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**Phase I
System Scoping and Feasibility Studies**

Topical Report

March 1993

Work Performed Under Contract No.: DE-AC21-86MC23167

For
U.S. Department of Energy
Office of Fossil Energy
Morgantown Energy Technology Center
Morgantown, West Virginia

By
Westinghouse Electric Corporation
Orlando, Florida

MASTER

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Office of Fossil Energy
Morgantown Energy Technology Center
P.O. Box 880
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By
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March 1993

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1 Executive Summary

The National Energy Strategy (NES) calls for a balanced program of greater energy efficiency, use of alternative fuels, and the environmentally responsible development of all U.S. energy resources. Consistent with the NES, a Department of Energy (DOE) program has been created to develop Advanced Turbine Systems (ATS). The objective of this 10-year program is to develop natural gas-fired base load power plants that will have cycle efficiencies greater than 60% (LHV), be environmentally superior to current technology, and also be cost competitive.

The ATS program has been structured into five elements:

- Innovative Cycle Development
- High Temperature Development
- Ceramic Component Development/Demonstration
- Base Technology
- Coal Cross-Cut Studies

Within each program element there are several planned phases. The Innovative Cycle Development Element includes four phases:

- Program Definition/Planning Studies
- Turbine Component Design and Testing
- Large-Scale Component and Integrated Subsystem Testing
- System Integration Demonstration

This report documents the response of the Westinghouse Electric Corporation to Element I, Phase I, Program Definition/Planning Studies of the Innovative Cycle Development. Westinghouse has selected a natural gas-fired ATS Reference Cycle to develop, manufacture and market, if turbine development costs were not an issue.

The Westinghouse Reference Cycle provides the means for meeting all the performance targets for the ATS program including high efficiency, reasonable plant and operation costs, low emissions, and adaptability to both natural gas and coal fuels. The cycle uses a gas turbine with a pressure ratio of 20 to 1. There is a surface type intercooler between the low pressure (LP) and high pressure (HP) compressors. Heat is recovered from the turbine exhaust gases successively in a recuperative air heater and then in a waste heat boiler. Turbine inlet temperature (TIT) will be 2680°F with a corresponding rotor inlet temperature (RIT) of 2500°F. The 230+ MW Reference Cycle should have a combined cycle efficiency of 57% (LHV). Use of a closed loop

steam cooling system should raise the overall plant efficiency to 61.5% (LHV). Other high temperature innovations using advanced cooling techniques with new materials, such as ceramics, will elevate combined cycle plant efficiencies to around the 65% (LHV) mark.

Where possible, the proposed Reference Cycle will utilize existing hardware that has been modified to produce a pressure ratio of 20 to 1 with a turbine 2680°F TIT (2500°F RIT). The HP compressor, and turbine will be a two bearing rotor in a single housing. The combustion system will be a low NO_x Westinghouse combustor design. The LP compressor and steam turbine will be individual rotors in separate housings. All components will be directly coupled to form a single shaft running at 3600 RPM. The design can be readily adapted, with a 3000 RPM, 50Hz version developed for export applications. The generator rotor will be double ended so that the steam turbine and combustion turbine can drive from opposite ends. The exhaust flow from the turbine is axial into the recuperator.

The natural gas-fired turbine will be designed so any advances in materials (e.g., use of ceramics) or advanced cooling to raise the RIT in the 2650° to 3000°F range can be incorporated with a minimal effort into the demonstration plant turbine.

The proposed Westinghouse ATS Program can be implemented within a 10-year development effort as outlined by DOE to develop an advanced cycle and hardware that can raise combined cycle efficiencies to the 60% (LHV) plateau. During Phase II, Conceptual Design and Product Development, tasks will be proposed that support compressor and turbine upgrading, as well as cycle innovations. Demonstration of the advanced cycle hardware would be performed in Phase III and IV of the proposed program.

Estimated schedules for the development work leading to advanced gas turbines suitable for operation with either natural gas or a coal-fueled system are included in this report. Also discussed is Westinghouse's plan to commercialize the natural gas-fired Reference System and the identification of potential market opportunities for ATS for utility and independent power producer plants.

2 Background

DOE's National Energy Strategy projects that the U.S. will need 10,000 megawatts (MW) per year of new electrical generating capacity in the first decade of the twenty-first century to meet the needs of the U.S. economy. There is also a 10,000 MW per year market to replace a portion of the 700,000 MW capacity that existed in 1990. Therefore, utilities and Independent Power Producers (IPPs) could order as much as 20,000 MW per year of new capacity. Much of this market will be for base-load systems.

Today's turbine systems feature high fuel-to-electricity efficiencies. Efficiencies, on a lower heating value (LHV) basis, for large natural-gas-fired combined-cycle systems for the utility market reach 53%. LHV efficiencies for single-cycle turbines in the industrial sector reach 40%. Other advantages of modern gas turbine systems include low capital costs and a short installation cycle.

Even though manufacturers will make improvements in the 1990's, efficiency levels will reach a plateau. A firing temperature of 2500°F may well be the limit for air-cooled, hot-gas-path airfoils. Only cycle innovations (rather than incremental design changes) will achieve LHV efficiencies in the 60% range for natural-gas-fired utility machines.

Advanced turbine systems using natural gas or coal-derived fuels are candidates for the post-2005 power-generation market. While utilities could use the mid-1990's class of turbines for the post-2005 market, the payoff for the U.S. consumer is large, if ATS are available. The availability of ATS would provide efficiency benefits (less carbon dioxide emissions and lower cost of electricity), environmental benefits (nitrogen oxide emissions one-tenth of today's New Source Performance Standards) and economic benefits (jobs and export sales).

Congress authorized funds in fiscal year 1992 to start planning for a comprehensive effort to develop advanced gas turbines systems. These advanced systems should be ultra-high efficiency, ultra-clean and cost-competitive. Base-load applications in the utility, IPP and industrial markets are the targeted uses. Cost of power for the advanced systems must be competitive with that of current systems. An ATS should be ready to enter the commercial market by the year 2002.

In response to DOE/METC's request to Westinghouse, an R&D plan for Advanced Turbine Systems (ATS) that will achieve LHV efficiencies in the 60% range for natural gas-fired combined cycles has been developed. Westinghouse has also identified one coal-fueled system that would be marketable after the year 2005 that incorporates ATS cycle features. The coal-fueled system will use the turbine developed in the natural gas-fired Reference System. A R&D plan was also developed to increase the chances of successful operation of the selected coal-fueled system.

To assist Westinghouse in this Phase I Program, Carnegie Mellon University (CMU) and Power Tech Associates (PTA) were selected as team members. CMU had the responsibility for producing material and energy balances, together with equipment descriptions and sizing for the advanced Brayton cycle power generation plant. Special features of the plant that were considered included evaporative compression cooling, gas turbine reheat, inter- and aftercooling,

steam cooling of expander blades, recuperative air heating, and steam injection. PTA provided suggestions, advice, and analysis concerning advanced turbine systems that have potential to meet the program efficiency, emissions, cost, operating and reliability goals.

Westinghouse's program for the development of proposed systems follows the phased approach requested by DOE. For each development requirement, a program was outlined that included:

- Background
- Objective
- Technical Approach
- Work Statement
- Schedule

3 Program Objectives - Phase I

Westinghouse performed System Scoping and Feasibility Studies for development of ATS. The goal of these studies were to identify incentives, technical issues and resource requirements to develop natural-gas-fired and coal-fired ATS. Where possible, Westinghouse made use of previous work. Westinghouse performed all analyses for ambient conditions (i.e., 59°F, sea level, 60% relative humidity).

3.1 Selection of Natural-Gas-Fired Reference System

Westinghouse selected a natural-gas-fired ATS Reference System that we would like to manufacture and market, if turbine development costs were not an issue. The natural-gas-fired Reference System could be demonstrated at full-scale, in the 1998-to-2002 time frame and would have the following characteristics:

- Efficiency, on a lower heating value (LHV), greater than 60% for utility-scale, natural-gas-fired, combined cycle systems.
- Expected base-load emissions consistent with the following:
 - No post-combustion emissions control devices.
 - 1991 state-of-the-art system efficiencies.
 - Nitrogen oxides (NO_x), 10 parts per million by volume dry (ppmvd) at 15% oxygen.
 - Carbon monoxide (CO) and unburned hydrocarbon (UHC), 20 ppmvd.
- Cost of electricity (COE) would be at 10% below COE of 1991-vintage power generation systems in the same market class and size range, and meeting the same emissions requirements.
- Advanced natural gas-fired turbine and ATS would be amenable to change for use in a coal-based system.
- Reliability, availability and maintainability (RAM) comparable to today's advanced power generation systems.

3.2 Description of Natural-Gas-Fired Reference System

Westinghouse has described and documented the selected natural-gas-fired Reference System which includes:

- A description of the natural-gas-fired Reference System for an advanced turbine system.

