPC) and a 6 stage low pressure turbine (LPT) on one shaft, and a single stage high pressure turbine (HPT) and a high pressure compressor (HPC) on the other shaft. The shafts are not concentric, which is atypical for a 2 shaft engine. The shafts are arranged in series to best accommodate the air path of the HAT cycle.

The air enters the LPC via a plenum chamber and progresses to a diffuser, and exits at a low velocity and enters a volute casing. The air exits the volute casing and is piped to an intercooler. The air returns from the Intercooler via a volute casing. The air proceeds through the HPC, is diffused to a lower velocity, and enters the HPC exit volute casing. The air exits the volute casing and is piped to an aftercooler, then through a saturator and finally through a recuperator where it is heated by turbine exhaust gases before returning to the turbine. The air reenters the turbine through an air return manifold. The air is distributed around the outer combustor casing via a series of long radius elbows from the combustor air return manifold. Elbows are used to reduce the effect of the differential and radial thermal growth between the combustor air return manifold and the combustor outer casing.

The HAT is arranged such that both ends of the machine are "cold"— i.e., the LPC entrance is at one end and the intercooled LPC discharge air enters the other end. Thus, the driving shaft connected to the generator is at the LPC end of the machine. The starting motor, turning gear, etc. can be attached to the HP rotor at the HPC end of the machine.

Bearings And Bearing Housings

The LP rotor thrust bearing is located at the No. 1 bearing location. The thrust bearing is a tilting pad "Kingsbury" type. The thrust bearing is 22" diameter and is capable of carrying approximately 200,000 lb thrust in either direction (in the event of surge or other thrust reversal) . The thrust, using 10th-stage air as a thrust balance in the LP end, is 36,185 Ib aft for the 8-stage LPC configuration and 125, 959 Ib forward for the 4- stage LPC configuration.

The LP set is horizontally split for its entire length to the combustor casing, including the bearing housings. The No. 1 and No. 2 bearings are split journal bearings required to support the heavy weight LP rotor.

Two conceptual designs have been completed. The first design is a "light-weight" design following the PW 4000 type construction for the rotors and stators. One of the features is a "piggy-back" No. 2 bearing housing. This housing contains the split No. 2 journal bearing for the LP rotor and a roller bearing to support the forward end of the HP rotor. This LP "piggy-back" design reduces the number of "hot" section bearing housings required. This housing is located in a very hostel atmosphere with turbine gas path temperature radiating to it, as well as being surrounded by 471 F (+) air. If an additional housing is used, it will be subjected to radiation from the combustor casing and surrounded by compressor discharge air leakage from HP turbine cooling air (604 F). The aft end (the LP end) of the HP rotor is supported by a ball bearing which also serves as the thrust bearing. This No. 4 bearing is located in the intermediate casing. The intermediate case and the remainder of the HPC are existing PW4000 parts. Preliminary calculations indicate the PW4000 HPC casings can be used as is. Even with the high pressure ratios, the allowable stress, at the lower temperatures, is proportionately higher.

The separate No. 2 and No. 3 bearing housing design utilizes the No. 3 bearing and the majority of the housing (except attachment to the combustor casings) on the PW 4000. Thus, both the No. 3 and No. 4 bearings would be from that engine.

The output shaft transmits approximately 295 MW (395,500 HP.) under max. conditions (59 F inlet temperature) which provides approximately 682,000 ft-lb torque. If a factor of 2 is used for a short circuit torque multiplier (customary for this size machine) the transient torque = 2,046,000 ft-lb. The smallest diameter on the LP rotor (12 in.) is at the No. 1 bearing. The shear stress is 16,179 psi at the bearing and 65,550 psi in the collar portion. For the projected material, PWA 733, those stress levels are acceptable (this material has a yield strength of 115 ksi at 500 F).

The LP rotor weight is 21,786 lb for the LP "light weight" rotor with the 8 stage titanium LPC . The rotor weight is 23,000 lb for the LPT "light weight" construction rotor. The estimated rotor weight is 30,150 lb for a frame industrial-type construction LPT attached to a 4-stage-type construction LPC .

Low Pressure Compressor

The original concept for the HAT intended to use as much of the PW 4000 as possible. With that precept, 3 stages were added to the existing 5 stage PW 4000 LPC. It required a variable inlet guide vane (VIGV) and 3 additional stages of variable geometry. Thus, the first stage of the 4000 LPC would require variable geometry. The remaining rotor blades and stator vanes and rotor could be the same hardware as the PW 4000. Construction for the new stages would be similar to the existing stages, i.e., titanium drum rotor construction.

It was later determined that the stator structure was being compromised by using the decreasing diameter in the latter stages of the LPC. Also, by increasing the diameter, the number of stages could be reduced to 4 new stages, which is basically the same number of stages to be developed for the 8 stage design. In addition, the parts count would be reduced by about a factor of 2.

Being a new design, the less expensive FT8 type construction and materials could be used. Blade weights (titanium blades) were estimated, using scaling criteria from current designs, and the disks were analyzed using the Autodisk computer programs. AS6415M (AIS14340) material was used for the disks, the same material used on the FT8. A minimum burst margin of 1.2 and 80% of the 0.2% yield strength at the respective temperatures were used for the allowables.

Low Pressure Turbine

Data typical of frame machine designs were scaled for the HAT disk analysis. The flow path has unshrouded 1st, 5th, and 6th stage blades and shrouded 2nd, 3rd, and 4th stage blades. Tip speeds are 1529 ft/sec for the 5th stage and 1680 ft/sec for the 6th stage.

The 1st, 3rd, 4th, 5th and 6th disks were analyzed using the Autodisk computer programs. The spacers were not included in the analysis since they are within the self-sustaining radius. The disks are of a modified PW 4000 construction, however, spacers are used rather than extension arms from the disks to reduce the thickness of the disk forging as much as possible.

The material used was PW 1085M (INCO 718) for the 1st, 2nd and 3rd LPT disks. It was necessary to use nickel for these stages due to the high absolute temperatures and 100,000 hr life requirements. Disk analysis was performed for the 1st and 3rd stage and interpolated for the 2nd stage based on blade pulls. LPT disks 4, 5, and 6 are made of PW 1029M (A286) material. The Hirth serration type construction is used for the attachments. Torque transmission calculations were used for sizing the serrations.

Compressor and Turbine Tip Clearances

The conical tips of the 5th and 6th stage of the LPT are a concern from a transient and perhaps a steady state tip clearance stand point. A stationary shroud device known as a "Constant Clearance Blade Tip Seal" is used, assuring that if an axial movement occurs, which would normally cause the radial tip clearance to increase, with this device a constant clearance would be maintained. This occurs due to an increase in clearance causing the pressure at the tip of the blade to decrease. When that occurs, the force due to the higher pressure upstream of the blade attempts to rotate the shroud which has an angled slot and pin arrangement, such that the shroud moves axially when it rotates, and the clearance decreases to obtain the initial clearance. This device is as yet untested, thus attempts should be made to select materials and cooling schemes to minimize the amount of axial movement of the rotor with reference to the casing.

The axial fixation point of casings is located at the air inlet end of the gas turbine. The rotor thrust bearing is also located at the inlet end, at the No. 1 bearing location. Thus, if the rotor and stator grew axially at the same rate, the relative clearance, due to axial movement, would be zero. Based on previous experience, this does not occur. A transient condition occurs during start-up where the rotor grows faster than the massive casings. Thus it is common to experience a rub during start-up transient conditions.

To enable use of less expensive materials, the outer casings of the turbine will be cooled. The rotor is also cooled. The casing thickness vs. rotor mass, as well as differences in thermal expansion coefficients, makes it difficult to make the radial and axial differential thermal growths minimal. A detailed heat transfer/cooling study will be required to define this.

Past experience with similar cooling schemes, casing and rotor masses and materials can be used to estimate starting points. For similarly constructed machines, minimum clearances occur between 10 and 15 minutes after start for a 15 minute load ramp. At that point, rotors achieve about 68% of their steady state temperatures while the stator casings, etc. reach only about 11%.

Unit Support System

The engine is supported a 3 locations:

(1) at the inlet casing (just below the horizontal flange),

(2) at the transition casing between the HPT and LPT, and
(3) at the volute casing at the HPC inlet from the intercooler.

The fixed unit support is at the inlet casing. This support point has a flat plate on either side of the casing designed to prevent axial movement from occurring, but will allow it to bend due to growth of the casing in the radial direction. There is a "gib" block at the bottom centerline which maintains the axial alignment of the machine at the LPC end. The gib block is also used to absorb shipping loads when shipping by rail. Shipping loads can be the most significant loads which the machine encounters (even greater than blade-out forces).

The 2nd support system is at the transition casing (between the HPT and LPT). This system is comprised of a "grasshopper" leg (a spherical joint at both ends). This type support provides only vertical support. The support legs are attached to the casing close to the machine centerline. There is a gib block at this location also. A pin can be inserted through the gib block such that the aft end of the LP set is supported while the HP set is removed when maintenance of that module is required.

The 3rd support is a set of flexible plates perpendicular to the axis of the machine. These support plates allow axial movement but provide vertical fixation. This set of plates are attached to the volute casing at the inlet to the HPC near the engine centerline.

Materials Of Construction

As indicated previously, the LP set is being considered as a heavy duty gas turbine. Also the entire engine must meet 100,000 hr life for non-gas path parts (disks, casings, etc.). The materials chosen were based on the least cost to operate under the pressure, temperature, and other loading considerations, encountered in an industrial atmosphere and operating conditions. Many of the materials are those customarily used in frame type gas turbines, as well as materials used in the FT8.

Much of the HPC has been assumed to be common to the PW 4000 HPC, thus the materials are common and not necessarily the least cost. However, if other materials such as AISI 403 stainless steel were used for LPC vanes, the cost of characterizing the material and new tooling could far out weigh the increased cost of titanium vanes.

Other materials which are common for industrial gas turbines but uncommon, due to weight considerations, to aircraft engine designers are cast iron and nodular iron. Cast iron may be used, due to the low pressures, for the inlet casing. Nodular Iron (ASTM A 536-60/40/18), also known internationally as Ductile Iron is a spheroidal-graphite cast iron. The spheroidal cast iron has about twice the tensile strength of flake cast iron, i.e.: 60 ksi tensile strength, 40 ksi yield strength, and 18% elongation. It has been found to be more thermally stable than cast carbon steel and as such is a good turbine/compressor casing material. As more detailed design is carried out, other materials may be identified as more suitable for the application; however, parts cost should be a primary criteria.

One part that requires a more thorough materials investigation is the transition gas path part between the HPT and the LPT. The transition casing is a major support casing which supports the aft end of the LP rotor, and transmits the loads across the gas path, to the unit supports. With sufficient cooling, Haynes alloy No. 230 may be used. The gas path temperature is a minimum of 2120 F. The gas path material needs to protect the struts, of Greek Ascaloy or Inconel 718, from the high temperature gas. Cooling air passes between the struts and the gas path liner, but is of insufficient amount to allow film cooling. Other materials were considered such as MA 956 and Columbium, however, MA 956 (and MA 6000) is/are difficult to work with (weld, etc.) and Columbium requires a coating which, if it were to be eroded or scratched off, would cause the materials to oxidize severely. A detailed heat transfer analysis may show the Haynes 230 acceptable since the 1st LPT vane cooling air can be routed through the cooling passages prior to entering the vane, thus cooling the transition piece sufficiently.

1.2.2 - HAT Cycle System Performance

The performance of the HAT cycle system was described in Section 1.1.1. The operating conditions for the balance of plant are given in Table 1.2.1 and the performance is summarized in Table 1.2.2. A flow schematic is shown in Fig. 1.2.1 and the state points are given in Table 1.2.3. The favorable part-load performance of the HAT cycle system is shown in Fig. 1.2.2. Here it can be seen that the heat rate is essentially flat over a wide range of load. This means the HAT cycle can be used to follow load without a significant penalty in heat rate.

Table 1.2.1 - Balance of Plant Characteristics for NGHAT

HEAT RECOVERY UNIT	
Heat Loss:	
GT.. Exhaust Side	1%
Preheated Humid Air Temperature Drop	5 F
Pressure Drop:	
GT.. Exhaust Side	12 in H_2O
Recuperator (Humid Air & Piping)	2.5%
Economizer (water side)	20 psi
Pinch temperatures:	
Economizer (cold side)	25 F
Effectiveness:	
Humid Air Recuperator	90%
Economizer	90%
Transition Duct from GT.. to Heat Recovery Unit:	
Heat Loss (Temperature Drop)	2 F
SATURATOR	
Blowdown (Percentage of Evaporated Water)	1.0%
Diameter	14 ft
Height of Packing	40 ft
Pinch Temp. Between Entering Gas and Exiting Water (at Bottom of Column)	20 F
Pressure Drop (Saturator & Piping)	1%
HEAT EXCHANGERS	
Fuel Gas Heater:	
Hot Side Pinch	20 F
Effectiveness	90%
Pressure Drop (both sides)	5 psi
Intercoolers I & II:	
Hot Side Pinch	20 F
Effectiveness	90%

Pressure Drop (water side)	5 psi
Pressure Drop (gas side)	see below

Intercooler III

Cold Side Pinch	20 F
Effectiveness	90%
Pressure Drop (water side)	5 psi
Pressure Drop (gas side - intercoolers I, II, and III total and piping)	

Aftercooler

Cold Side Pinch	20 F
Effectiveness	90%
Pressure Drop (water side)	5 psi
Pressure Drop (gas side & piping)	1 %

PUMPS

Pressure at Pump Discharge over Saturator Pressure:

Saturator Bottoms Circulation	+100 psi
De mineralized Makeup Water	+40 psi

DEHUMIDIFIER

Gas Side Pressure Drop	4 in H_2O
Water Recovery	80%

Table 1.2.2 - HAT Cycle Performance
(ISO Conditions)

Inlet Flow, lb/sec.	627
LPC Pressure Ratio	7.34
LPC Corrected Flow, lb/sec.	636
HPC Pressure Ratio	6.01
HPC Corrected Flow, lb/sec.	97
Rotor Inlet Temperature, F	2770
Overall Pressure Ratio	44.1
Water/Air Ratio in Saturator Exit, lb/lb	0.23
Fuel Flow, lb/sec.*	21.3
Gas Flow (Turb exit), lb/sec.	761
Stack Temperature, F	238
Gross Output at Generator Terminals, MW	298
Net Plant Output, MW	295
Net Plant Heat Rate, Btu/kWhr (LHV)	5550
Net Plant Efficiency, % (LHV)	61.5