- A projection of the performance (output and heat rate) of the natural-gas-fired Reference System.
- A projection of COE for the natural-gas-fired Reference System.
- Projection of RAM for the Nth plant version of the natural-gas-fired Reference System.
- ATS program structured into four Phases: Phase I (1992) - System Scoping and Feasibility Studies, Phase II (1993-1994) - Conceptual Design and Product Development, Phase III (1994-1997) - Testing of Large-Scale Components and Intergrated Subsystems, and Phase IV (1998-2002) - System Demonstration. An estimated schedule for performing the program along with potential levels of private sector cost-sharing for the program and benefits of the program versus costs are presented.
- Identification of intermediate development needs.
- The contractor's plan to commercialize the natural-gas-fired Reference system, after a successful full-scale demonstration.
- Identification of potential market opportunities for the ATS in utility and independent power producer sectors.

3.3 Conversion To A Coal-Fueled Reference System

Westinghouse has identified one coal-fueled system that we feel would be the most marketable after 2005. The coal-fueled system must use the turbine used in the natural-gas-fired Reference System.

Westinghouse has provided:

- A brief description of design changes needed to adapt the gas turbine in the natural gas-fired Reference System to the selected coal-fueled system.
- The development work that would be undertaken to increase the chances of successful operation of the selected coal-fueled system.

Westinghouse concentrated on the design and development issues related to the gas turbine and the turbine's interface with the appropriate combustion system.

4 Selection of Natural Gas-Fired Reference Systems

Westinghouse pioneered gas turbine development in North America, and its product line development mirrors the general gas turbine inlet temperature trend shown in Figure 4.1. Combined cycle efficiency improvements have followed the general advance in gas turbine technology reflected in this inlet temperature trend.

For example, in 1970, a Westinghouse supercharged, evaporative cooled, W301 (1450°F turbine inlet temperature) gas turbine exhausting into a heavy fired, reheat boiler achieved an annual average efficiency (LHV) of 39.6%. This West Texas Utilities, San Angelo Power Station reportedly held the distinction of achieving the highest operating efficiency in the US for a number of years.

Today, with a gas turbine inlet temperature of about 2460°F [2300°F rotor inlet temperature (RIT)], a 501F based combined cycle is quoted at 53.2% (LHV) plant efficiency, ISO with: 1) dry low NO_x; 2) 3 pressure level, reheat, heat recovery steam generator (HRSG); 3) rotor cooling air heat transfer into HRSG; and 4) fuel gas heating.

As can be seen from Figure 4.2, the bottoming cycle has been fine tuned to convert as much of the turbine's exhaust heat into electricity as possible. Additionally the gas turbine incorporates all the material, aerodynamic and cooling technology gains of the last 25 years.

Figure 4.1 Gas Turbine Inlet Temperature Trend

Figure 4.2 501F Based, Natural-Gas-Fired, Combined Cycle

The target of 60% combined cycle efficiency by the year 2000 is shown to be beyond reach in Figure 4.3, where cycle efficiency is plotted against the turbine inlet temperature trend of Figure 4.1.

Figure 4.3 Combined Cycle Efficiency Trend

At about 2700°F turbine inlet temperature, a combined cycle efficiency just under 57% is projected. To investigate the possibility of overcoming this 3.5 percentage point shortfall, reconfiguration of the gas turbine based combined cycle was studied.

4.1 Cycle Diagrams

The first step in the reassessment of the traditional approach was a review of the thermodynamic fundamentals inherent in the gas turbine cycle. With this background, methods for increasing basic cycle efficiency should become more evident.

The basic cycle employed in a gas turbine power plant is the Brayton cycle illustrated on a temperature-entropy (T-S) diagram in Figure 4.4; it includes an adiabatic air compression, a constant pressure heating by combustion, an adiabatic expansion, and a constant pressure heat rejection process. In the most simple plant configuration, heat rejection is accomplished by the exhaust of the still hot combustion gases from the turbine to the atmosphere. Irreversibilities in the compression and expansion processes, due to fluid friction and gas leakage, result in small entropy increases. Pressure losses in flow ducts and in the combustor are represented by short, horizontal (isothermal) line segments throughout the diagram. The work input for compression or output from expansion is equal to the product of the heat capacity of the air or combustion product gases and the corresponding temperature difference. The heat input from combustion or output from rejection is represented by the areas under the corresponding curves on the T-S diagram. The energy losses due to irreversibilities, including the pressure losses, are indicated by the areas under the corresponding line segments. Finally, the work output of the cycle (per given quantity of air input) is equal to the heat input from combustion minus the output from rejection. All of these quantities are indicated in Figure 4.4.

The efficiency of the plant is defined as the quotient of the work output and the heat input. Increased efficiency in this basic cycle can be achieved by:

- Increasing the temperature level at which heat is supplied to the cycle - increasing the maximum firing temperature. Figure 4.5 illustrates this cycle modification. Firing temperatures ultimately are limited by the materials of combustor and turbine construction.
- Decreasing the temperature level at which heat is rejected from the cycle - increasing the pressure ratio over which the combustion product gases are expanded so that the exhaust gases emerge from the turbine at a lower temperature. Figure 4.6 illustrates this cycle modification. The turbine exhaust pressure is essentially atmospheric; increasing the pressure ratio requires increasing the compressor outlet pressure at significant complexity and cost.
- Using the rejected heat in a second, bottoming Rankine cycle. Figure 4.7 illustrates a single pressure level steam-based Rankine cycle positioned below the air-based Brayton cycle. The temperature at which heat is rejected from this combined cycle arrangement is significantly reduced; and the efficiency, increased. The difference in temperature between the rejected heat of the Brayton and the absorbed heat of the Rankine cycles, however, represents an irreversibility which reduces the overall system efficiency. To

Figure 4.4 T-S Diagram for Basic Brayton Cycle

Figure 4.5 Increasing Firing Temperature

Figure 4.6 Increasing Pressure Ratio

Figure 4.7 Adding A Bottoming Rankine Cycle

some extent this temperature difference can be reduced by using a multi-pressure (perhaps 3) level Rankine cycle.

- Using the rejected heat of the turbine exhaust gases in the basic Brayton cycle by recuperatively preheating the air after its compression - and perhaps also the fuel. Figure 4.8 illustrates recuperative air preheating. Because the heat capacity of the air is only slightly less than that of the combustion products a relatively small temperature difference between the two gas streams can be established and maintained, if a heat exchanger with sufficiently large area is used. Irreversibility in this process is thus minimized, and cycle efficiency enhanced.
- Removal of heat from the air compression process - cooling or intercooling either by transfer to heat exchanger surfaces or by evaporation of injected liquid water - lowers its temperature, enhances its ability to absorb heat over a wider range of temperature, and incidentally reduces the work of compression. Figures 4.9 and 4.10 illustrate various approaches to cooling in the process of air compression including:
 - compressor inter- and aftercooling either by surface or evaporative heat removal.
 - evaporative cooling of the compression process itself by the introduction of fine water mists into compressor stages.
- The ability of methane or natural gas fuel to absorb rejected heat from the turbine over the available temperature range can be enhanced by adding steam and promoting the endothermic methane-steam reforming process. Figure 4.11 illustrates the temperature of the reacting methane-steam mixture as a function of the heat added. The ability of the compressed air stream to absorb heat can also be enhanced by adding liquid water, which vaporizes over the available temperature range of the rejected heat.
- Reducing the irreversibilities due to fluid friction and gas leakages in the compressor and turbine and to pressure losses throughout the system as a whole.
- Altering the cooling of turbine blades. The effects of cooling by air extracted from the compressor are schematically illustrated in Figure 4.12. The reduction of the blade temperature, usually by a combination of surface and film cooling, results in a reduction of both the temperature and the entropy of the expanding product gases, while increasing their quantity. The mixing of the coolant air and hot expansion gas also causes a momentum or pressure loss. As a consequence, the work output of the turbine is reduced as indicated in Figure 4.12, which illustrates the introduction of cooling air in 4 stages. Blade cooling losses might be eliminated by the use of high temperature materials, ceramics; or reduced by the use of more effective cooling arrangements -- for example, internal blade cooling using air or steam returned to the cycle; transpiration cooling using smaller quantities of air or steam coolant.

Figure 4.8 Recuperative Air Heating

Figure 4.9 Intercooling and Aftercooling of Compression Process

Figure 4.11 Fuel Reformate Temperature vs. Heat Absorption

- Increasing the temperature level at which heat is added to the cycle by the employment of gas turbine reheat -- expansion of the combustion gases to an intermediate pressure, a second combustion process, and a final expansion to the outlet turbine pressure. Figure 4.13 illustrates this overall process. Reheat:

- increases the work output of the cycle per quantity of air compressed.
- correspondingly decreases the size and cost of heat recovery equipment per quantity of work output.
- increases the extent to which the oxygen in the air is consumed in the plant, and thus presumably reduces NO_x formation.

Reheat, however, increases the quantity of cooling air which may be required by the turbine blades. It is not clear, therefore, the extent to which plant efficiency may be increased.