* 20873 Btu/lb at 491 F

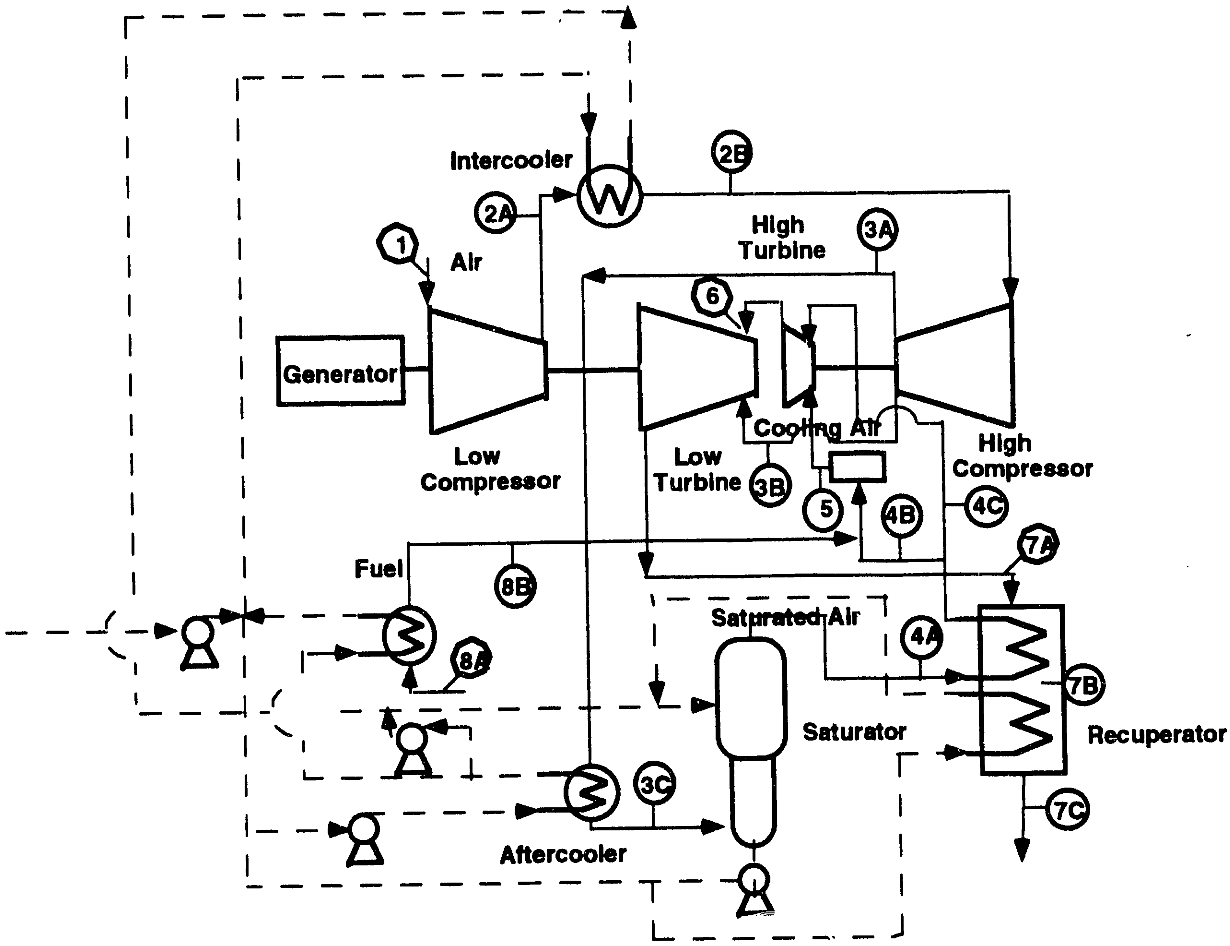

Fig. 1.2.1 - Flow Streams for Natural Gas-Fired HAT Cycle

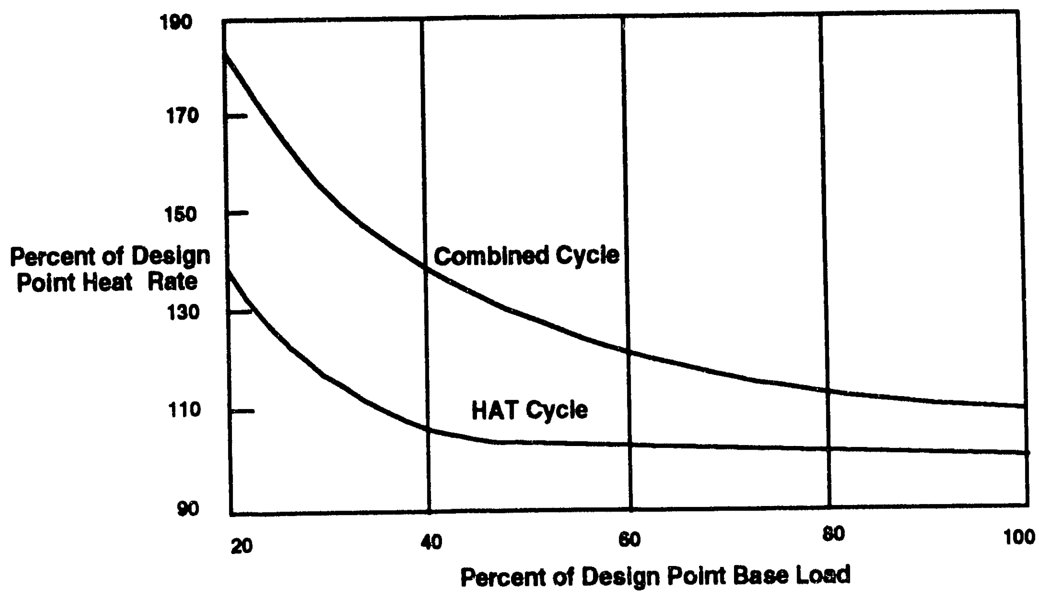

Fig. 1.2.2 - Partload Efficiency Comparison

Table 1.2.3 - State Points for NGHAT Cycle

Station Number	Pressure psi	Temperature F	Mass Flow lb/sec
1	14.54	59	627
2A	102.6	497	627
2B	98.2	85	625.3
3A	582.4	493	507
3B			118*
3C	573.3	240	5 07
4A	561.6	352	622.4
4B	544	917	567
4C	550	917	55.5
5	532	2920	587
6	342	2380	690.5
7A	15.6	980	761
7B	14.7	537	761
7C	14.7	238	761
8A	549	137	21.3
8B	549	491	21.3

* Sum of all cooling air.

1.2.3. - HAT Cycle Capital and Operating Costs

The cost of electricity calculations in Section 1.1.3 identified the operating and maintenance costs and the capital costs of the HAT cycle system. These are summarized in Table 1.2.4.

Table 1.2.4 - Summary of HAT Cycle Costs
(000's)

Annual Fixed Operating Costs	$4181.1
Annual Variable Operating Costs	$1899.5
Total	$6080.6
Total Plant Investment (3 yr. construction)	$133,042

1.2.4. - Maintainability

Rather than a projection of RAM for the nth plant, a complex calculation, a discussion of system maintainability is given. An outline of UTC's RAM philosophy is contained in Appendix D.

The maintenance philosophy on this machine is to think of the LP set as a "heavy frame" gas turbine and the HP set as a light-weight aircraft derivative engine.

The LP set is split horizontally, as well as vertically, to accommodate maintenance of the heavy weight components such as the LP rotor (21,000 - 30,000 lb). The bearing housings have an upper and lower half with split journal bearings to accommodate rotor removal. The bearing housings are separate castings than the casings to allow for easier machining of the bores, etc. The housings are supported from their lower halves by ears integral with the horizontal split on each side of the housing. Straps (or channels) are used to bolt the housing to the supporting casing which allows relative radial growth of the housing with reference to the supporting casing. Each bearing has a gib key to engage in a gib block which is integral with the supporting casing. The gib key is used to maintain alignment of the bearings and seals with the casing centerline, as well as resist rotation (especially for the "Kingsbury" type thrust in the No. 1 bearing housing).

Using the split casing concept allows rotating in many of the parts rather than necessitating "stacking" rotor-stator-rotor, etc. The rotor is lowered into the lower half of the casings and the seals etc., are rotated into the lower half around the rotor (to keep the seals from being damaged during assembly.)

Another casing which is split horizontally is the HPC discharge volute. The spacer casing between the combustor outer casing and the HPT volute casing is also split horizontally. In both instances, the reason for splitting the volute casing is to remove the upper half, then allow lowering the lower half to make axial room to unbolt the outer combustor casing and move it aft to allow inspection and/or repair of the float wall panels on the annular combustor, without complete disassembly of the HP set. The upper half of the spacer casing will need to be removed to obtain access to disconnect the HPT, turbine on-board injection (TOBI), cooling system air supply line (tube coupling) and the oil drain/supply line to/from the No. 3 bearing housing. Also, a temporary support would be needed at the forward end of the HPC casing (rear diffuser) to keep the stators in radial alignment. An alternative arrangement, would require the No. 3 bearing housing to remain assembled to the forward outer combustor casings to support the rotor.

1.2.5 - Preliminary Development Program for Proposed System

The four-phase ATS Program schedule identified by DOE is shown in Fig. 1.2.4. The first phase is the scoping study which identifies a reference ATS for each contractor. Phase 2 will allow comparison of the selected reference system with other alternatives, further define reference system concepts and design features, and initiate critical component testing. Phase 3 will provide for testing of large-scale components and subsystems, and the overall system would be demonstrated in Phase 4. Also shown in Fig. 1.2.4 is the proposed UTC schedule for the NGHAT development program. In the following section, the costs and schedules developed by the UTC HAT Cycle team are described and their inter-relationship to the ATS Program identified.

1.2.5.1 - Projected Schedule and Estimated Costs

The overali UTC development plans for the HAT Cycle identify four different engine configurations:

- 60 Hz NGHAT
- 60 Hz IGHAT
- 50 HZ NGHAT
- 50 Hz IGHAT

All of these need to be developed to assure that the marketing requirements (see Section 1.2.8) for fuel flexibility and global competitiveness are met.

Preliminary Schedule and Costs

A preliminary program schedule for the four configurations is given in Fig. 1.2.3. This schedule is predicated on the availability of the required funding and human and physical resources. At this time, only the NGHAT schedule can be realistic from a time line basis, again given the required resources. In Figs. 1.2.3 and 4, it can be seen that the NGHAT demonstration plant using FT4000 technology is projected to begin installation in 1997 with the plant in operation in 1998, some 18 to 24 months ahead of the DOE suggested schedule. This is because of the large existing technology base in the PW4000/FT4000 and the continuing contribution of on-going UTC R&D efforts in commercial and military aircraft engines.