4.2 Flow and T-Q Diagrams

The overall flow diagrams and associated temperature-heat (T-Q) diagrams for the heat recovery processes corresponding to these various cycle descriptions are shown in Figures 4.14 to 4.28. T-Q diagrams are useful in representing and analyzing the various heat recovery arrangements. Constant pressure heating or cooling of single phase streams are represented by lines whose slope equals the heat capacity of the stream; boiling or condensing of single component streams by horizontal lines. The corresponding temperatures of streams in heat transfer equipment are directly represented, and heat transfer area computations can be carried out from the temperature difference and heat quantities directly represented on the diagram.

$$\text{Area} = \text{integral over } Q \text{ (dQ/heat transfer coefficient)}^*$$

The effectiveness of a heat recovery process in enhancing cycle efficiency can in part be judged by how closely the temperature of the stream receiving heat approximates the temperature of the stream from which heat is being recovered.

The flow diagrams for conventional Brayton cycle plants combined with 1- and with 3-pressure level Rankine bottoming cycles are shown in Figures 4.14 and 4.15, respectively. The Rankine bottoming cycle comprises the heat recovery boiler with economizer, evaporator, and superheater sections; steam turbine; condenser; feedwater heater and deaerator; and, condensate and feedwater pumps, and stack. (If cooling water is not available, an air-cooled condenser may be required.) Corresponding T-Q diagrams, Figures 4.16 and 4.17, illustrate the recovery of heat from the turbine exhaust by each of these Rankine cycles. The flow diagrams also show the extraction of blade and disc cooling air for the turbine from three intermediate pressure stages and from the outlet of the compressor. A portion of the outlet air is cooled to assist in lowering the temperature of severely stressed turbine parts. All of the cooling air is eventually bled into the expanding combustion gases as a consequence of the combined surface and film cooling.

* temperature difference between fluid streams

Figure 4.13 Reheat Brayton Cycle

Figure 4.14 Conventional Brayton Cycle with Single Level Rankine Bottoming Cycle

Figure 4.15 Conventional Brayton Cycle with Three Level Rankine Bottoming Cycle

Figure 4.16 Single Level Heat Recovery Steam Generator, T-Q Diagram

Figure 4.17 Three Level, Reheat, Heat Recovery Steam Generator, T-Q Diagram

Air recuperative heating, as the primary means of heat recovery is shown in Figure 4.18. Compressor inter- and aftercooling, Figure 4.19, are used to maximize the quantity of heat transferred to the compressed air and to minimize that portion of heat rejected by the gas turbine transferred to the less efficient Rankine cycle. The corresponding T-Q diagram is shown in Figure 4.20. Compressor cooling can be carried out either in surface exchangers or in evaporators where water is sprayed into the hot compressed air. The water for evaporative cooling can effectively be heated either in a surface intercooler of the air compressor [as in the humid air turbine (HAT) cycle] or in the economizer of the heat recovery boiler. The evaporated water increases the mass flow of gases to the combustor and the turbine, thus increasing the work output per quantity of air compressed. More than one stage of intercooling of the air compressor may prove advantageous.

As a limit, continuous evaporative intercooling of the air compression process might be carried out, as indicated in Figure 4.21, by creating a mist of fine, preheated water droplets, perhaps about 5 micron in diameter, throughout the passages of the compressor. These droplets would evaporate, cool the air, and reduce the work required for compression. Creating such a mist, avoiding its coalescence, preventing compressor blade damage by impact, and providing the time/volume required for its evaporation requires a novel development and design effort.

As indicated by the T-Q diagram for air preheating, Figure 4.20, there is an increasing temperature difference between the turbine exhaust gases and the air. It is possible, therefore, to inject liquid water into compressed air as it passes through the recuperative heat exchanger as shown in Figure 4.22. Again, the evaporation of this water increases the quantity of heat recovered from the exhaust gases and the quantity of gas expanded in the gas turbine. As more water is injected into the air, the temperature of the exhaust gases leaving the recuperator can be lowered to the point at which a Rankine bottoming cycle is no longer required to achieve high operating efficiency (Figure 4.23).

As an alternative to recuperative air preheating, exhaust heat can be recovered in a natural gas fuel stream as shown by Figure 4.24. (No air compression cooling is carried out in this plant configuration since it is not required to enhance the recovery of exhaust heat; in this case compressor cooling would merely increase the consumption of fuel.) Water or steam is mixed with the fuel in order to carry out the highly endothermic methane-steam reforming reaction

The water gas equilibrium reaction results in the simultaneous formation of CO_2

It is necessary to provide excess H_2O in the fuel gas to prevent the deposition of solid C in the fuel reformer by the reaction

Figure 4.18 Recuperative Brayton Cycle with Single Level Rankine Bottoming Cycle

Figure 4.19 Intercooled and Aftercooled, Recuperative Brayton Cycle with Single Level Rankine Bottoming Cycle

**Figure 4.20 Recuperator Plus Single Level Heat Recovery
Steam Generator, T-Q Diagram**

Figure 4.21 Continuous Evaporative Compressor Cooling

Figure 4.22 Water Evaporation Air Recuperator

Figure 4.23 Water Evaporation Air Recuperation, T-Q Diagram

Figure 4.24 Chemical Fuel Recuperation

T-Q diagrams, Figures 4.25 and 4.26, illustrate the recuperative recovery of exhaust heat by methane fuel reforming. The introduction of H_2 and H_2O into the fuel gas stream may provide some advantage in the design and operation of a low NO_x combustor, but it is not apparent that recuperation by fuel reforming provides any other advantage over air recuperative preheating. Indeed, it seems likely that the heated catalyst bed required for fuel reforming will prove bulkier and more costly than an air heater.

The most straight forward, as well as the most problematic, way to minimize the irreversibilities and the efficiency loss associated with turbine blade cooling is to develop blades using materials, such as ceramics, which may require no cooling. Perhaps this might be most readily accomplished for the first row stator blades. Advanced cooling techniques, using either air or steam, including transpiration or closed circuit internal cooling, as shown in Figure 4.27, might be developed to enhance cycle efficiency.

The reheat gas turbine cycle, shown in Figure 4.28, generally makes use of high pressure ratios and thus involves compressor intercooling. The high air temperature at the inlet of the second combustor requires additional development. Blade cooling is obviously required in both sections of the turbine.

4.3 Performance Table

Calculations were made for 1 lb. mole of air introduced into the cycles discussed in the previous section to estimate the effects of reconfiguration on cycle efficiency. The results of 15 separate calculations are summarized in Table 4.1.

The information in the table was generated using eight spread sheet computational programs [i.e., two for compression, one for combustion, expansion and recuperative heat recovery, one for the three-pressure level Rankine cycle, two for chemical recuperation, and two for turbine cooling (one for air and one for water vapor)].

The efficiency and output estimates are based on material and energy balance, pressure loss, and heat transfer computations for the overall plant burning methane fuel. Assumed pressure losses through the gas turbine and its auxiliaries are based on current experience. Polytropic compression and expansion efficiencies used in numerical integration computations to calculate the work of compression and expansion are likewise based on current gas turbine performance. The evaporative water process for intercooling and aftercooling of the compressed air or for continuous cooling of the compression process assumes that 85% saturation is achieved or maintained. The turbine work computations are based on the turbine inlet temperature and take into account the staged introduction of cooling air at various pressures and temperatures into the stream of expanding combustion gases. The quantity of cooling air is based on current turbine practice extrapolated by film cooling correlations to account for differences in turbine inlet temperature and in coolant -- air or steam -- temperature and properties. The calculation of heat removed in closed circuit cooling of the stator blades is based on estimated external heat transfer coefficients. Assumed isentropic efficiencies for the steam turbine expansions have been selected based on current experience.

Figure 4.25 Fuel Reforming Recuperation, T-Q Diagram

Figure 4.26 Fuel Reformer/Heat Recovery Boiler, T-Q Diagram

Figure 4.27 Closed Circuit Nozzle Cooling

Figure 4.28 Reheat, Evaporative Intercooled, Recuperative Cycle

Table 4.1
Estimated Cycle Performance

	Cycle Description	Cycle PR	Turbine Inlet Temp. (°F)	Cooling Flow Inlet Flow (%)	Cycle Efficiency (%) LHV	Work, Btu LB-Mole Air	Water Use,* LB-Mole LB/Mole Air
1	W501F, 1 Press Level HRSG	14.3	2460	21.4	53.2	6851	(.2666)
2	W501F, 3 Press Level HRSG	14.3	2460	21.4	55.1	7091	(.2710)
3	Increase Temp, 1 Press Level HRSG	14.3	2680	24.0	54.5	7723	(.2972)
4	Increase Temp, 3 Press Level HRSG	14.3	2680	24.0	55.3	7829	(.2874)
5	Increase Press and Temp, 1 Press Level HRSG	20.0	2680	24.0	55.0	7191	(.2534)
6	Increase Press and Temp, 3 Press Level HRSG	20.0	2680	24.0	56.3	7332	(.2604)
7	Ref. Cycle: I/C and A/C, Recuperation, 1 Press Level HRSG	20.0	2680	19.6	57.1	8554	.1927 (.1434)
8	Ref. + Continuous Compressor Cooling	20.0	2680	19.6	60.3	8686	.1478 (.1311)
9	Ref. + Closed Circuit Air Cooling of First Vane	20.0	2680	9.6	61.5	9575	.1927 (.1181)
10	Ref. + No Cooling	20.0	2680	0.0	67.2	10662	.1927 (.0810)
11	Ref. + Evaporative Recuperation	20.0	2680	19.6	61.3	10946	.3927 (.0581)
12	Chemical Recuperation (H ₂ O/CH ₄ = 3)	20.0	2680	26.5	57.2	6893	.1021 (.1409)
13	Chemical Recuperation (H ₂ O/CH ₄ = 6)	20.0	2680	26.5	57.9	8446	.2474 (.0247)
14	Turbine Reheat	64.0	2680	21.6	53.4	11542	.3058 (.1034)
15	Ref. + Cont. Compressor Cooling + Closed Circuit Air + Evaporative Recuperation	20.0	2680	19.6	65.5	11970	.3478 (.0303)

*Note: Water use: evaporation in Brayton cycle indicated by numbers without parenthesis
to cooling tower in Rankine cycle indicated by numbers in parenthesis

The overall efficiency calculations are based on the LHV of the fuel and on air at 59°F and 60% humidity; they include the effects of auxiliary power consumption and generator losses. The work output per pound mole of dry air (28.97 pounds) is given as one measure of plant sizing. The calculated water use in pound moles per pound mole of dry air for evaporative processes in the Brayton cycle and for an associated cooling tower in the Rankine cycle -- in parenthesis -- are given in the table.

From the information presented in the table the following conclusions can be drawn:

- The reference cycle (compression ratio 20; turbine inlet temperature 2680°F; recuperative air heater with an approach temperature difference 50°F; single pressure level Rankine bottoming cycle) has an cycle efficiency of 57%.
- An increase in the turbine inlet temperature to 3000°F or more would be required to reach an efficiency of 60% in the reference cycle,
- Each of three cycle/turbine features can achieve 60% efficiency in the reference cycle without the need of further temperature increase;
 - closed circuit air cooling of the first stage nozzles with return of the air to the combustor, or ceramic (uncooled) nozzles and buckets, or (perhaps) transpiration blade cooling. (Transpiration cooling calculations have not as yet been carried out.)
 - evaporative recuperative air heating -- spraying water into the compressed air stream in the recuperative air heater.
 - continuous, evaporative cooling of the compression process.

The combination of these three features provides an overall cycle efficiency of 65.3% without the need for a bottoming Rankine cycle.

With the cycle configurations and at the operating conditions studied, recuperation by fuel reforming showed negligible efficiency advantage over air recuperative heating. What small increase occurred might well be attributed to increased steam injection at high temperature to the combustor. Gas turbine reheat showed a decrease in cycle efficiency due to the increased cooling air requirement. Further study of cooling in reheat turbines is required to clarify this situation.

4.4 Discussion

Before proceeding further in the selection of the reference cycle, assumptions made in the generation of Table 4.1 were reviewed.

An assumption of 21.4% of compressor inlet air was used for cooling and isolating the discs and casing from the hot gas path air. The pressure ratio and turbine inlet temperature (TIT) were valid. At a 2460°F TIT, the anticipated rotor inlet temperature is 2300°F.

Figure 4.29 shows the definition of turbine inlet temperature and rotor inlet temperature. Rotor inlet temperature is the mass averaged absolute temperature at the first rotating blade leading edge plane. It is lower than the TIT (160°F in the 501F) due to the introduction into the gas path of row 1 vane shroud and aerofoil cooling air, and shroud and disc isolation (leakage) air. Manufacturers generally rate their engines at RIT, not TIT because the first rotating blade is usually the key component limiting the temperature level of firing.

Figure 4.29 Rotor Inlet Temperature vs. Turbine Inlet Temperature

Holding constant RIT and varying cooling air usage will give significantly different results than will maintaining TIT while cooling air consumption is varied. The comparisons in Table 4.1 were calculated at TIT, therefore changes in cooling air directly affected RIT. When cooling air percentage was increased, RIT dropped relative to TIT; when cooling air percentage was decreased, RIT increased relative to TIT.

The calculated cycle efficiencies assume new, clean expected performance and do not include margins for degradation or balance of plant power loads. Therefore, comparison of these numbers to warranted combined cycle plant quotations could be confusing. For instance, the current plant efficiency for a 501F is in the range of 53 to 54%, not the 55.1% shown in Table 4.1, for cycle 2. In that the calculations were intended to show the relative advantages of

cycle reconfiguration, the discrepancy between quoted and estimated baseline values should be ignored. The relevancy of the trends is of primary importance.

Cycles 3 to 15 were analyzed with a TIT increased by 220°F to 2680°F.- Cycles 3 through 6 all assumed 24.0% cooling air (an increase of 2.6% of compressor inlet flow used for cooling) which is reasonable for cycles 3 and 4, where pressure ratio has not changed. Cycles 5 and 6 should have had higher cooling flow because the leakage portion will increase due to the higher expansion ratio through the leakage flow paths. An additional increase of 2.8% of compressor flow would have been more reasonable, i.e., total cooling of 26.8%.

The change in efficiency from cycle 4 to cycle 6, when PR is increased from 14.3:1 to 20:1, as TIT is held, will likely be a decrease of 0.2 percentage points when the increase in leakage air of 2.8% of inlet flow is considered. Increasing pressure ratio increases Brayton cycle efficiency considerably, but the increased cooling air in combination with the PR reduces exhaust temperature so much that Rankine cycle power drops enough to drag the combined cycle efficiency down. Thus, cycle 6 should end up at about 56.1%, quite close to the number arrived at in the preliminary calculations.

The tentative reference cycle, cycle 7, shows a large drop in cooling flow requirement to 19.6% of inlet air flow. This is due to the reduction in cooling air supply temperature with inter- and aftercooling. This cooling air requirement has been reduced too much; rotor cooling air was previously cooled to 392°F and would be only slightly reduced. The two lowest pressure cooling air streams have experienced no change in inlet temperature yet the gas path temperature and moisture content has increased, and leakage air flow has increased for the two highest pressure streams because the cooler air/steam will now flow through the leakage path. A better estimate might be 22.0% cooling, resulting in an efficiency of about 56.6%.

This would mean that an evaporative intercooler and aftercooler, plus recuperation would increase cycle efficiency only about 0.5 percentage points over the traditional configuration.

The very large increase in cycle efficiency with continuous evaporative compressor cooling (cycle 8 vs. cycle 7) can best be explained by comparing Figures 4.9 and 4.10. The work input per pound of air for compression in Figure 4.10 is only 50% of that required in Figure 4.9. This 50% reduction in air compressor work requirement results in more power output, thus the large increase in efficiency shown in Table 4.1 seems reasonable.

Cycle 9 overestimates the impact of closed circuit cooling of the first vane because 10.0% cooling air would not currently be required for this vane at 2680°F TIT (a more reasonable estimate would be 6.5%). In addition, increasing RIT by approximately 200°F and subsequent rows by lesser amounts (130°F extra exhaust) would require additional cooling air to be consumed in the remainder of the turbine. A better estimate might be a reduction of only 3 to 4% in cooling air, with a corresponding cycle efficiency increase of about 1.5 percentage points.

Reducing cooling air to zero, as shown in cycle 10, is completely unrealistic since disc cooling/sealing and shroud seal air will be required, although it does emphasize the penalty explained for cooling air in Figure 4.12.

Cycle 11 (Figure 4.22) compared to cycle 7 presents an optimistic comparison because the cooling flow percentage was held constant in the comparison. Comparing cycles 11 and 7, it can be seen that water use doubled. The impact would be to require significantly more cooling air to maintain similar metal temperatures in the cooled aerofoils because:

- for the same 2680°F TIT, exhaust temperature increased about 125°F, and thus all blades would experience higher gas path temperatures, and
- the additional 20% moisture was added in the recuperator downstream of all cooling air supplies, thus additional heat would be convected from the hot gas into the aerofoils due to the higher thermal conductivity of the high moisture content air, yet the internal cooling medium effectiveness would not change.

The two calculations with chemical recuperation (cycles 12 and 13) were made for the configuration shown in Figure 4.24. Cooling air fraction was increased over that of cycles 3 to 6 due to the additional gas side moisture content (with internal cooling remained as in cycles 1 to 6).

Comparing the efficiencies of cycles 12 and 13 to those of cycle 6 and 7, it may be concluded that chemical recuperation may be more effective in raising efficiency than the in intercooled recuperative (ICR) option of cycle 7. The option of inter- and aftercooling to reduce compression work and cooling air requirement in consideration with chemical recuperation appears to be promising.

The efficiency for cycle 15 is too high since the cooling air requirement will be greater than 19.6% as explained previously.