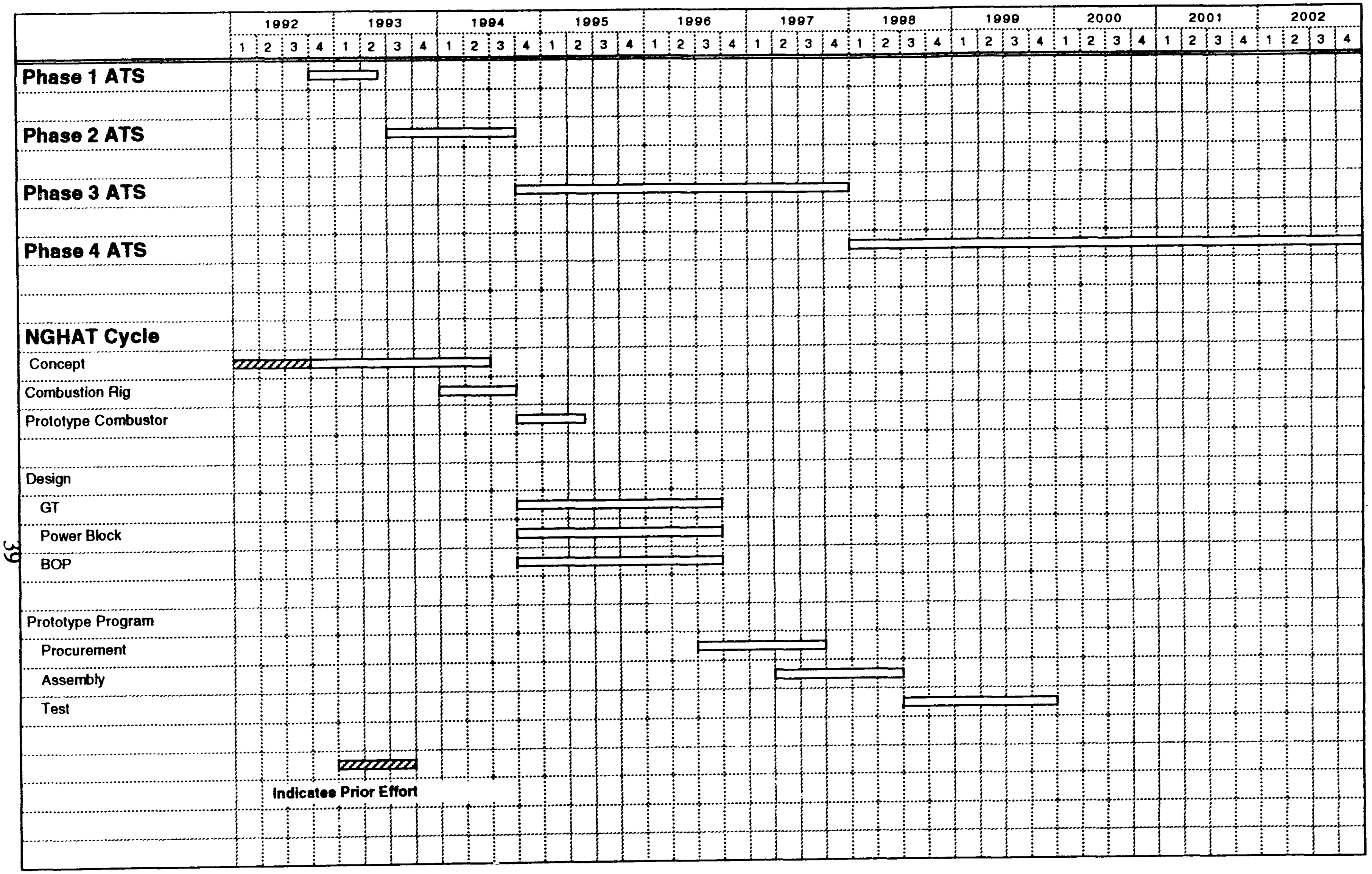

Fig 1.2.3 - Proposed Schedule for DOE ATS Program

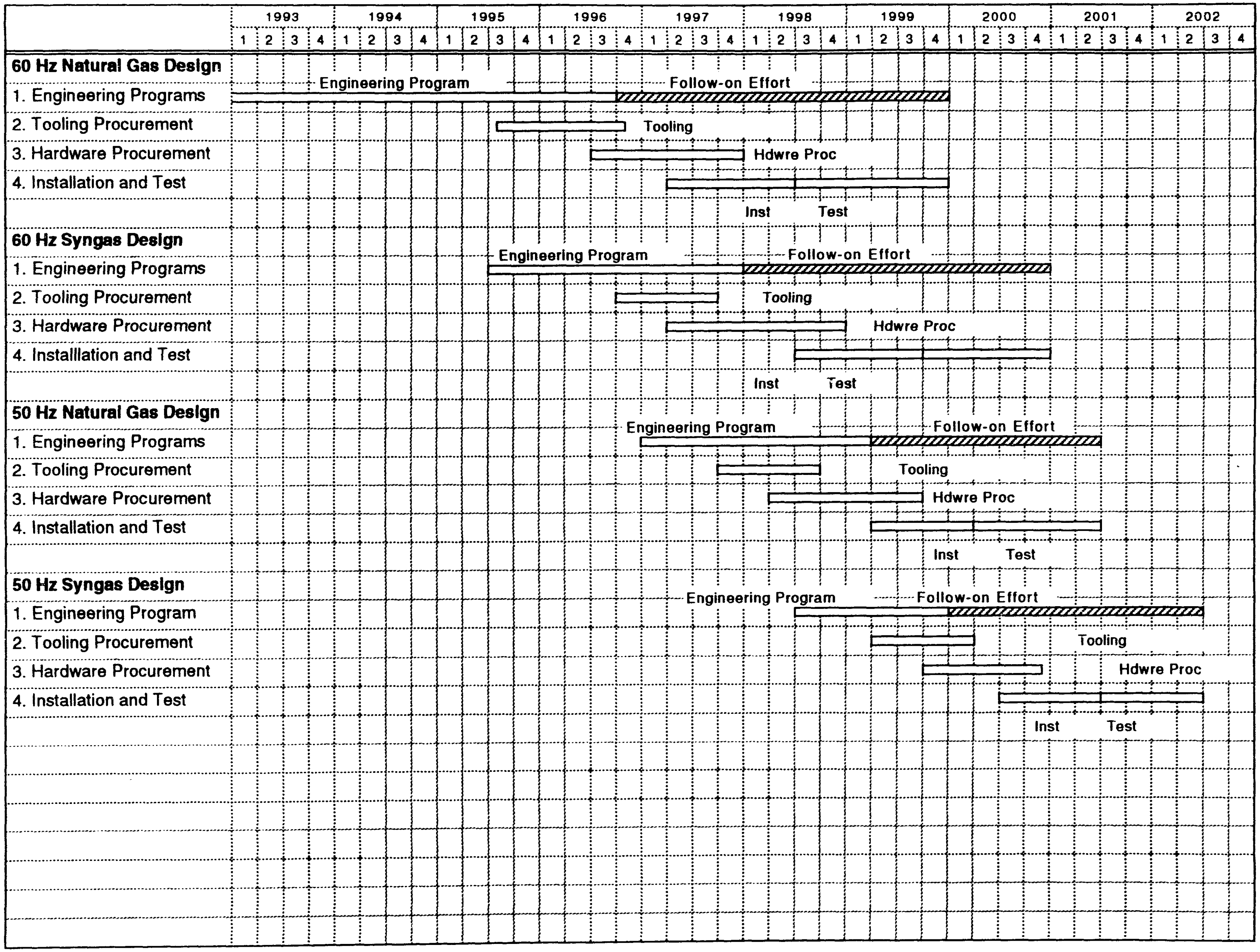

Fig. 1.2.4 - FT4000 HAT Cycle Preliminary Program Schedule

The estimated costs (1991 $) for the programs in Fig. 1.2.4 are given in Table 1.2.5 for the period 1993 through initiation of NGHAT testing in 1998. Engineering, component and package development, and rig test have an estimated cost of approximately $144 million (91 $'s). Greater detail for a portion of the costs making up this area are given in Table 1.2.6. When costs for turbine testing and refurbishment (estimated $23.8 million), assembly and tooling (estimated $25.7 million), and unique first plant costs, including contingency, (estimated $39 million) are added, the projected cost for the NGHAT program becomes approximately $232 million.

Table 1.2.5- Preliminary Estimates of FT4000 HAT Cycle Development Program
(To Initiation of Test of NGHAT)

	1993	1994	1995	1996	1997	1998	Total
1 60 Hz Natural Gas	13.2	36.6	40.6	28.7	11.0	8.5	138.6
Package Development	0.2	1.1	1.2	.8	.9	1.0	5.3
Total 1991 $	13.4	37.7	41.8	29.5	11.9	9.5	143.8
2 60 Hz Syngas Option			20.9	42.8	21.4	8.5	93.6
Package Development			0.4	2.4	2.6	1.1	6.5
Total 1991 $			21.3	45.2	24.	9.6	100.1

Note: These estimates do not include prototype parts/assembly/tooling/refurbishment/test support.

It should be noted that the prototype, after refurbishment, can be operated as a commercial plant. The value of this plant will reduce the net cost of the overall program. Details of this plan will be developed in Phases 2 and 3.

1.2.5.2 - Cost Sharing

Phase 1 of the ATS program was approximately $100,000 with an industry cost share of approximately $500C (Program cost of approximately $95,000). Future phase will have increasing amounts and percentages of cost sharing, e.g., UTC has submitted a proposal for Phase

Table 1.2.6 - Detailed Cost Breakdown for Selected Development Tasks

TASK	COMPONENT AREA	COST $ (M 1991)	TASK SCOPE
	COMBUSTOR (cont.)		• INCLUDES DESIGN/ANALYTICAL FOLLOW-UP DURING HARDWARE MANUFACTURE AND TEST PHASE.
			• DESIGN AND ANALYTICAL EFFORT FOR FIVE DIFFUSER/COMBUSTOR RIGS.
			• DESIGN/ANALYTICAL FOLLOW-UP DURING TESTING OF FIVE DIFFUSER /COMBUSTOR RIGS.
	HPT	11.1	• ANALYTICAL, DESIGN AND DRAFT ING OF A SINGLE BLADE/DISK AND VANE STAGE.
			• MATERIAL SELECTION TO OBTAIN CREEP, OXIDATION AND CORROSION LIFE GOALS OF 15K HRS AND 9K HRS,RESPECTIVELY.
			• INCLUDES DESIGN/ANALYTICAL FOLLOW-UP DURING PROTO-TYPE ENGINE TESTING.
	LPT	19.8	• ANALYTICAL, DESIGN AND DRAFT ING FOR SIX STAGE TURBINE.
			• MATERIALS CAPABILITY TO MEET CORROSION AND OXIDATION LIFE GOALS.
			• DESIGN COMPATIBILITY WITH MANUFACTURING CAPABILITY.
			• 1ST STAGE BLADE AND VANE COOLING REQUIRED.
			• DESIGN "HOT STRUT" TRANSITION CASE.
			• "HOT STRUT" OIL COKING ANALY-SIS
			• ASSESS AXIAL ROTOR GROWTH EFFECT ON BLADE TIP CLEARANCE AND BLADE/VANE SPACING.

Table 1.2.6 - Detailed Cost Breakdown
for Selected Development Tasks (cont.)

TASK	COMPONENT AREA	COST $ (M 1991)	TASK SCOPE
			• TURBINE CASE COOLING REQUIRE MENTS.
			• LOW ROTOR SHAFT DESIGN. INCLUDES ASSEMBLY AND MAIN TAINABILITY ITEMS.
			• EXHAUST CASE AND ASSOCIATED DUCTING.
			• INCLUDES DESIGN AND ANALYTI CAL FOLLOW-UP DURING PROTO-TYPE ENGINE TESTING.
	EXTERNALS	6.1	• ANALYTICAL, DESIGN, & DRAFT ING OF ALL FUEL, OIL,AND AIR PLUMBING
			• INCLUDES PROVISION FOR DUEL FUEL (GAS WITH OIL BACKUP) MANIFOLDS
			• TWO SEPARATE OIL SYSTEMS (FOR JOURNAL AND AIRCRAFT BEARINGS
			• BLEED AND COOLING AIR VALVING AND DUCTING.
			• ELECTRICAL AND CONDITION MONITORING WIRE HARNESSES.
	INLET/EXIT SCROLLS	3.0	• ANALYTICAL, DESIGN, & DRAFT ING OF LOW LOSS/CIRCUMFEREN TIAL DISTORTION COMPONENT SCROLLS.
	BEARINGS & SEALS	6.5	• ANALYTICAL, DESIGN, & DRAFT ING OF ALL BEARINGS, SEALS AND SCAVENGE COMPT'S.
			•INCLUDES ANALYSIS FOR HEAT REJECTION, SCAVENGE SIZING AND FLOW REQUIREMENTS.
			• REQUIRES ANALYSIS FOR AIR CRAFT TYPE AND JOURNAL TYPE BEARING COMPARTMENTS.

2 with an increased cost share. In future phases, UTC expects to bring in partners to assist with cost share in the development program and in marketing the commercialization of the technology.

1.2.5.3 - Cost/Benefits

The investment of nearly $232 million to develop and operate a prototype NGHAT must be justified by the potential benefits. Given that the installed cost of the "nth" unit NGHAT would be $511/kW, (described in Section 1.1.3), a cost difference of nearly $1000/kW would be realized vis a' vis a fossil steam plant and over $125/kW vis a' vis a combined cycle plant. When combined with the efficiency advantage, the savings to the electric power consumer could be over 20% compared to the best current technology combined cycle. For example, if only the 1.035 x 1011 kWhr of energy projected for gas-fired combined cycle generation in 2001 (Section 1.2.8) were to be generated by the NGHAT at a conservative 8 mills/kWhr savings over the combined cycle systems, the annual return to the consumer would be more than $\$8 \times 10^8$, several times the development cost. Penetration further into the utility base-load market would increase the benefits.

The social benefits resulting from reduced emissions of Greenhouse gases and CO and HC are difficult to quantify. However, there would be advantages in siting and dispatch because the NGHAT could be located closer to load centers. Also, emission credits could be sold or traded within the locale, creating further benefits for the operator.

1.2.6 - Intermediate Development Needs

The majority of the HAT cycle system is based on technology that is commercial or technology that is being used in state of the art aircraft engines and slated for future use in aero-derivative industrial engines. There are two areas that remain to be fully developed to assure the meeting of DOE ATS goals:

- development of advanced aircraft engine cooling techniques for the HAT engine;

- development of a combustor that will burn a mixture of water vapor and natural gas at high efficiency and low emissions.

The cooling is discussed in Appendix B. While Appendix C addresses emissions, it does not discuss the other combustor concerns.

The major problem with the HAT combustor is the lack of empirical data on burning of natural gas with water vapor fractions approaching 30% of air flow. This is compounded by the eventual requirement for ultra-high turbine inlet temperatures (>2700 F). Air management is a major concern in all turbines as air is needed for combustion, for cooling the turbine, and for cooling the combustor walls. As turbine temperatures increase, all three areas want more air. Also, the amount of air needed for the combustion process increases as water vapor fraction increases.

A major reason for using the annular burner approach is to reduce the "pattern factor" effect, i.e., the occurrence of uneven radial and temperature variations which affect turbine cooling requirements (see Appendix B). It is anticipated that the large amount of water vapor could actually help alleviate the pattern factor effect because of the need for larger combustion volumes (times) resulting in both better mixing and better conduction of heat. In addition, the presence of water vapor will reduce the radiation of heat to the combustor walls, the main heat transfer mechanism.

All of the questions and potential advantages can be answered only by testing of HAT combustors and reiterating on combustor design. The first series of these tests is called for by UTC as part of Phase 2 of the ATS Program.

1.2.7 - NGHAT Commercialization

Successful commercialization will be possible only if the HAT does indeed demonstrate the potential to meet the goals for performance, emissions, and costs summarized in preceding sections. To assure that customer requirements in these areas are met, UTC has been working closely with a number of US. utilities as part of the early, privately funded HAT study. In addition, other potential investors, both US. and foreign, have been contacted to provide global insight into design requirements as well as participate in the program as associates or partners. As the HAT cycle development continues, UTC will work with its utility and manufacturing partners to assure global competitiveness and market success.

1.2.8 - Market Opportunities

All markets for industrial gas turbines are expected to grow over the foreseeable future, with electric power generation remaining the major market. Industrial gas turbines are now the unit of choice for electric power generation for most areas of the world due to the efficiency, lower emissions, and lead times of one to two years versus many years for plant siting and permitting for other types of power (Table 1.2.7)

Table 1.2.7- Average Lead Time

Type of Plant	Lead Time(Years)
Nuclear Power - Steam	15
Coal Fired - Steam	10
Oil Fired - Steam	7
Gas Fired - Steam	4
Gas Turbine - Simple and Combined Cycle	1-2

The industrial gas turbine market turned up sharply beginning in 1986. As shown in Fig. 1.2.5, the world market for 20 MW and larger industrial gas turbines increased from 5000 MW in 1985 to over 25,000 MW in 1991, a fivefold increase. The US. market represents approximately one-third of the world market, and the 60 Hz electric power generation market is approximately 40 percent of the world market.

The North American Electric Reliability Council (NERC) monitors capacity and planned power generation additions by US. utilities versus requirements (Ref. 5). Forecasts vary from a shortfall of 2 to 3 percent per year. Either 2 or 3 percent growth represents a very large potential market. Fig. 1.2.6 shows the expected shortfall in current planned additions based on 2 percent and 3 percent annual growth through 2001. Historically, electric demand capacity has tracked the growth in US. Gross Domestic Product except for periodic adjustments due to upheaval in fuel cost.

Of the planned additions shown in Fig. 1.2.6, only 20 percent of this capacity is under construction. At 3 percent per year growth, the shortfall is 137,000 MW, and at 2 percent per year growth, the shortfall is 91,000 MW.

Seventy-five percent of this shortfall is expected to be met with gas turbines because it will be filled on a short lead time basis. This is equal to a total market of between 127,000 MW at 2 percent growth and 162,000 MW at 3 percent growth for the planned additions and the shortfall from 1992 to 2001.