Intercooling and recuperation as basic changes to the conventional (Westinghouse) utility sized gas turbine appeared to offer advantages to cycle efficiency, when analyzed on a pound mole basis.

4.5 Intercooling, Aftercooling and Recuperation Study

To further assess the effects of intercooling, aftercooling, and recuperation on cycle efficiency, a study was done using actual engine characteristics (501B, 501D5, and 501F), and extrapolating them appropriately.

The analytical procedure is described below:

- The analytical performance was made by hand using the material and assumptions in Table 4.2. The Keenan and Kaye Gas Tables were used and the fluid expanded through the turbine was assumed to be represented by their table called 200% theoretical air with a fuel of H/C ratio of 0.33. The steam side cycle efficiency was assumed to be of the same quality as that would result in a combined cycle of 55% efficiency using the 20:1 pressure ratio at 2500°F TIT.

Table 4.2

Assumptions in Cycle Study

ISO Conditions; Gas Fuel; 2500°F Rotor Inlet Temp.

20:1 Pressure Ratio; 501F Modified Turbine

Coolant Flow = 0.258 of Compressor Flow

Keenan & Kaye Gas Table

Turbine Cooling Flow Effects (i.e., Efficiency for Power and Efficiency for Temperature)

Compressor Efficiency = 0.923 (Polytropic)

Combustor Pressure Loss = 1.0615

Turbine Pressure Ratio = 18:1

Inlet Loss = 2.5 Inches of Water

Exhaust Back Pressure = 16.25 Inches of Water

Recuperator Effectiveness = 0.95 (Temperature)

Recuperator Pressure Loss = 1.02 (Air Side)
1.02 (Gas Side)

Added Recuperator Losses = 2% Flow Leakage & 40°F Heat-up on Passing Air Around the
Transitions

Intercooler = 10°F Approach with a 1.04 Pressure Loss from Low-Pressure Compressor Exit to
High-Pressure Compressor Inlet

Steam Cycle: Carnot Effectiveness = 0.7984 (from 0.55 501F Advanced); Stack Temperature =
240°F and Condenser Temperature = 80°F

Generator Efficiency = 0.987

Mechanical Efficiency = 0.995

- The particular method of determining the steam side efficiency for all cycles was to use a Carnot effectiveness. (The Carnot efficiency is the ratio of the temperature at which heat is added to the cycle minus the temperature at which it is rejected from the cycle divided by the temperature at which it is added to the cycle.) The Carnot effectiveness assumes that one can approach the Carnot efficiency by the same fraction independent of the details of the cycle. Heat is added to the steam side of the cycle between the gas turbine exhaust temperature and the stack temperature. An arithmetic average was assumed since the specific heat does not change too much between these two temperatures. In any case the specific heat is less at the lower temperature and the true weighted average would give a slightly higher temperature level. The heat rejected from the cycle was rejected up the stack and in the condenser. The ratio of the stack rejection to the condenser rejection was calculated by heat balance to be 0.5 and was assumed constant for all cycles. The stack temperature of 240°F was kept constant for all cycles. With an atmospheric temperature of 59°F, the average rejection temperature up the stack would be 150°F. The temperature in the condenser was assumed to always be 80°F. The result is a rejection temperature of 103°F.
- The actual steam side efficiency was defined as that amount of exhaust energy available between the exhaust gas energy and that in the stack divided into the steam side output. For the simple combined cycle this was found to be 0.3395 and the Carnot efficiency was 0.4879 giving an effectiveness of 0.7984 which was used for all subsequent cycles studied.
- The gas turbine was assumed to always have the same expansion ratio such that additional pressure losses in the cycle were made up by increasing the pressure ratio of the compressor side.
- The efficiency of the 501F gas turbine was used. There are two efficiencies. One is used to calculate the power using the ideal enthalpy drop from the combustor outlet temperature to the turbine exhaust pressure and the first stage vane inlet flow. The other is used to calculate the exhaust temperature using the same ideal enthalpy. These efficiencies are a function of the amount of cooling flow. The function was determined by using the 501F, the 501D5, and the 501B cooling flows supplied by the design computer program.
- All turbine calculations were made at a constant turbine rotor inlet temperature of 2500°F. The combustor outlet temperature was calculated assuming the same proportions of cooling flow distribution to the first stage vane.

The results of the analytical study are summarized in Table 4.3. Cycle efficiency (LHV) and net power output from each of the gas turbine, steam turbine and combined cycle are tabulated for 6 different cases.

Table 4.3**Effects of Intercooling, Aftercooling and Recuperation On Cycle Performance**

Case No.	Configuration Description	Power (MW)	Cycle Efficiency (%) LHV
1	RIT = 2500°F PR = 20:1 $G_{cool}/G_{in} = 25.8\%$	GT = 167.8 ST = 84.4 CC = 248.2	55.0
2	Case 1 with recuperator (effectiveness = 95%)	GT = 162.2 ST = 69.4 CC = 231.6	56.5
3	Case 2 plus surface intercooler (10°F approach)	GT = 199.8 ST = 34.0 CC = 233.8	56.4
4	Case 3 plus spray aftercooler (85% of sat). water = 28.7 lb./sec.	GT = 222.4 ST = 22.4 CC = 244.8	53.9
5	Case 4 with spray water heated in surface intercooler water = 37.3 lb./sec.	GT = 229.4 ST = 23.0 CC = 252.4	54.6
6	Case 1 with coolant reduced to 21%	GT = 171.9 ST = 91.2 CC = 263.1	56.0

where: GT = Gas Turbine
ST = Steam Turbine
CC = Combined Cycle
G = Gas

For continuity, the base configuration, case 1, was defined as 2500°F RIT, 20:1 PR to be similar to cycle 6 of the previous theoretical study. A 25.8% cooling air requirement was defined, and 55.0% chosen as a realistic combined cycle efficiency. (The Rankine cycle was defined to achieve this as just described.)

In Configuration 2 the cycle was modified by adding air recuperation only (as per Figure 4.8 and Figure 4.18). With a temperature effectiveness of 95% (possible for high cooling air fractions) in the recuperator, it was found that cycle efficiency increased 1.5 percentage points. Assumed penalties in cycle pressure losses, leakages, and heat pickup across the transitions were realistic. Thus, this calculation confirmed that recuperation was a valid mechanism for increasing combined cycle efficiency at elevated firing temperature.

Configuration 3, shown in Figure 4.30, separated the compressor, and cooled the air leaving the low pressure compressor to 69°F (10°F above cooling water temperature) using surface heat transfer. Interestingly, cycle efficiency dropped 0.1 percentage points from configuration 2's

Figure 4.30 Intercooled, Recuperative Combined Cycle

efficiency although the steam turbine power output was halved. This cycle was analyzed using the same 25.8% cooling air as in Configurations 1 and 2. The actual cooling requirement will decrease significantly due to the reduced cooling air supply temperatures to the rotor and stage 1 and 2 stationary air circuits. Even with RIT held constant, exhaust temperature will increase with the reduced cooling air flow and cycle efficiency will increase.

If cooling flow dropped to 21% then Configuration 6 showed that cycle efficiency may increase 1.0 percentage point. Configuration 3 should be close to 57.5% cycle efficiency, a 2.5 percentage point increase over the base case. This again confirmed that ICR was attractive.

Configurations 4 and 5, with evaporative aftercooling, were analyzed to determine if aftercooling in combination with recuperation makes sense. Pre-heating aftercooler spray, Configuration 5, in the intercooler, as shown in Figure 4.31, was the best of the two alternatives, although cycle efficiency actually dropped 1.8 percentage points from that of Configuration 3. Rotor and stage 1 vane cooling air temperature will be further cooled with aftercooling so the cooling air requirement for Configuration 5 will be further reduced from that of Configuration 3. It is not clear that the reduction in cooling air flow would be enough to raise cycle efficiency to that of Configuration 3. Therefore, the influence of evaporative aftercooling on combined cycle efficiency is not clear, but intercooling and recuperation were confirmed as beneficial to cycle efficiency.

Thus, the selection of a natural-gas-fired reference system configured in an intercooled, recuperative format made most sense. Within this configuration, the addition of closed loop steam cooling of stationary (and possible rotating) hot gas path components could be added, and would significantly boost efficiency, no matter what form of intercooling or recuperation was chosen.

Figure 4.31 Intercooled, Aftercooled (Evaporative), Recuperative Combined Cycle

Figure 4.32 shows the reference system selected by the Westinghouse team to best meet the requirements of achieving 60% LHV efficiency. It was felt that this combination should result in a plant overall efficiency of greater than 58%. Further improvements will result from component improvements, additional cooling improvements, and reductions in leakage/pressure/windage losses. A combination of these improvements should allow cycle efficiency to surpass 60%.

Figure 4.32 Selected ICRCC Reference System

5 Description of Natural Gas-Fired Reference System

5.1 System Description

From the cycle studies evaluated in Section 4, selection of a single shaft, intercooled, recuperative, combined cycle was chosen for the Natural Gas-Fired Reference System. The combustion turbine components will utilize existing Westinghouse engine designs as the basis for modifications to produce a cycle pressure ratio of 20 to 1 with a turbine rotor inlet temperature of 2500 F. The high-pressure (HP) compressor, combustion system, and turbine will be a two bearing rotor in a single housing. The combustion system will be a Westinghouse low NO_x combustor design. The low-pressure (LP) compressor, generator and steam turbine will be individual rotors in separate housings. All components will be directly coupled to form a single shaft running at 3600 RPM. The generator rotor will be double ended so that the steam turbine and combustion turbine can drive from opposite ends. The exhaust flow from the turbine is axial into the recuperator.

A single shaft machinery arrangement has been selected, as shown in Figure 4.32, because it has the following characteristics:

- load pick-up and shedding is instantaneous
- variable compressor inlet guide vanes can modulate the air flow from 70 to 100%
- matching of the components is not as critical to the design performance
- the blading of the compressor and turbine can be tuned away from harmful engine speeds since the shaft operates at only one speed
- critical shaft speeds can be designed away from the operating speed and shaft balancing is simplified, and
- the number of the shaft bearings is a minimum.

It should be noted that for a multi-shaft engine, the power turbine and compressor-turbine(s) are connected only by the working fluid flow. Therefore, anything that affects the flow conditions between the differing shafts also affects the speed between the shafts. The net effect is that during the design phase of the machinery, the losses and performances of all components must be more precisely known or adjustments must be provided such as variable geometry in the turbine component(s). During the operating phase of the machinery, changes such as compressor fouling, blade erosion, wear, etc. affect the engine performance to a greater degree than in the single shaft arrangement.

5.2 Projected Cost of Electricity Improvement

The projected cost of electricity (COE) for the Westinghouse ATS reference cycle is expected to be 10% below the COE of 1991 vintage power generation systems in the same market class and size range, and meeting the same emissions requirements. This expectation is achievable, but is dependent upon continued advances in design, manufacturing, and materials technologies not

only for the gas turbine but all system components, gained both from within the ATS Program and outside of the program.

Assuming a nominal plant size of 228 MW, initial estimates comparing a natural gas-fired combined cycle plant (having a Rankine bottoming cycle and an operating efficiency of about 53%) with the ATS reference plant (with a projected efficiency of 60+%) showed that the ATS plant would use approximately 12% less natural gas than today's vintage combined cycle plant. The fuel cost term of the COE accounts for about one-fourth of the total COE, while the capital charges, fixed and variable operations and maintenance, consumables and by-products charges account for the other three-fourths of the COE. Thus is the challenge for achieving the 10% reduction in 1991 COE levels. With the addition of more system components to improve overall system efficiency, offsets will have to be made in capital costs and reliability, availability and maintainability (RAM) through technological advancements in design, manufacturing and materials.

A thorough evaluation of the COE using the Electric Power Research Institute Technical Assessment Guide procedures will be performed in the second phase of the ATS Program once the exact size and configuration of the ATS is selected.

5.3 RAM Projection for Nth Plant

The RAM of the Westinghouse ATS will be comparable to today's advanced power generation systems. The ATS can be expected to have an overall lifetime availability of 92% and in a worst case year (i.e., year in which the planned maintenance is greatest), availability would be expected to no less than 84%. These figures can be improved upon by employing greater redundancy and backup arrangements.

To achieve 92% average lifetime availability, certain design rules have been established to make today's combined cycles more reliable. The two general rules are as follows: the failure of any active balance of plant equipment item or system shall not cause the loss of the combustion turbine, and all components that provide services to the combustion turbine shall have an installed spare, normally not operated, that shall be capable of restoring 100% of the normal operating capacity in the event of the failure of the operating component.

Westinghouse combined cycle plants are designed to be maintainable with a minimum impact on availability. The steam turbine is capable of being sufficiently isolated within a 72 hour period following the shutdown of the combustion turbine to permit major servicing of the turbine.

The RAM program will be realized by continuous assessment of the design in relation to RAM requirements. The process is achieved through a series of iterations in which the RAM program is applied to the design process. The assessment will be done first at the plant level and then at lower levels as RAM requirements are allocated and equipment designs specified at each level, generating requirements for systems and equipment still lower in the design hierarchy.

The process applies the level of analysis appropriate to each component. All components will initially be screened, using past performance data, checklists and qualitative methods. All equipment which is evaluated as potentially capable of causing significant outage time or maintainability burdens will then be analyzed in more detail. Methods applicable in this phase

are Failure Modes Effects and Criticality Analysis (FMECA), Reliability Block Diagrams (RBD), Fault Tree Analysis (FTA), and event trees. All of these methodologies have been computerized by Westinghouse to increase productivity.

The RAM Data Bases (VISTA, RAMP) will be used to communicate field experience to the design process. RAM personnel assigned to database activities will reduce the data available from these sources down to meaningful reliability parameters suitable for use in the analysis portion of the RAM effort. This includes failure rates, repair times, planned maintenance frequencies and durations, etc.

A main objective of the reliability tasks is to confirm that component failure mechanisms that have historically caused significant outage time or maintenance cost have been identified and removed. New equipment designs will be reviewed with special emphasis to ensure that no new failure mechanisms are introduced. Analyses in this area will be done with FMECA, FTA, and RBD.

The reliability, availability, and maintainability for the ATS program will be modeled after the program that has been used for conventional addition of new products to the Westinghouse product line such as the introduction of the 501F as an extension of the 501D5 technology.

The following is a summary of the process.

- The first step of the typical analysis process, a block diagram is created to describe the top-level functioning of each system and the overall interrelationships of the system components. Each block diagram will illustrate the major component redundancies.
- The RAM Program then defines reliability requirements for components from the allocated availability requirements and from program objectives that limit the frequency of equipment-related unplanned outages. The Reliability requirements will be directed to maximizing the Mean Time Between Failures (MTBF) for components. Major equipment service life will be estimated as the basis for confirming that replacement outages and costs meet the program requirements in support of the Life-Cycle Cost evaluations to be performed within the RAM program. Where possible, actual historical data will be utilized to confirm these estimates and therefore reduce program risk.
- For equipment whose inoperability causes plant full or partial outage, Westinghouse will derive quantitative restoration time requirements from the numerical plant availability and equivalent availability requirements, to maximize the beneficial effects on the overall plant. Westinghouse will also consider constraints originating in the requirements to control operation and maintenance costs, plant staffing, spares inventory and related logistics factors, as defined by the Life-Cycle Cost Analyses.

5.4 Intermediate Development Needs

A preliminary design should be completed for the ATS Reference Natural Gas-Fired Gas Turbine including enough layouts, to determine the physical configurations for components that require testing in Phase II. Detail drawings for test components should be made. The remaining layouts and detail drawings would be completed in Phase III, Testing of Large Scale

Components and Integrated Subsystems. This section discusses the upgraded or new hardware components that are to be considered in the development of the ATS.

Each major area of current designs will be discussed in terms of existing technology and what barrier problems must be addressed and solved by development programs to achieve the full benefits of the ATS program.

Briefly, the component areas of the Westinghouse engines that must be considered to meet the full objectives of the ATS program are:

- Low Pressure Compressor
- High Pressure Compressor
- Combustion System
- Turbine
- Aero-thermodynamic
- Cooling
- Materials

5.4.1 Low Pressure Compressor Requirements

The compressors for the ATS program will come from the highly efficient and reliable Westinghouse 501 compressor family that have demonstrated performance over many hours of utility and industrial applications.

- Blading

The LP compressor will utilize the first eight stages of the 501F compressor that has demonstrated a polytropic efficiency at the design point of greater than 93%. A new exit guide vane row, diffuser and scroll system will be designed to efficiently guide the air to the intercooler and return the cooled air to the HP compressor inlet bell mouth. Proven modifications from the 501F compressor, as it matures, will be incorporated in the ATS compressors (i.e., controlled diffusion blading, end bend airfoils, abradable tip and interstage seals, etc.) as they become available, obviating the need for separate major development programs.

- Rotor

The rotor will use the 501F design concept modified on the exhaust end after the eighth stage to add a bearing and seal system so that the rotor is self contained. A coupling will also be required on the exhaust end to directly connect to the HP compressor.

- Cylinder

Modifications to the 501F cylinder will be required to add an exhaust diffuser - scroll combination to direct the air to the intercooler. The design of this element and the seal system is important to minimize cycle losses due to seal leakage and pressure losses. An interstage bleed may be required in the cylinder to tap off air for turbine cooling which will be determined in the

design phase. A support will also be required at the exhaust end to make the LP compressor self contained.

For starting of the ATS, a combination of the variable LP compressor inlet guide vane and an air bleed off the exhaust of the LP compressor is sufficient.

Due to the importance of minimizing cycle losses, model tests should be conducted to assure the design of the LP compressor exhaust, piping to and from the intercooler, and the inlet to the HP compressor. Description of this test program will be outlined in Section 5.5.2, Critical Component Design and Testing, Development Task 1.