The existence of this shortfall does not mean that this, plus the nearly 5000 MW of retirements, would be open to ATS penetration, even were an ATS currently available. As noted approximately 20% of that

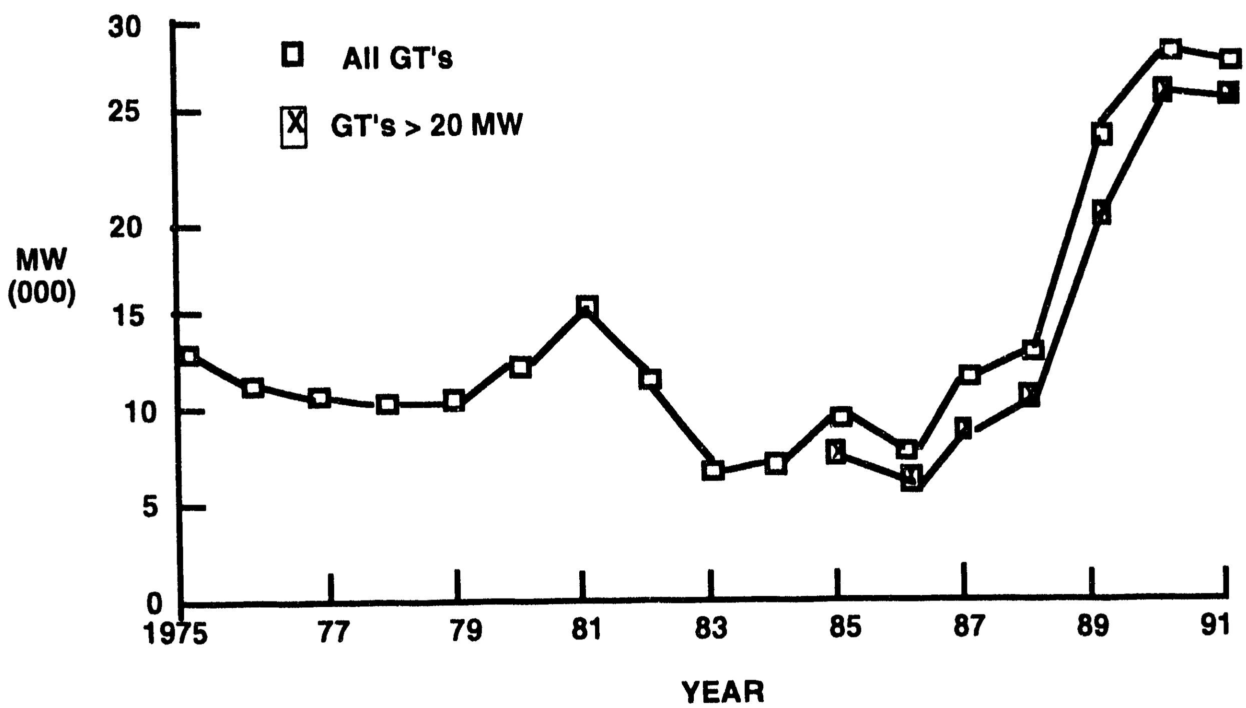

Fig. 1.2.5 - World Market for Gas Turbines

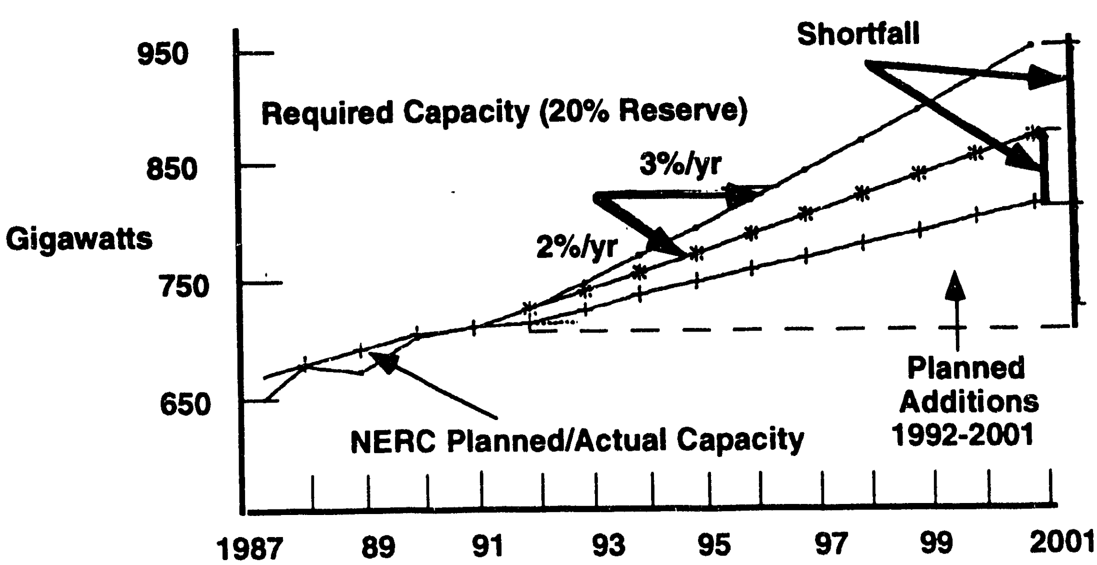

Fig. 1.2.6 - Generating Resource Shortfall

amount is already under construction. Also, the primary energy mix includes coal, nuclear, and hydro. Given the present status of coal gasification, few commercial plants could be built within this period, making gas and oil the only near-term viable fuels. In the years beyond 2001, given current growth rates, electrical consumption should continue to follow gross domestic product and grow at roughly 3.5%/yr. (Table 1.2.8). To keep pace, installed capacity would also increase in parallel (~ 10-12,000 MW/yr. in North America, nearly the same in Western Europe, with Eastern Europe and the Far East combined, again representing about the same (Ref. 6)). Coal could now become a viable ATS fuel, usually as a gas, although direct coal use cannot be ruled out.

Table 1.2.8 - Orders for New Equipment

	1990	91	92	93	94	95	96	97
Orders								
All Equipment Types - GW	74	73	68	68	73	76	79	81
Capaclty Growth Rate - %	3.8	3.7	3.2	3.1	3.3	3.3	3.3	3.3
Share bv Equipment Type - %								
Gas Turbine								
Combined and SimpleCycle	39	38	38	39	40	40	41	42
Fossil Steam	37	36	35	37	36	36	36	36
Hydro and Geothermal	20	23	24	24	24	21	20	19

	1998	99	00	01	02	03	04	05
Orders								
All Equipment Types - GW	84	87	89	93	96	98	102	105
Capaclty Growth Rate - %	3.3	3.3	3.3	3.3	3.3	3.3	3.3	3.3
Share bv Equipment Type - %								
Gas Turbine								
Combined and SimpleCycle	44	44	46	46	47	49	50	52
Fossil Steam	36	36	34	34	34	32	31	29
Hydro and Geothermal	18	18	17	17	16	15	15	14

The opportunity to successfully address the market with an ATS is dependent on a number of factors, of which some, like rate of world economic growth or availability of suitably priced fuel, will have great effect on market size, but are completely outside the control of ATS developers. Key market factors, however, that can be addressed as the ATS develops include:

- Reasonable first cost
- High efficiency
- Fuel flexibility
- Ease of permitting,
- World wide application
- Time of appearance

In order to acquire a market share large enough to pay for development and provide a reasonable return on investment, consideration of each of these must be used to shape ATS characteristics.

1.2.8.1 First Cost

Fossil steam plants using current technology for generation and emissions control cost upwards of \$1500/kW. Nuclear plants are even more. The goal for an ATS would be to have a capital cost significantly lower, say half, to ensure rapid market penetration. Currently, gas turbine prices are in the \$200 - \$210/kW range for 125 MW - 200 MW heavy frame machines; lower output aero-derivative machines have a wider range, \$275 - \$375/kW in the 25 MW - 50 MW sizes (Ref. 7). In Section 1.1.3, it was estimated that the gas turbine portion of the NGHAT system would be approximately \$201/kW, certainly in line with other large turbines. The total plant investment was estimated to be approximately \$511/kW, well below that for steam and nuclear and some 20 % less than the projected 1994 technology level combined cycle plant in Ref. 1.

1.2.8.2 High Efficiency

The desire for high efficiency is driven mainly by the interest in reducing fuel consumption and thereby reducing the cost of electricity. Due to the increased environmental responsibility of power generators, an additional benefit arises from higher efficiency , i.e., the accompanying reduction in emissions on a unit generated basis. The utility (or IPP) can realize reduced operating costs and has an opportunity to amass pollution credits which can be banked, traded, or sold. In the specific example of the NGHAT, the cycle efficiencies of over 61% and emissions (estimated) of less than 5 ppm for NOx, CO, and HC make this system extremely attractive (Section 1.1.3).

1.2.8.3 Fuel Flexibility

Historically, the utility industry has always required fossil plants to be capable of using different fuel forms. While gas and oil are the most easily exchanged forms, coal-fired plants are also sometimes run on alternative fuels. Nearly all of the gas turbine installations have dual-fuel (gas/liquid) capability. It has been estimated (Ref. 5) that oil will continue to be the fuel of choice for simple cycle gas turbines (1,744,000 MWhr oil vs. 667,000 MWhr gas in 1991 with projections of 6,590,000 MWhr vs. 685,000 gas in 2001). For combined cycles, the use pattern is reversed with gas having 144,100,00 MWhr vs. 539,000 MWhr oil in 1991 and projections of 103,478,000 MWhr vs. 803,000 MWhr in 2001. This nearly ten-fold increase in 10 years reflects the projected increase in installed gas turbine capacity as well as the increased capacity factor anticipated for these systems.

While the supply of natural gas is currently forecast to be sufficient for the foreseeable future, prudent operators will continue to require dual-fuel capability. As coal gasification technology becomes commercially viable, or natural gas becomes uneconomic for baseload generation, the ability of an ATS to adapt to this fuel will become important. The turbomachinery for the HAT cycle will follow in the footsteps of other UTC offerings and will have dual-fuel capability. When gasification becomes economic, the NGHAT can be converted to syngas rapidly and relatively inexpensively.

1.2.8.4 Ease of Permitting

A very great concern of utilities and IPP's is the time it takes to obtain the various environmental permits now necessary to construct and operate a power plant. Because of the excellent emissions characteristics of the NGHAT, its smaller plant footprint compared to steam and combined cycle systems , the requirement for less water, and modest thermal emissions to cooling water (only from the intercooler), the NGHAT should be easier to permit. The potential for reducing the permitting period results in lower front end costs and earlier operation.

1.2.8.5 World Wide Application

The ATS must respond to global market needs. This is because North America represents only about 35% of the world's power consumption (Ref. 8). Western Europe represents about 36%, Eastern Europe including the CIS, about 14%, and the Far East about 15%. Africa and Central/South America represent less than 1%. Eastern and Western Europe operate at 50 Hertz (Hz), as does most of Asia with the exceptions of parts of Japan and Korea (Ref. 7). What this means is that the ATS must be able to operate at 50 Hz with little or no compromise in performance. As part of the Ref. 1 study, commonalty of components for 50 Hz and 60 Hz IGHAT operation was considered. This will be reviewed for NGHAT application and component/system changes will be established as part of the development program.

1.2.8.6 Time of Appearance

The development schedule for the ATS would have the first commercial plant appear in 2001. To accomplish this at reasonable cost requires that the technology for the ATS be well along in its definition and development. The application of advanced aerospace technology through aero-derivative engines has been pioneered by UTC and has allowed the timely appearance of high performance engines to meet market demands. The development plan outlined in Section 1.2.5 calls for the transition from PW4000 to FT4000 to intermediate power FT4000 to NGHAT in a 6 year period. If the funding level were to be maintained to achieve these milestones, a prototype commercial-scale NGHAT could make an appearance before the DOE goal of 2001.

1.3 - Conversion to a Coal-Fueled Reference System

The NGHAT system described in Sections 1.1 and 1.2 can be converted to being coal fueled (IGHAT) with minimum change.

1.3.1 - Design Changes from NGHAT to IGHAT

A good deal of thought has gone into selecting the common turbine approach to the NGHAT and IGHAT. By allowing the turbine corrected flow rate to be the same for the NGHAT and IGHAT, the overall development costs are estimated to be the minimum. The only significant changes would be made to the compressors.

In the NGHAT, at the baseline 2408 RIT, an inlet flow of air at ISO of 627 lb/sec is required to mix with 15.9 lb/sec fuel, and 150.5 lb/sec water vapor to give a high pressure turbine corrected flow of 97 lb/sec. For the IGHAT, the flow of air would be 492 lb/sec, water vapor, 113 lb/sec, with remainder, 190 lb/sec of saturated syngas. The compressor work split must be changed to match the new high and low turbine work split. When going form NGHAT to IGHAT, the LPC and HPC ratios are respectively decreased from 6.9 to 4.9 and increased from 6.1 to 8.4. This would be accomplished by taking stages from the front end of the LPC and adding stages to the back end of the HPC.

An aerodynamic analysis done as part of the ongoing UTC effort showed that the gas turbine compressor, combustion, and control systems could be derived from existing hardware. This was discussed in Section 1.1.4.

1.3.2 - Required Development Work

The Ref. 1 study emphasized the use of syngas and identified that, as in the case of the NGHAT, the combustor remains the largest unknown. The use of syngas complicates the combustion in that it is itself saturated with water vapor (a Texaco-type quench system is envisioned for the syngas) and it may be approaching the capability limit of syngas to reach the ultra-high turbine inlet temperatures projected for the NGHAT. Thus, the main development need is in the combustor area. The combustor program initiated as part of the NGHAT program would be extended to IGHAT. Research facilities for evaluating syngas combustion already exist at UTC although they are currently laboratory scale.

A thorough heat transfer analysis will be necessary to establish the blade and vane differences between the NGHAT and IGHAT. The higher moisture content of the IGHAT gas would increase the effectiveness and could reduce the required cooling flow. This difference could be small and any cooling penalty could be offset by the added costs of producing different blade sets.

1.3.3 - Performance and Emissions of IGHAT

The HAT cycle system can make use of the low grade heat in the gasification process at high efficiency since it eventually shows up as moisture used at the highest cycle temperature (a schematic drawing of the IGHAT is given in Fig. 1.3.1). The efficiency of the IGHAT at a RIT of 2408 F is given in Ref. 1 as 42. 5% (LHV). Without going into the complicated cooling changes, it would be anticipated that the IGHAT would have the same incremental benefits from technology as the NGHAT (Section 1.1.3). Therefore, an efficiency gain of approximately 5.8 pts to 48.3% (LHV) could be projected.

Emission estimates in Ref. 1 for the IGHAT indicated NOx to be at 7 ppmv (dry, 15% O_2) with CO_2 at 1.676 lb/kWhr. Complete combustion of CO was projected. The NOx calculations are based on theoretical considerations and need to be experimentally verified.

1.3.4 - Schedule and Costs for IGHAT

The schedule and costs for the additional IGHAT effort beyond the NGHAT were developed as part of Section 1.2.5. Based on that schedule, if the IGHAT effort were to start in the third year of NGHAT development, it would take an additional four years and $100 million to demonstrate the IGHAT.