5.4.2 High Pressure Compressor

- **Blading**

The Reference Cycle selected for the ATS program incorporates an intercooler between the LP and HP compressors. The intercooler reduces the work of compression, and also reduces the volume flow through the HP compressor. The compressor blading must be designed to accommodate these different thermodynamic conditions. The exit flow from the HP compressor is approximately 62% of that of the Westinghouse 501D5 compressor and about 52% of that from the 501F compressor. The HP compressor blading will be comparable to an approximate 0.8 scale of the last 14 stages of the 501D5 compressor. To transfer the high efficiency of this compressor to the HP compressor for ATS, the aerodynamic loadings will be kept low. Clearance and leakage will also be kept to a minimum.

- **Rotor**

With the blade path being approximately an 0.8 scale of the 501D5 rotor, the discs and rotor assembly, including an inlet end bearing and thrust bearing, will be redesigned. Also, the connection to the 501F turbine through the torque tube will need to be modified. The HP compressor and turbine will be a single shaft, two bearing system. With the first eight stages of the compressor now a separate LP compressor, the axial space between the HP compressor and turbine can be lengthened to provide extra space for HP compressor air to exit to an aftercooler and return from the recuperator to the combustion system. Because of these changes, a complete HP compressor and turbine rotor system analysis will be required.

- **Cylinder**

The cylinder will be redesigned to include a HP compressor inlet air scroll - support system and one or two bleed ports for turbine cooling air extraction. It is expected that this cylinder will utilize blade rings similar to the 501F design. Also, modification will be required for the section of this cylinder that forms part of the combustion cylinder.

5.4.3 Turbine Requirements

The aero-thermodynamic requirements for the turbine component of the ATS program can best be accommodated by modifications to the Westinghouse 501F turbine component. This turbine is currently operated at a RIT level of 2300°F, but was designed for temperature growth. As

cooling and material technology advancements are incorporated, the temperature level capability of the turbine can efficiently be advanced. For the ATS program, the requirement of a higher temperature drop through the turbine (due to the higher temperature level and larger pressure ratio) can be accomplished by closing the front end of the turbine and modifying the blading as indicated.

The blading will be redesigned with three-dimensional design techniques where required. Special attention will be focused on cooling and sealing flow management to minimize cycle losses.

The 501F turbine is a conservative four-stage design that can accommodate efficiently the increased work requirements resulting from cycle improvements such as ceramic materials incorporation. The full applications of ceramic elements such as thermal barrier coatings on ring segments, blading, and improved sealing control will result in a very significant improvement in energy conversion efficiency.

- Turbine Cooling Development

Advanced materials construction and cooling techniques will be required to accomplish an increase in RIT from 2300° to the 2500°F range. In the 501F the stage 1 vane, a cored casting, cooled by convection and impingement augmented by film, is near the limit of the technology currently employed for air cooling of utility type combustion turbines. The length and high degree of twist of downstream airfoils render difficult the incorporation of internal passages, particularly if cooling is required near the leading and trailing edge. The upstream endwalls, currently cooled by relatively unsophisticated techniques, will require new approaches.

Westinghouse has considered several approaches to increasing cooling. One of the most promising of these is a fabricated Shell/Spar construction. Shell/Spar vanes and blades are formed by bonding a thin non-porous airfoil shaped metal sheet (the shell) to an inner hollow cast support member (the spar). Cooling air for the assembly is supplied from alternate cavities within the spar (the supply cavities). It flows along chordwise convection cooling channels between the spar and the shell and returns to the remaining spar cavities (the discharge cavities). The heated or spent cooling air is then routed from the discharge cavities to the main gas stream through exit holes in the shrouds or airfoil.

Another means of increasing cooling with a relatively minor modification to 501F hardware is to apply a thin (10 to 30 mils) ceramic coating to the hot gas side of the blade and vane surfaces, known as a thermal barrier. In highly cooled upstream rows a substantial temperature drop is obtained, permitting the metal to operate at a reduced temperature.

Another method of obtaining increased cooling levels with 501F technology is to employ high pressure steam, available in a combined cycle, as a coolant. Although high levels of cooling can be achieved with this approach, minimization of parasitic leakage in the course of ducting the steam through the engine to the airfoils could present a problem. The use of steam in place of compressor discharge air for cooling leads to increased power output and possibly an efficiency advantage. A steam supply/control system would have to assure an uninterrupted supply of coolant. In addition to more effective cooling, more heat resistant materials and coatings,

including directionally solidified and possibly single crystal alloys, may be introduced that will reduce the amount of incremental cooling required for the higher firing temperature.

Spanwise holes which taper from hub to mid height provide an improvement over the constant diameter holes used in rotor blade cooling. With this technique, increased cooling in the critical mid height region is obtained by reducing over cooling in the hub region resulting from the lower coolant temperature as the cooling air enters the hub region. Increased hub region flow area obtained by tapering reduces heat transfer in that region, thus reducing temperature rise in the coolant as it flows radially outward.

Another variation in cooling hole geometry is to integrally cast circumferential ribs periodically spaced along the length of each cooling hole. These ribs act as flow turbulators to produce turbulent rather than laminar heat transfer for the cooling air. Field tests have been run on a 501 engine comparing smooth and turbulated holes. These tests verified that the heat transfer coefficient was doubled by the use of the turbulators.

A combination of turbulators and tapered holes could provide a very significant improvement in the cooling effectiveness required for the first stage rotating blade and possibly the second stage rotating blade at 2500°F.

- Rotor

As previously stated, the part of the turbine rotor that extends to the HP compressor will have to be modified. The remainder of the rotor, including the rotor cooling supply system is expected to be the same as the 501F.

- Cylinder

Separating the compressor into two sections, LP and HP compressors, permits some lengthening of the combustion zone without a major effect on the rotor dynamics. The additional axial length is expected to create sufficient room to incorporate the supply/return circuit to be within the confines of the engine longitudinal. Sixteen (16) combustors will be used. Both the compressor, and turbine cylinders will be enlarged to provide sufficient space for the extraction flow and return flow piping. Extensive compressor-combustor cylinder modification will be required to provide sufficient space for even flow distribution to the 16 combustors. In addition, the design for separating and sealing the compressor discharge air to the aftercooler from the air returning to the combustion system will be studied. One approach would be to follow a past Westinghouse design of adding insulation internally to the combustion cylinder and seal wall between the compressor exit and the combustion turbine cylinder. Heat transfer analysis would be conducted to determine insulation requirements that will insure control of the cylinder metal temperature in the combustor region to match cylinder temperatures in the compressor discharge area.

The critical structural problem is thermal stress and distortion caused by the temperature difference between the compressor and combustor legs of the compressor-combustor cylinder and the sealing design between compressor discharge air and recuperator return air. An extensive finite element heat transfer and stress analysis will be required to redesign both cylinders and flanging so that stress and distortion limits are met.

Because of the 20 to 1 pressure ratio, 2 1/4 Cr-1Mo steel will probably be adequate for the cylinders when operating at the higher RIT of 2500° F. Horizontal joint and vertical flange bolting may have to be changed to X-750 or INCO 718 to provide sufficient strength against stress relaxation.

The heating of the compressor discharge air as it flows around the combustor transition pieces and through this part of the housing represents a loss in the cycle, as this reduces the heat pickup in the recuperator. While some cooling is necessary for the transition pieces, the use of a thermal barrier should reduce this requirement. Further, a thermal barrier may be of benefit along with insulation in the combustion area.

Turbine development should consist of some component testing along with design improvements and material development. Turbine testing consists of a static test rig to determine the swallowing capacity of the first row vane. This program will be outlined in Section 5.6.2, Phase II - Critical Component Design and Testing, Task 3. The design improvements for sealing and clearance control will be discussed in Section 5.6.3 Phase III - Testing of Large Scale Components and Integrated Subsystems where the remainder of the engine design tasks will be completed.

5.4.4 Combustion System Requirements

The emission goals for the combustion system are to achieve <10 ppm NOx and <20 ppm CO in a dry combustor. It has been demonstrated, with existing gas turbines in the field, that the use of a system that pre-mixes the fuel and air, prior to combustion, will allow combustion to occur at temperatures below the threshold level for NOx formation. Such a pre-mixed lean combustion system will, in itself, produce zero NOx. An alternative combustion technology that will also, in itself, produce zero NOx is found in the use of a catalytic reactor within the combustor.

Both of these approaches require pre-mixing of the fuel and air, with staging of the fuel and air as the combustion system is brought from light-off up to idle and then full load of the turbine. Staging of the fuel is much easier to achieve than staging of the air, which implies variable geometry of the combustion system.

Both of these systems require a pilot burner to assure that flame blow out will not occur at any point in the operating range. The pilot burner becomes the source of any NOx produced by the combustor, and hence the amount of fuel burned in it must be minimized to the greatest extent possible.