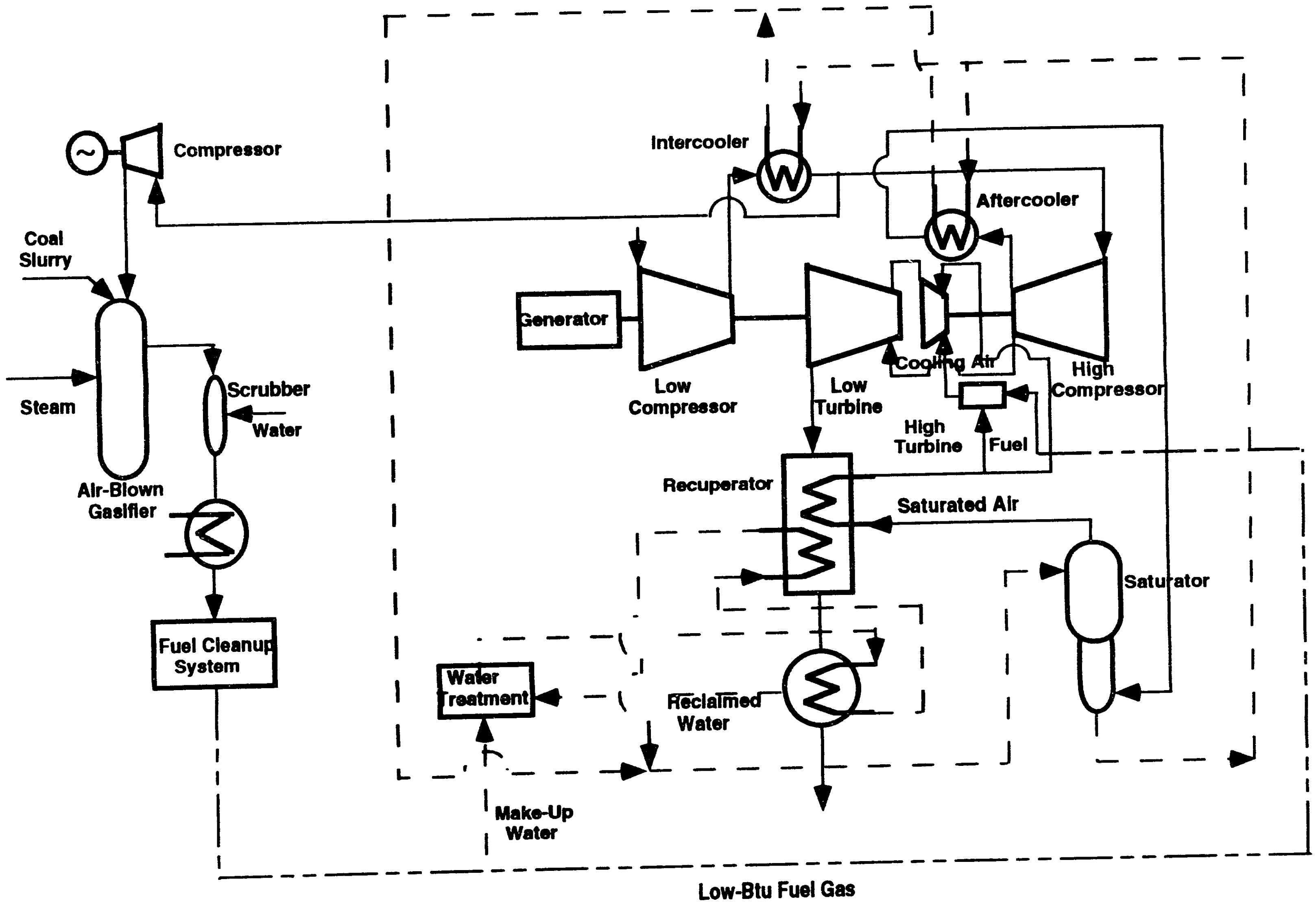

Fig. 1.3.1 - HAT Cycle with Air-Blown Gasifier

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APPENDICES

Appendix A -State-of-the-Art Performance Program

All of the performance studies for the proposed program will be carried out using a sophisticated power plant performance analysis computer system called "State-of-the-Art Performance Program" (SOAPP). This system utilizes modularized representations for each type of power plant component or subsystem permitting virtually complete freedom in selecting the desired configuration. It is capable of analyzing gas turbine engines, steam cycles, and combined cycles plus a complete power station including a gasifier and cleanup system, conventional or fluid bed coal combustor, heat storage, and other types of advanced technology. It is easy to use, versatile, and fast. It incorporates the most up-to-date calculation techniques available. The full system is currently in use on the IBM 3090 computer, in both batch and interactive modes of operation.

The development of SOAPP was done at Corporate expense. The basic system structure, communication techniques and mathematical routines were developed by Pratt & Whitney Aircraft (PWA) in 1970, as were the first component calculation modules and component map routines. Other maps and modules for aircraft gas turbines have been added continually by PWA. In the intervening years, PWA assisted the United Technologies Research Center and the TurboPower Marine Division with the extension of the SOAPP library to include modules necessary for advanced industrial power plants. These additions included boilers, heat exchangers, fuel cells, coal and oil gasifiers, fuel gas cleanup systems, coal-fired conventional and fluid bed combustors. SOAPP has been used in a large number of Corporate- and Government-funded projects involving aircraft gas turbines, industrial gas turbines, combined cycle power systems, integrated gasification and fluid bed-combined cycle power systems, fossil- and nuclear-fueled closed cycle systems using fluids such as helium and argon, and nuclear aircraft power plants.

<u>Functional Construction of SOAPP</u>

SOAPP is a computer system comprised of a preprocessor coupled to a library of modules and maps. The following general description is provided as background information concerning the functional construction of SOAPP from an overall viewpoint. It is this overall functional construction which provides the program's extreme versatility.

Streams and Modules: Just as a power plant system can be considered to be made up of major component elements connected by pipes and ducts, the basic SOAPP structure simply involves streams and modules. The streams represent identifiable, homogeneous mass flows comprised, in general, of a mixture of several constituents and passing through connecting pipes and ducts. The modules are engineering representations of individual component elements programmed into computing code. The physical properties of each stream (i.e., pressure, temperature, enthalpy, etc.) are fixed except as modified by the module calculation. In other words, the modules are connected by the streams. A module would then be a representation of any power plant element which acts to change physical properties, or constituency, of the flow passing through. Examples would include: compressors, turbines, combustors, heat exchangers, pumps, boilers, etc. The streams act to maintain material and energy balances between component modules.

Map/Module Pairs: In order to define a specific piece of hardware, its performance characteristics and physical geometry must be established along with the general physical operating behavior. For example, a compressor is represented by an ideal isentropic compression process, modified by an efficiency, and delivers flow according to some rotor speed and pressure ratio schedule. The compressor module would consist of the necessary equations and calculation steps to represent these functions for any general compressor. However, a given compressor of a specified number of stages and aerodynamics design would perform its function with flow, rotor speed, pressure ratio, and efficiency all related by a particular relationship, usually determined empirically by a rig, model, or full-scale test. These empirical relationships are called compressor maps in conventional gas turbine terminology. In SOAPP language, the term maps extends to any empirical relationship connecting any variables of any real component. Thus, if a gasifier is represented in module form by chemical equilibrium calculations, its map might be an empirically derived graph which relates the percentage of unconsumed carbon content in the product gases to the oxygen/carbon ratio. Therefore, a complete component element representing specific physical hardware must be described by a map/module pair, where the module computes the operation of a general component type, and the corresponding map furnishes actual data for the particular component desired.

Preprocessor: The preprocessor is the heart of SOAPP. This is a self-contained computer program which automatically translates a description of any given stream and component configuration arrangement into a working computer model. The preprocessor sets up all internal computing logic and iteration procedures necessary for a complete performance calculation, as well as the output format. To accomplish this, the preprocessor draws upon the SOAPP library according to coded instructions.

Library: The SOAPP library keeps in central storage a complete array of maps and modules. Their location in the library is designated by a descriptive code name. In addition, the library is used to store preselected configuration input. This enables an engineer to set up a desired configuration just one time and keep it available for future running for an indefinite period of time. Further, if it is subsequently desired to change a given power plant configuration after it is initially set up, it may be called in from the library, modified as desired, then returned to the library as modified for future reference.

Operating Modes: The SOAPP system has four basic operating modes: (1) configuration; (2) design; (3) run; (4) print.

In the configuration mode, the complete power plant system is set up according to components and streams in a given order.

In the design mode, the engineer must give the computer system the numerical constants required to define the starting point for a given operating condition. These constants represent the minimum number of independent variables required to define the equilibrium operation of the simulated power plant. This is the starting point of a performance calculation and for an open cycle configuration it requires at a minimum a specification of ambient conditions. It would also include specification of a component, physical variables if its size or geometry are unspecified, or, vice versa. For example, a heat exchanger might have effectiveness specified, with tube surface area left as a dependent variable to be calculated, or tube geometry must be specified with effectiveness to be calculated.

In the run mode, a series of design point calculations or off-design conditions can be found. After the initial design case has been established by the design mode, a series of cases may be run from run mode input instructions.

Print mode controls the calculation output. The engineer has at his disposal a series of options. He can elect the standard output print, he can selectively display calculated data on a graphic scope, or he can instruct the computer to automatically plot output variables either for graphics scope display or on hard copy on an automatic plotting machine.

Inter-Active Operation: By means of the remote terminal (which is usually a PC or work station equipped with a graphics monitor) the engineer can use SOAPP on a inter-active basis. The program is stored on a central IBM 3090 computer which is coded for operation on a time-sharing system. Calculations from the remote terminals are queued in and out of the computer in a matter of seconds, for essentially instantaneous access. As a consequence, the engineers can run data, inspect and evaluate the results, make further modifications, and run new data at their own pace.

Summary: The engineers can use the SOAPP system to complete an entire simulation problem from start to finish without any special computer programming, providing all necessary elements reside in the library. They can set up their system in the configuration mode, describe physical hardware and external conditions in the design mode, and run performance calculations in the run mode, with results automatically provided.

Available SOAPP Modules

SOAPP modules all contain design, off-design, and dynamics representation capability. The SOAPP library also stores gas and physical property data for all potentially important working fluid constituents that might occur in the system. The program automatically handles thermodynamic properties for mixtures of substances in homogeneous flow streams.

A partial list of modules currently available in the SOAPP library includes:

Compressors	Steam Boilers	Pumps
Drive Turbines	Gasifiers	Flow Mixers
Oil and Coal Combustors	Regenerators	Flow Dividers
Ducts	Gas/Gas and	Two-Phase Flow
Valves	Gas/Liquid Heat	Turbines
Reformers	Exchangers	Saturators

Brief descriptions of some of the standard system component modules are provided in the following paragraphs.

Gasifier: Three different representations for gasifier modules can be used in SOAPP. The are input/output, equilibrium, and set yield models. The input/output model is a very simple, but extremely useful model. For this model the gasifier inlet and exit composition and temperature are specified (to correspond to known gasifier performance), but the reactant and product flows are scaled to meet the requirements of the power plant being simulated.

The equilibrium gasifier model employs a computational procedure developed at UTRC for the solution of specific combustion problems involving airbreathing and rocket propulsion devices. With this method, the equilibrium composition and thermodynamic properties of a reacting mixture are obtained by applying a successive approximation procedure to find simultaneous solution of the standard equations of chemical equilibria and conservation of (atomic) mass for specific values of pressure and either temperature or enthalpy. This model is capable of estimating gas composition for a number of gasifier types (e.g., Texaco, Shell, Koppers-Totzek) which operate at high temperature.

Corrections to equilibrium estimates can be made to better simulate real gasification processes by using the set yield model. This model uses empirical data to specify a fixed or preset yield of one or more species such as carbon or methane. These yields are essentially set aside at the start of the calculation and the remaining constituents brought to equilibrium. The set yield species are taken to be at the equilibrium temperature and an overall heat balance is achieved. Use of the set yield option for carbon enables an accounting to be made of soot.

Gas Turbines: Pratt & Whitney Aircraft has had on-going programs for over twenty-five years to channel component and full engine data into the improvement of simulation representations and the measurement of their accuracy. Installation effects have been determined experimentally for a number of industrial installations and these have been modeled with the SOAPP program.

Because of the availability of these background data, proven capability exists to model the wide range of industrial turbines in use today as well as their derivatives. The SOAPP representation will model the wide range of industrial turbines in appropriate detail to describe the compressor and turbine systems, combustor systems, air and gas valves, inlet guide vanes, dual-fuel capability, generator and normal controls. The assumed component technology level, the component design characteristics (scaled to appropriate size), and other physical and empirical relationships will be based on industrial engine designs. Also, both single and multi-spool engines can be simulated by relatively simple input changes.

Ducts and Pipes: Ducts and pipes are represented in SOAPP with standard routines that calculate off-design pressure losses by varying the fluid total pressure loss with the square of the stream flow parameter for low Mach number gas streams, or with the square of Mach number for high Mach number gas streams. This method works will for constant friction factor assumptions, but for the case of low Reynolds number or contraction or expansion losses, empirical representations can be used. Liquid stream pressure losses are normally estimated as a function of the square of mass flow.

Steam Boiler: The steam boiler and other combined cycle heat exchange components including superheater, reheater, evaporator, economizer, deaerator, feedwater heaters, condenser, etc., are modeled in SOAPP with a single multi-purpose module that predicts relative design values of tube surface area, mass volume, etc., and performs the off-design overall mass and energy balances, phase change, etc. Temperatures, pressures, and flows are estimated using simplified relationships to obtain values for component overall heat transfer effectiveness or empirical data describing specific installations. This same boiler representation can be used to simulate any heat recovery boiler in a gasification process.

Steam Turbines: The SOAPP steam turbine model uses published flow and efficiency prediction techniques for large steam turbines. Other representations can be added to model auxiliary steam turbines (i.e., boost and oxygen plant compressor drive turbines).

Gas Saturator: An existing model of a gas saturator in the SOAPP library is used to simulate the process of adding water vapor to a gas stream as would be done in a packed tower or using a spray vaporizer. The module maintains the necessary overall material and energy balance relationships along with the psychometric calculations required to describe this component. The exit temperature of the gas leaving the module is equal to the dew point of the water vapor.

Reformer: The reformer module is intended to be used in the turbine exhaust stream. Heat from the stream is used to provide the energy needed to produce hydrogen from a mixture of steam and methane. The module simulates the catalytic process by calculating the chemical equilibrium at the reactor temperature and maintaining the necessary heat balance. The SOAPP system provides for the communication of heat and mass flow and for the necessary fluid properties. Equilibrium data and heat of reaction are provided by the individual module.

Gas Regenerators, Intercoolers, and Other System Heat Exchangers: Several heat exchanger representations are available in the SOAPP library ranging in complexity from simple single element calculation procedures, utilizing log-mean-temperature difference and effectiveness methods, to highly detailed multi-element procedures. SOAPP can accommodate parallel flow, counter flow, cross flows and multi-pass combinations.

Expansion Turbine: Because SOAPP automatically scales representative hardware data to account for gas composition effects upon component performance, the expansion turbine can be represented at design with the appropriate gas turbine module with good accuracy. Thus, the modeling detail described for the gas turbine components is applicable to an expansion turbine as well.

System Compressors: The system compressors for the air-blown plant booster compressor, oxygen plant air compressors, or exhaust gas recirculation compressors may be modeled in the same detail as the gas turbine compression system using existing representations.

Gas Properties: In order to achieve gas property representation accurate enough for thermodynamic calculations, yet compact enough for computational speed, the gases in the gas turbine, combustors, gasifier, etc. will be treated as having no more than 10 constituents. The properties of the gas mixture will be calculated using the technique described in Keenan and Kaye Gas Tables. The properties of the constituent gases, O_2, N_2O, A, H_2O, CO, and CO_2 are based on K&K data; the properties of other constituents are based on JANNAF Thermochemical Tables.

Steam Properties: Steam properties are calculated using the ASME industry standard, the 1967 IFC Formulation for Industrial Use.

Appendix B - Advanced Technology Turbine Cooling and Materials

Turbine Cooling

Current turbine cooling technology is based on the use of air extracted from various locations in the compressor and piped to the desired locations in the turbine. As turbine temperatures increased, the amount of cooling air increased (Fig. B.1). However, increases in cooling effectiveness have flattened this increase.

The turbine cooling requirement is determined by the mission of the application. Component lifetime varies from aircraft (1000's of hr) to utility (10,000's of hr)use. Also, the type of component failure mechanism influences cooling,e.g., aircraft turbine failure is normally by low-cycle fatigue while oxidation/erosion and creep are the concerns in utility applications. This means that thermal stress (bulk temperature related) governs aircraft while local hot-spot temperature is the key in utility use.

The methodology used for determining the cooling flow is based on design parameters derived from experience. The first step is to determine the design criteria for the application. For the baseload utility application, the criterion is 100,000 hr, B5; this means that there would be 50 blade failure in 1000 engines, each having run 100,000 hr. Stress analysis identifies material creep characteristics. The material properties set allowable metal temperatures, Tm (bulk temperature for creep, hot-spot temperature for corrosion/ oxidation).

A value for air cooling effectiveness is determined from the equation

$$\emptyset = (Tg-Tm)/(Tg-Tc)$$

The cooling flow is found from Fig. B.2.

Where Tg= Tg(ave)+dT (burner pattern factor)
+dT (trim)
+dT (deterioration)
+dT (control tolerance)

Tc= Tc(ave)+dT (passage heating)
+dT (blade pumping)

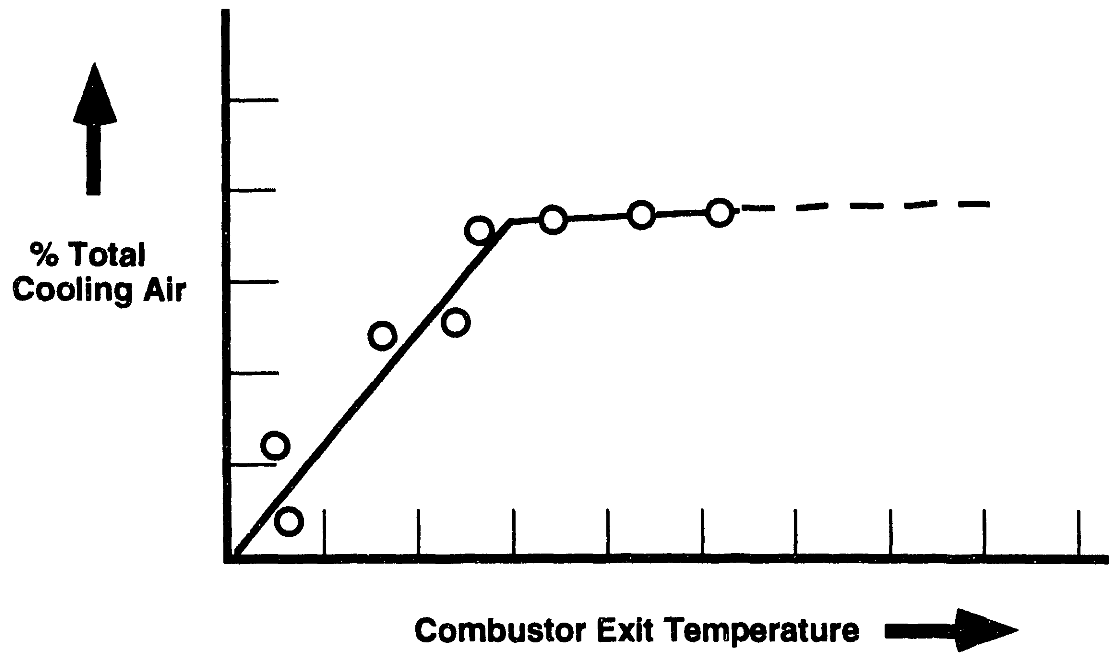


Fig. B.1 - Trend in Cooling Air.

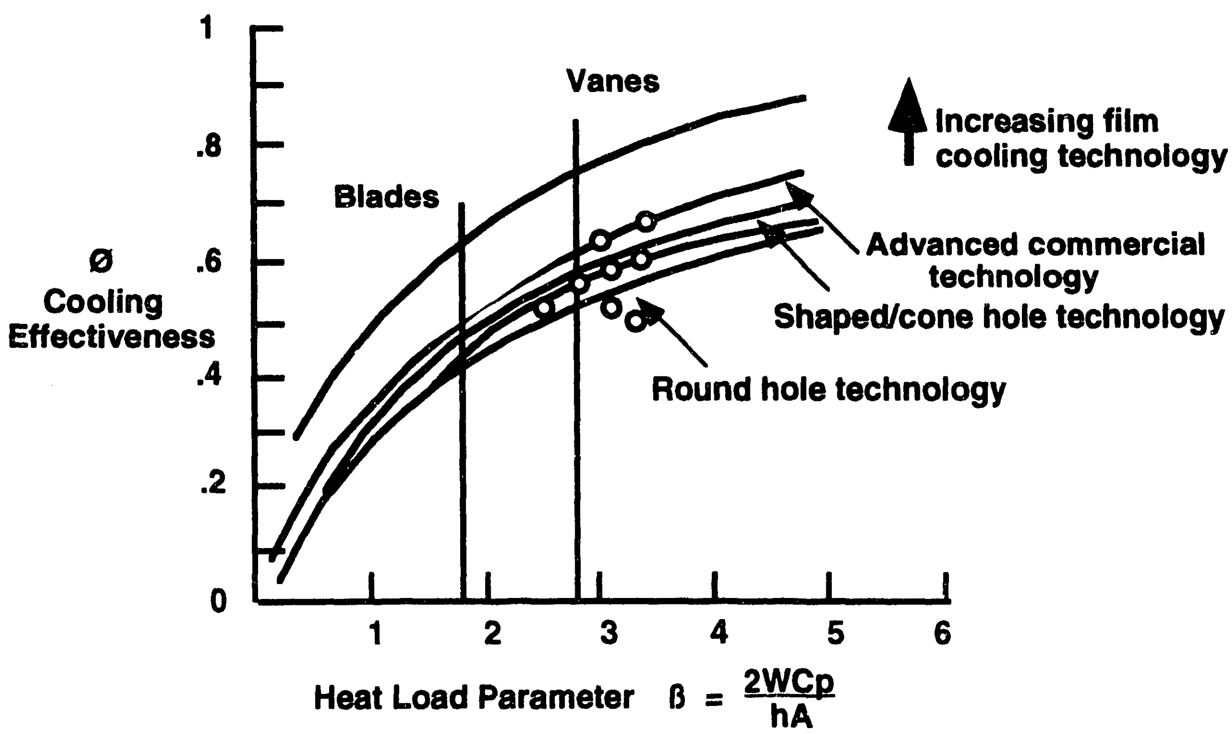


Fig. B.2 - Determination of Cooling Flow

-dT (blade TOBI)
+dT (deterioration)

Tg(ave) is total temperature for vanes and relative temperature for blades.

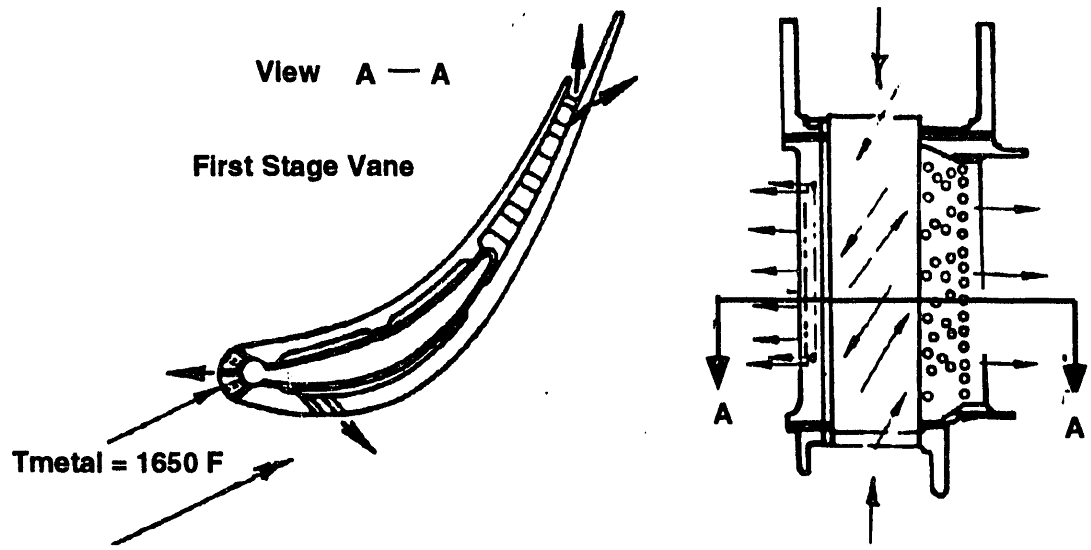
ß = heat load parameter
W = the required cooling flow
Cp = specific heat of coolant
h = heat transfer coefficient
A = cooling area

Note: The burner pattern factor is radial and circumferential for vanes; radial only for blades. An attenuation factor is applied for downstream rows.

Knowing Ø for the technology level at which the engine is operating, a value for ß is found and from that, a value for W, the cooling flow.

As an example, consider the first stage vane in the FT4000 HAT engine (Fig. B.3) The maximum metal temeprature (assumed for this calculation to be 1650 F) will occur at the leading edge. The gas stream temperature is 2538 F (combustor exit temeprature) plus 408 F pattern factor correction, plus 100 F correction for trim, deterioration, and control tolerance; a corrected temperature of 3146 F. The coolant for the first vane is taken from the diffuser leading to the recuperator and is a mixture of water vapor and air at a temeprature of 372 F. From the equation for effectiveness, a value of Ø is found to be 0.54 resulting in a cooling flow of 52.1 lb/sec. Similar calculations for the first blade estimates Ø at 0.46 and a cooling flow of 19.1 lb/sec.

There are several levels of cooling technology shown in Fig. B.2. By using the technology now appearing in commercial flight engines, significantly higher cooling effectiveness can be realized. (The top curve in Fig. B.2 is representative of advanced fighter engines, a goal that could be attained in industrial engines in the next decade.) The effectiveness for the advanced cooling scheme in vanes would be 0.63 and for blades, 0.525. Keeping metal and cooling air temperatures, lifetime, and cooling flow rate constant, the higher effectiveness would result in increases in allowable gas temperature as shown in Fig. B.4. Note that the incremental temperature gain is limited by the blade to 300 F; the vane could sustain a 700 F increment. Thus, the firing temperature of the engine could be increased by 300 F and would result in efficiency gains (Fig. B.4) that meet the DOE goal of 60% as well as power gains that would reduce turbine specific costs ($/kW) .



Tg(max) = 3146 F

Tg(avg.)	= 2538 F
Δ T (pattern factor) =	408 F
Δ T (trim, det., tol.) =	200 F
	3146 F

Tca (avg) = 352 F
Tca (max) = 371 F

Ø = (3146 - 1650)/(3146 - 372)
Ø = 0.54

Fig. B.3. - Determination of Cooling Effectiveness

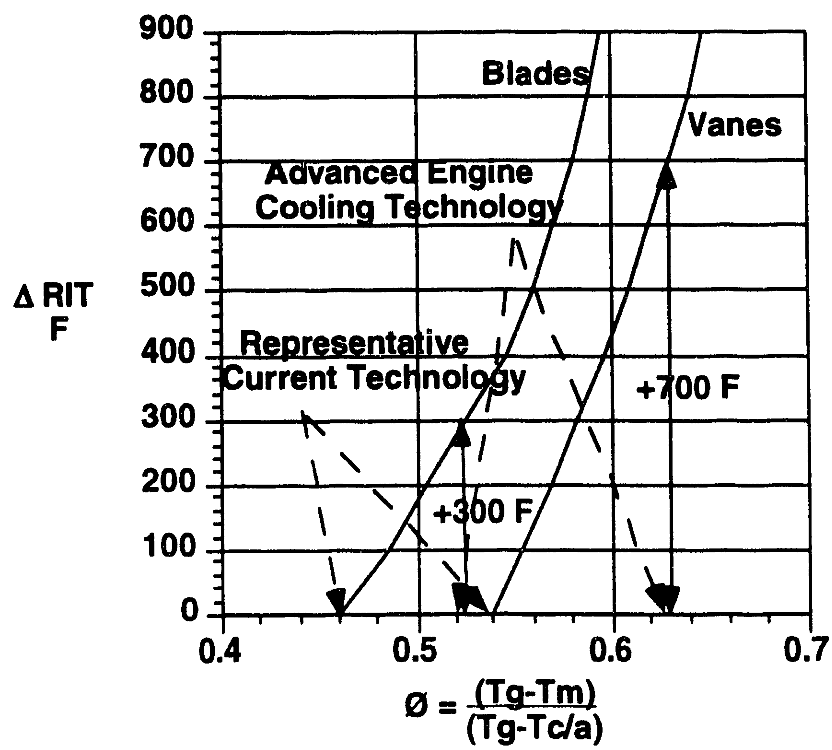


Fig. B.4 - Gain in RIT from Advanced Cooling Scheme.

A tremendous amount of UTC resources are expended on the continuing development of cooling technology for military and commercial aircraft. These advances will be available to bring the FT4000 from its introductory configuration to one that meets, and potentially greatly exceeds, the DOE goal of 60% efficiency.

Materials

The discussion of materials could, in itself, be hundred's of pages. Rather than a long discussion, only two trends will be shown. In Fig. B.5, the increase in metal temperature is shown as a function of time. While the chart purposely doesn't go beyond 1990, it can be seen that metal temperatures in advanced military aircraft applications are in the neighborhood of 2000 F. The last entry on Fig. B.5 indicates the use of single crystal materials. The advantages of this material are shown in Fig. B.6. The FT4000 and its HAT turbine derivative make use of single crystal technology developed by PW and used widely in the newest aircraft engines.

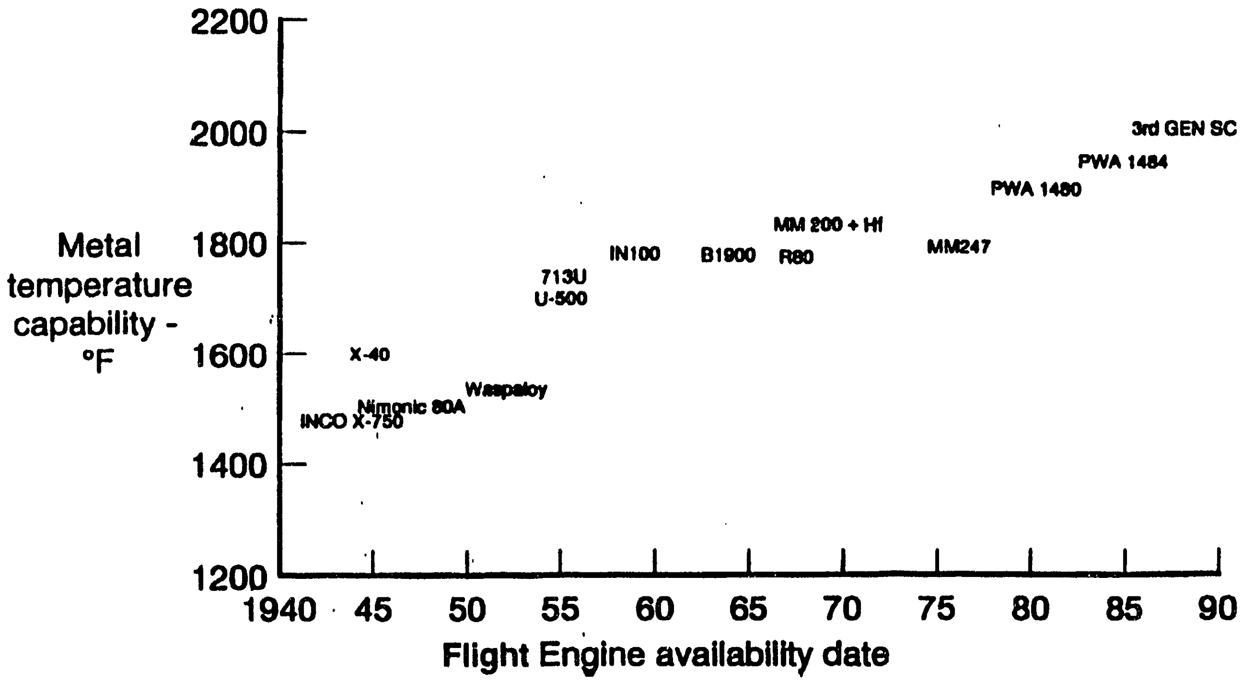


Fig. B.5 - Metal Temperature Capability

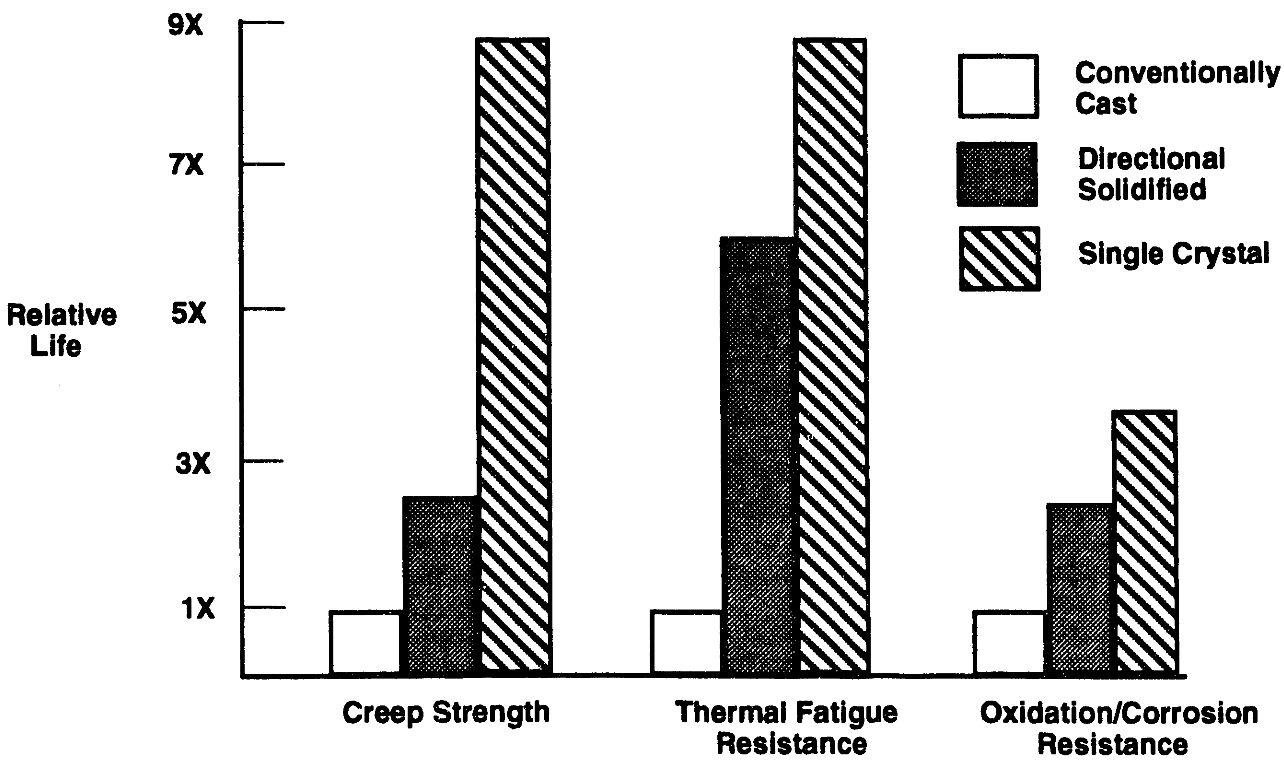


Fig B. 6 - Advantage of Single Crystal Construction

APPENDIX C - EMISSIONS

The physics of NOx production in gas turbines is well known. The rate of NOx formation increases exponentially with temperature, and linearly with time. All the approaches to minimizing NOx within the combustor rely on reducing the reaction temperatures. These approaches do not include catalytic combustors that work on the exhaust gases.

Significant rates of NOx formation occur above 2800 F; however, there must be sufficient residence time within the combustor at temperatures above 2200 F to permit carbon monoxide to be converted to carbon dioxide. In order to achieve the very low NOx levels required of current industrial gas turbines (25 ppm corrected to 15% oxygen), it is necessary to stay within this narrow band of temperatures on both a global and local level. This becomes more difficult as manufacturers make efforts to meet DOE's aggressive goal of 5 ppm. Pockets of poorly mixed fuel and air will burn at temperatures above 3000 F regardless of the overall average temperature.

The temperature of the combustion process is a function of the local equivalence ratio (Figure C.1). Equivalence ratio is a normalized fuel-air ratio. An equivalence ratio of 1 represents a stoichiometric mixture where the proportions of fuel and air permit complete combustion with no excess of either fuel or air. An equivalence ratio less than 1 represents a lean mixture (excess air), greater than 1 represents a rich mixture (excess fuel). All gas turbines operate to an overall equivalence ratio that is lean. However, locally within regions of the combustor, the fuel-air mixture may be rich or lean.

The most common method of NOx suppression in conventional gas turbines is the use of water or steam injection. Use of water or steam has the additional effect of increasing power, although water injection does hurt heat rate. In order to achieve maximum effectiveness, the H_2O must be injected into the primary zone of the combustor. The effectiveness of this method is well established (See Figure C.2 from Reference 1). The ordinate, RNox, is the ratio of NOx with water or steam added to the dry level. The abscissa is equivalent water to fuel ratio. For water, the figure is used as is, steam/fuel values are divided by 1.6 to become equivalent water/fuel values. However water injection beyond about 1.25 (the actual value depends upon the individual combustor) leads to significant increases in carbon monoxide. Therefore it is important to start with a dry NOx level that is reasonably low, so as to minimize the H_2O required.

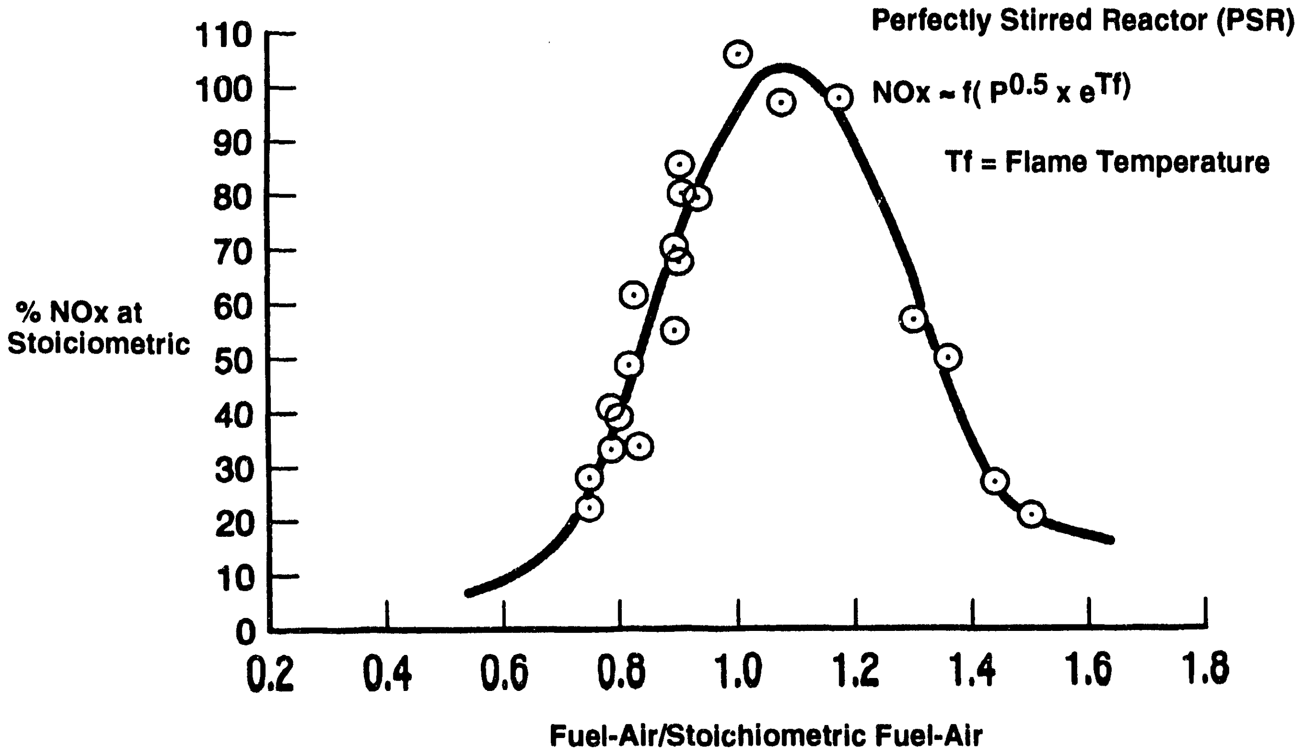


Fig. C.1 - NOx Generation.

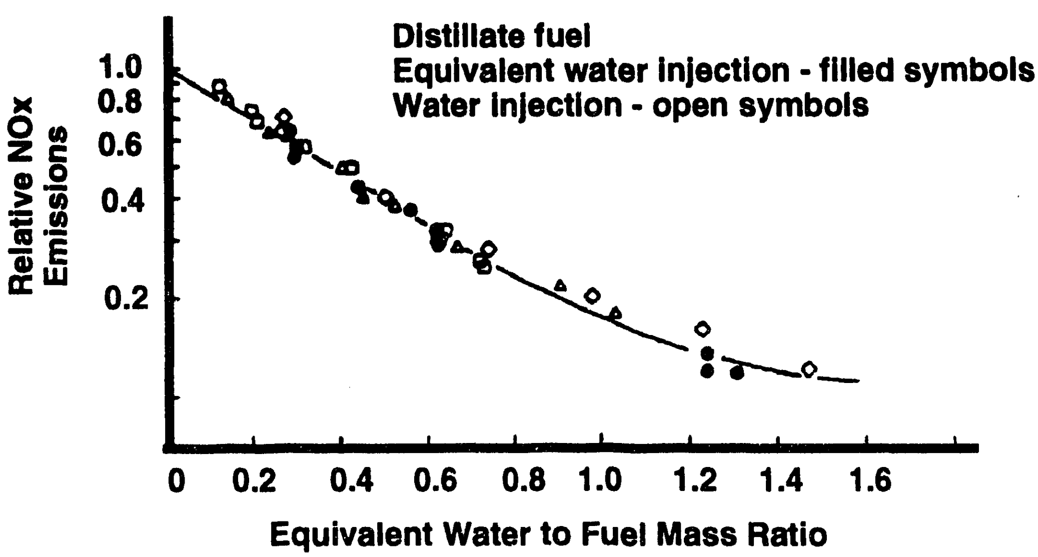


Fig. C.2 - Effect of Water Injection on NOx Emissions.

There are two basic dry approaches (no water injection) to low NOx. One is lean combustion where a relatively large amount of air is introduced into the front end of the combustor to minimize the temperatures. Proper implementation of this approach requires introducing a very well mixed fuel and air into the combustor or else local high temperature combustion will occur. This is the approach taken by all current low NOx industrial combustors since gas, the more important industrial gas turbine fuel, can readily be mixed with the air outside of the combustor. There has been less success with liquid fuel which has to be vaporized before good mixing can occur. There is a greater tendency to auto-ignition of a liquid fuel/air mixture prior to entry into the combustor. One successful lean liquid approach was implemented in the FT4 (Reference 2).

There are some fundamental difficulties with the lean approach. First, the combustor is barely stable. At powers below 100% it wants to blow out. There are a number of approaches to solve this problem. One is a variable area device which would reduce the air to the front end at low power. Another is a piloting scheme where a small amount of fuel is injected into a region with less air flow at low power to enhance stability. Still another approach is fuel staging where injectors are progressively shut off as power is reduced thereby effectively enriching those nozzles that are still flowing fuel. Variable area devices have been notoriously unsuccessful in the combustor environment and staging produces distortions in the gas temperature distribution entering the turbine.

Second, with so much of the air tied up in the front end, very little is left for cooling or development of exit temperature profiles. The higher the combustor exit temperature the more severe the problem since more air is required at the front end in just those combustors that have the highest heat loads and need the most cooling. While a very lean gas flame is expected to have lower heat loads than conventional combustors, the system must also operate on liquid fuel. Even if some degree of premixing were achieved with the liquid fuel, heat loads would be higher (more luminous flame, poorer mixing).

The second approach to dry low NOx is the rich burn quick quench (RBQQ) technique. Here the front end runs very rich (equivalent ratios of 1.5), thereby suppressing the flame temperature and NOx formation. Since the overall combustor fuel-air ratio is lean, air must be rapidly added to the combustor to quench the reaction and go rapidly from rich to lean. The mixing must be carried out swiftly and efficiently to minimize NOx formation. The rich primary zone (See Figure C.3) eliminates the stability problems associated with the lean approach. The

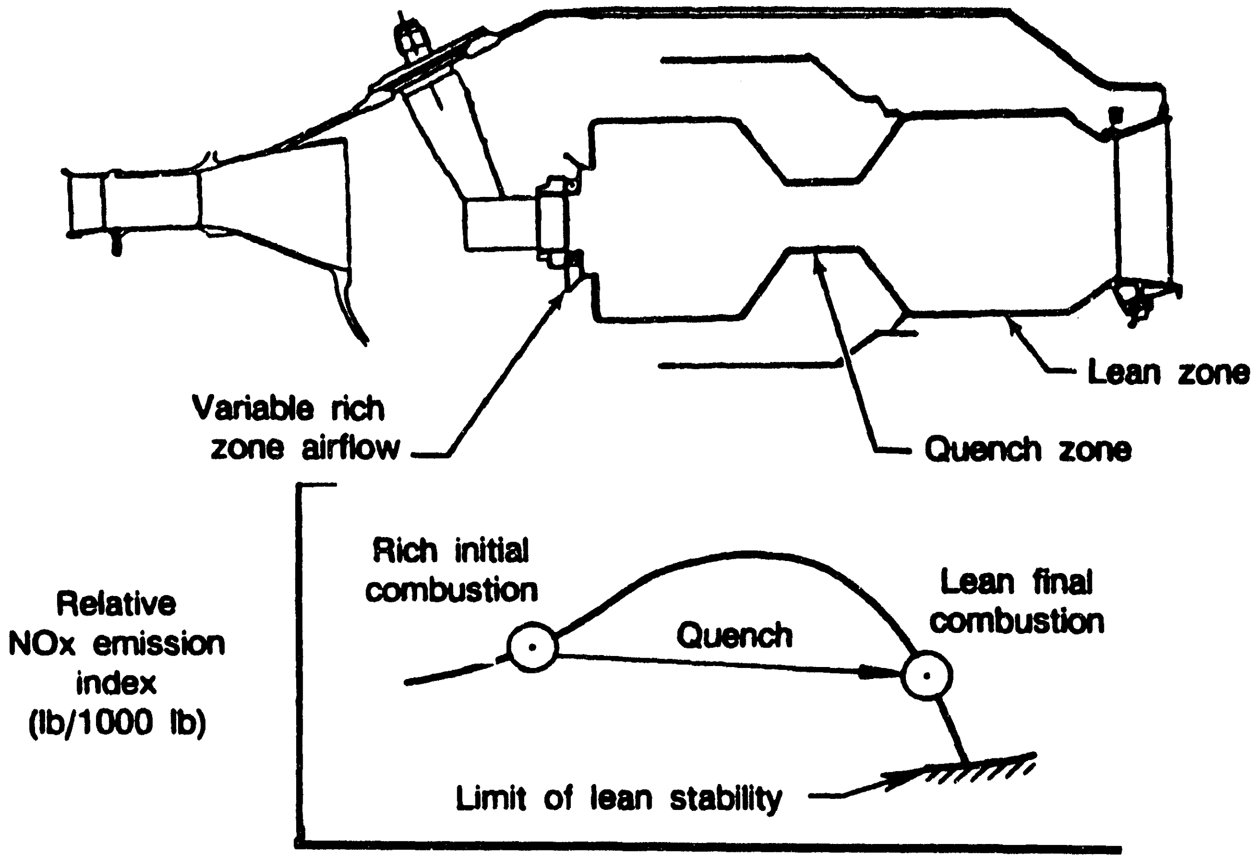


Fig. C.3 - Rich Burn Quick Quench Combustor

fuel system is simpler. However, the requirement to operate so rich precludes cooling air entering into the primary zone. There is insuffi cient pressure drop available to convectively cool the primary zone and also inject the spent cooling air in the rapid quench zone. An RBQQ approach in high pressure gas turbines requires combustor construction not currently available (ceramics).

Both of these dry approaches (lean and RBQQ) may be further improved with the addition of water or steam. In the lean approach, H_2O would be injected into the primary zone, much the same as in current applications. In the RBQQ approach, the injection would be in the quench zone. In the lean approach, the big unknown is the effect of H_2O injection into a system that is already marginally stable. Testing is required to establish stability limits. In the RBQQ, stability is not an issue, although CO levels may be a problem. There has not been much work in this area.

The simple cycle FT4000 dry low NOx combustor is designed to achieve 25 ppm NOx at a combustor exit temperature of 2500 degrees F. If the engine were to be operated at higher combustor exit temperatures by increasing fuel flow, NOx levels would climb exponentially, assuming no hardware changes. (see Table C.1). If the hardware could be redesigned, then very high levels of air flow through the fuel injector would be required to maintain 25 ppm NOx. There would be very little air remaining for cooling or for adjusting the exit gas temperatureture. A combustor construction requiring little or no cooling air (water cooled combustor or ceramic construction) would be necessary.

An alternative would involve injecting water or steam into the fuel injector. This violates the gas fuel, dry low NOx concept for the engine. However, dual fuel units will have provision to inject water or steam to meet the 42 ppm NOx requirement. There will most likely be stability problems associated with water or steam injection. Stable operation will probably require pilot fuel flow and extensive testing to map out the stability limits of the configuration.

Table C.1 - Estimated NOx Production

Combustor Exit Temperature F	2500	2600	2700
Hardware Unchanged			
Dry NOx ppm - 15% O_2)	25	40	100
Water/fuel mass ratio to 25 ppm NOx		0.4	0.8
Steam/fuel mass ratio to 25 ppm NOx		0.6	1.3
Redesigned Hardware			
Additional Injector Air (% Wab*)	75	80	85

*Baseline = FT4000 dry low NOx simple cycle

As the requirement for higher exit gas temperatures approaches 2800 F, the dry techniques described above, while reducing NOx levels below those from conventional combustors, cannot achieve the 25 ppm goals. The combustor exit conditions are in the region of high NOx formation. The front end of a lean burn combustor must operate richer than the exit conditions and the RBQQ combustor cannot quench below the exit conditions.

The FT4000 operating on a HAT cycle, may have 25 to 45 weight percent moisture content in the fuel. The moisture, either as part of the fuel or injected separately, should reduce emissions. Even if fuel flow were increased to maintain the combustor exit temperature, the presence of water vapor should effect the NOx formation chemistry. However the same mechanisms that suppress the NOx chemistry may also impair the combustion stability. Bench scale testing under these conditions is necessary to determine the effect of water vapor on the stability and NOx formation.

A preliminary analysis (Ref. 3) was made of the combustion stability and potential emissions from a combustor having moisture contents typical of NGHAT. It was estimated that combustion times 3 to 5 times that of normal combustors would be required (Fig. C.4). Estimated emissions of NOx and CO appear to be at levels which are close to or meet the DOE goals (Fig. C.5).

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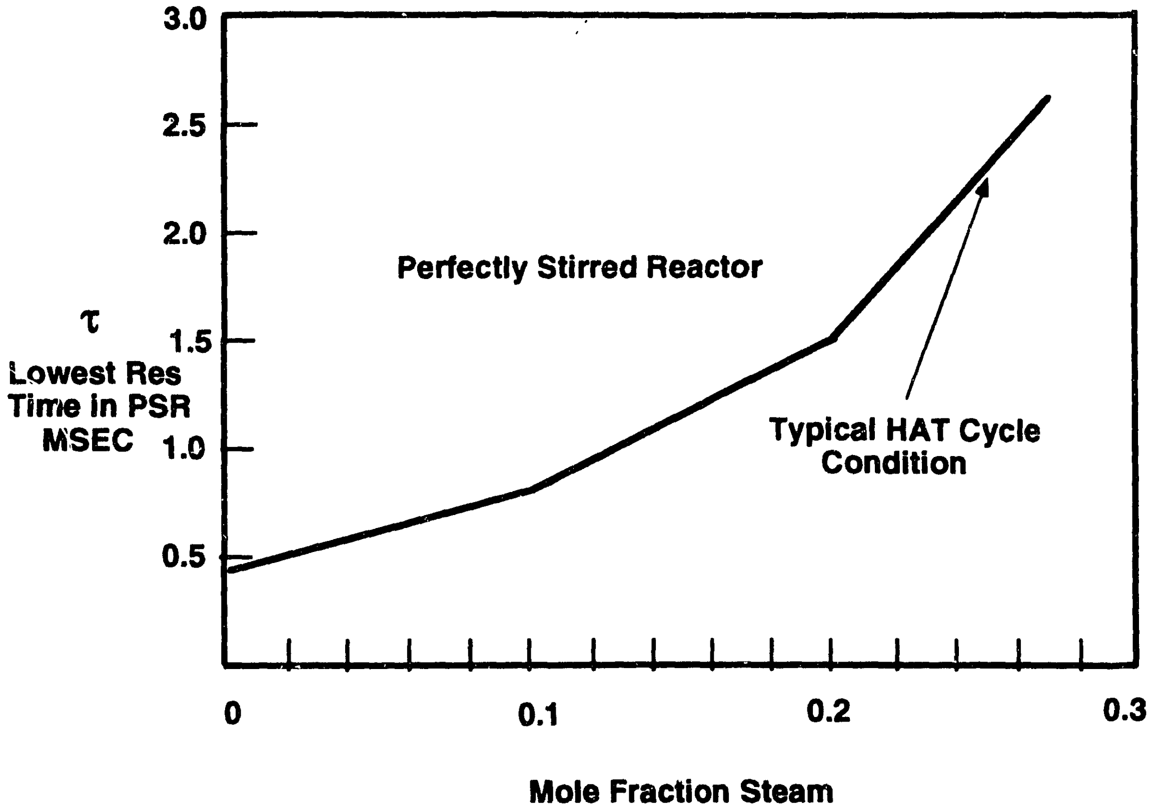
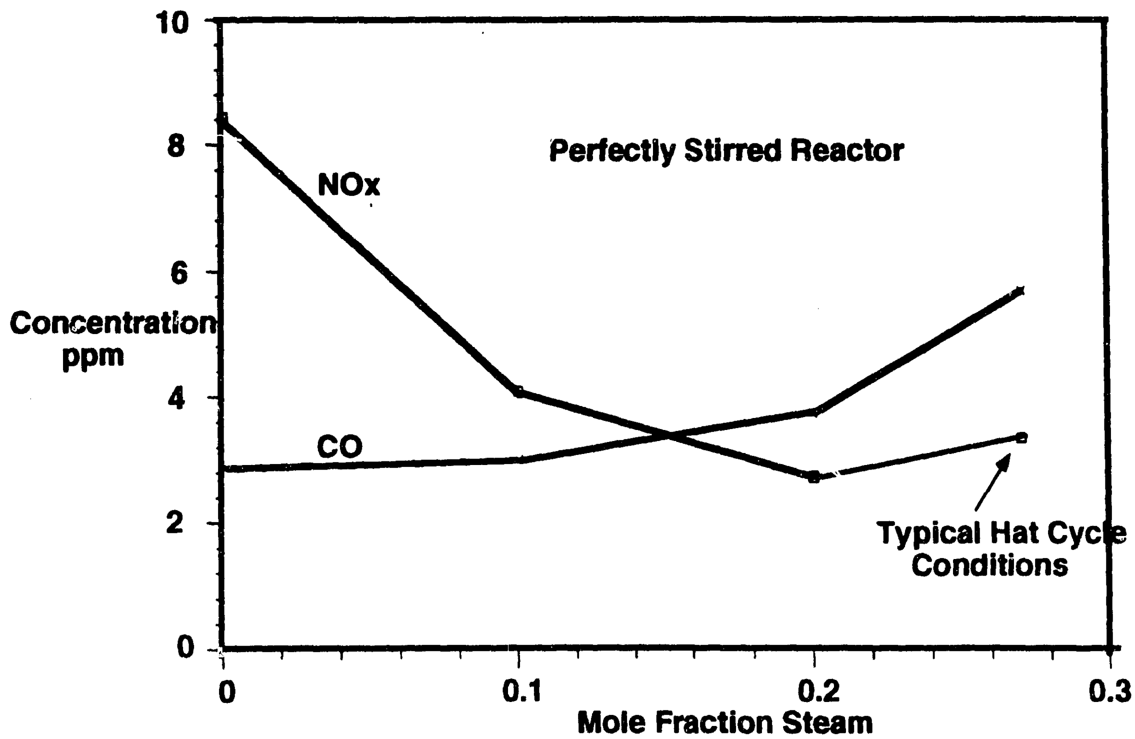


Fig. C .4 - Critical TAU (PSR) at Lean Limit.



ʼq. C.5 - Estimates of Emissions at 15% O2

Appendix D - Reliability, Availability, Maintainability

Reliability, availability, and maintainability (RAM) are key concepts in the design of an ATS. A full RAM analysis of the UTC HAT-cycle is beyond the scope of this Phase of the DOE ATS Study. However, it is instructive to describe in general, the approach used by UTC in performing RAM analysis.

During the past decade, considerable progress has been made in applying RAM principles to the design of UTC turbines and turbine systems. UTC has placed a significant effort on integrating existing gas turbine operational, failure mode, and maintenance data with standard reliability analysis methods and techniques in support of RAM evaluations and design recommendations for the state-of-the art FT8 26 MW industrial turbine and for the support systems for the Siemans V84 series of heavy frame engines marketed by UTC. This design for reliability philosophy will be extended to the ATS program.

In 1989, UTC contracted with Strategic Power Systems, Inc. (SPS) to provide reliability engineering and consulting expertise to the design of control and ancillary systems for gas turbines. A methodology was developed that effectively combined reliability engineering tools, operating history and analysis to meet established RAM goals. The primary scope of a full RAM assessment is:

- Develop representative and meaningful component and system failure rates, forced outage factors, unavailabilities, and starting unreliabilities based on generic field data;

- Provide design recommendations minimizing the impact of component unreliability;

- Identify maintainability issues and other RAM related considerations pertinent to overall design;

- Establish a rigorous and repeatable RAM process which provides the ability to update system assessments based on new or more current operating data; and

- Identify how the overall RAM program for configuration control, tracking and monitoring field performance data, will be extended to the design for reliability process and follow-on product line activity.

In the ATS program, as in other engine development programs, the issuers to be addressed can be organized into the following:

- Identification of critical systems and components
- Anticipation of system failure modes
- Assessment of operating and maintenance practices on RAM
- Development of RAM parameters for specific systems and components
- Achievement of goals for increased reliability and minimized costs

UTC's goal is to secure availability and reliability growth through the continuous improvement process. To assure this, it is essential that: 1)a valid and accurate data base system be used to collect timely operational, failure, and maintenance data on a continuing basis; and, 2)a realistic reliability engineering assessment must be made to estimate the benefit or value of product improvement. UTC and SPS have worked together to provide the tools to achieve these goals.

- ORAP - ORAP (Operational Reliability Analysis Program) is a reliability, availability, and maintainability reporting system with a specific focus on gas turbines in various applications, cycle configurations, duty cycles, and across various manufacturers.

- CARD - CARD (Computer-Aided Reliability Design) is a reliability engineering tool allowing the development of "automated" models based on plant arrangement and system design. These models directly interface with ORAP data to develop estimates of RAM performance.

This approach was used in the development of the FT8, UTC's current offering to the utility and IPP market. Data from the North American Electric Reliability Council (NERC) on operation, failure, and maintenance of gas turbines were used by ORAP and CARD to identify:

- System and component RAM estimates
- Current RAM trends
- Critical component identification
- Root cause failure information

Once the data assessment had been made on a system and component basis, CARD's reliability block diagram program was used to develop system models based on the schematic component flow (Piping, Instrumentation, and Control). This process allowed the FT8 design to meet customer expectations of availability in excess of 95%, reliability approaching 98%, and starting reliability in excess of 95%.

END

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