Both of these systems will benefit from the higher temperature air (1070°F) entering the combustors from the recuperator. In the case of the catalytic combustor, this higher air temperature will enhance the reactivity threshold of the catalyst for natural gas fuel. Consequently, the amount of pilot burner flame required may be very minimal. In the case of the lean pre-mixed combustor, this higher air temperature should broaden the flammability range, again helping to reduce the amount of fuel burned in the pilot burner.

The burning of additional fuel, which would normally be required in going from 2300°F to a 2500°F turbine RIT, may not be required because of the higher air temperature entering the

combustor. The expected burner outlet temperature (BOT) for a 2500°F RIT is 2680°F. For the 2300°F turbine RIT, without intercooling or recuperation, the measured BOT is 2460°F. Thus, the delta temperature rise across the combustor, for a 700°F compressor outlet temperature, is 1760°F. For the ATS, the delta temperature rise across the combustor is 1610°F. Thus, more air should be available for wall cooling and turbine inlet temperature profile control.

Design and development studies to configure the reference dry low NO_x system into the new turbine cylinder, with the higher temperature air from the recuperator, must be carried out. The fuel control system must be adapted to the additional requirements, such as a load dump, of having the extra energy storage in the piping and recuperator.

5.4.5 Rotor System

As illustrated in Section 5.1, System Description, all components, LP compressor, HP compressor-turbine, generator, and steam turbine are connected together as a single 3600 RPM shaft system. Analysis will be required to insure that the supporting structures of all components will maintain proper shaft alignment and satisfactory rotor dynamics.

- **Bearings**

The bearings are expected to be the same as used in the 501F engine for the HP compressor-turbine rotor. The size of the LP compressor rotor bearings may be dictated by the shaft size necessary for transmitting the load (torque) to the generator.

- **Thrust Management**

Analysis of the unbalanced thrust in the combustion turbine will determine the method of thrust management. Since the HP compressor is reduced in diameter relative to the 501F compressor, additional unbalanced thrust will result. Some of this additional thrust can be favorably offset between the LP compressor exhaust and the HP compressor inlet. In summary, there may be enough increase in unbalanced thrust to exceed thrust bearing capability. If this is so, a balance piston would be required. It is expected that the thrust bearing will be located at the inlet to the HP compressor. This will minimize the axial growth for the seals in this region.

- **Seals**

The air and oil seals at the low pressure ends of the compressor and turbine will be similar to current Westinghouse designs. The design of these seals between the LP compressor and HP compressor is critical to minimize leakage losses.

5.4.6 Intercooler Requirements

The reference cycle incorporates an intercooler between the LP and HP compressors. Since the air will be cooled below its dew point in the intercooler, condensation will occur. The quantity of condensation will depend upon the moisture of the incoming air and the temperature and pressure level in the intercooler. At ISO conditions of 60% relative humidity, the condensation will be approximately 21 gpm. On the most humid day with cold cooling water the condensation could be as high as 130 gpm.

This condensation not only represents a small thermodynamic performance loss, but will require a mist eliminator following the intercooler to prevent moisture erosion of the HP compressor blading.

5.4.7 Recuperator Requirements

Preliminary sizing of the recuperator has been performed based on a projected set of operating conditions. The recuperator must achieve an effectiveness of .942 with a combined pressure drop of no more than 3.45% at an exhaust flow rate of about 1,146 lbs/sec. The high effectiveness, low pressure drop, and high flow rates dictate the size of the design. These requirements can be met by a system of eight counterflow heat exchangers, each handling one eighth of the total flow of air and of gas. The heat exchangers will be constructed of stainless steel and be of a plate-fin design. The approximate weight of this system will be 285,000 lbs. not including the required ducting, manifolds, and support structures. The gas inlet face area will total approximately 1,665 ft². The cores of each of these eight heat exchangers will have a flow width of 11.8 ft., a flow length of 15.5 in., and a stack height (no-flow direction) of 14.7 ft. Additional analysis will be required as the component performance requirements are further defined and as additional fluid property data is obtained during preliminary design.

5.4.8 Piping/Control Valving Requirements

The use of a recuperator requires that the air from the HP compressor be discharged through large pipes connected to the discharge area of the compressor cylinder. This will require up to four pipes, and the associated penetrations of the cylinder above and below the horizontal joint. Similarly four pipes, even larger in diameter because of the heated air returning from the recuperator, must be connected to penetrations in the turbine cylinder in the vicinity of the combustion system.

The temperature will be about 400°F in the piping to the recuperator and about 1100°F on return to the combustors. This is within the range of existing piping material, but development design must deal with three areas: the piping arrangement to reduce and control thermal expansion stresses and piping reactions on the turbine cylinder; the requirement for a by-pass valve, if it is needed to prevent turbine overspeed in the event of sudden load loss; and, minimizing of piping pressure losses.

Engineering studies will be required to determine the best arrangement for piping to and from the recuperator. For example, evaluations will be made to determine whether the individual pipes should be combined into one pipe to and from the recuperator or should they be kept as separate pipes. In addition to the control of thermal expansion stresses and piping reactions, the studies must consider minimizing piping pressure losses as these losses have an adverse effect on cycle performance. Because of the complexity of the piping arrangement, few pipes or several pipes, and the balancing of the mechanical design and flow requirements, trade-off studies will be made.

In addition, control valve(s) will be required for control of the ATS during sudden loss of generator load or loss of firing temperature. The location of these valves will be considered in the study.

5.4.9 Control Concept

Conventional combustion turbines have the following primary control system:

- Fuel control for ignition and acceleration to synchronous speed. This includes the control of valving for the fuel staging in the pre-mix combustion system.
- Air flow control during startup, loading and unloading and shutdown.
- Speed control for synchronizing, on-line at load, and upon loss of generator load. In addition to these primary controls, there are extensive secondary control systems.
- Logical controls for startup/shutdown and loading sequencing of main and auxiliary units.
- Limiting controls for load and temperature.
- Protective controls in case of failure of any of the primary, logical or limiting control systems.

The purpose of the primary control system is to manage the mass and energy flows of the plant. The advanced turbine system heat and flow calculations for the zero to full load range are the necessary basis for control system design. Assuming that the recuperator will be a large air mass/energy storage component, it is likely that control valves and a control strategy will be a major requirement for transient control of this mass and energy. Both the primary and secondary control systems will require development to accommodate the needs of the advanced turbine system.

The following discussion of plant components, including the recuperator, contains assumptions in advance of the availability of design heat balance calculations. For the primary control purposes listed above, the following final control elements would be considered.

- Steam Turbine Throttle Valve

A full-arc admission steam turbine is assumed, with a throttle valve used for controlling steam flow when the generator load is dumped. During startup and loading, the valve would be fully open unless used for steam turbine warm-up flow control.

- Combustion Turbine Fuel Flow Control System

The fuel flow control system controls fuel flow for ignition, acceleration to synchronous speed, synchronizing, and loading to maximum load for the plant ambient conditions. When the generator is tripped off-line, fuel is controlled to limit overspeed, resynchronize and reload. During overspeed, fuel is low-limited to prevent flame-out. Determination of the minimum fuel requirement may require testing because of the affects of the temperature and flow of the air coming from the recuperator. This is discussed further below.

- Inlet Guide Vanes

Inlet guide vanes control LP compressor inlet air flow, for the same purposes associated with conventional cycles.

- Intercooler

The intercooler cools air coming from the LP compressor to the HP compressor. Loss of cooling water may necessitate tripping the unit because of excessive HP compressor inlet temperature and pressure.

- Compressor Bleed Valve(s)

These valves are open during startup to prevent compressor surge. The valve inlets may be located after the intercooler or in either or both compressors.

- Recuperator

In order to arrest the acceleration of the rotor system upon sudden loss of generator load, the total power input to the rotor must be immediately reduced to the sum of the compressor power, plus the unit's total bearing and windage losses. For load dumps from the higher load ranges, the recuperator remains a source of considerable power input as the pressure and energy of its stored air mass is dissipated by continued flow into the combustor and turbine. If only the fuel to the combustor is reduced by action of the turbine's speed governor, the rotor may still continue to accelerate. Therefore, the input energy control system design will need to be evaluated and incorporate some of the following actions as means of power input reduction:

- Reduce fuel flow to well below the normal value for synchronous idle. This may include control of combustor air inlet valving to prevent flame-out even as the fuel flow is reduced to a pilot level.
- If there is still excess power input to the turbine after the combustor's air input flow is stratified and fuel input is reduced to a pilot level burned in a fraction of the total air flow from the recuperator, the next step could be the opening of a large recuperator bypass valve to mix lower temperature air with the air exiting the recuperator.
- Instead of adding only a normally closed bypass valve the bypass valve could be made a part of a three port diverter valve. The bypass input port would control flow from the compressor outlet, the second inlet port would control flow from the recuperator output and the single outlet port would pass mixed flow to the combustor. This valve would operate with a low pressure drop and would act to reduce the flow of the hot air from the recuperator as the bypass flow increased. No coordination of separate isolation and bypass valves would be required.

After overspeed is controlled following a generator load dump, the recuperator's stored energy and air mass would be dissipated as it approached its normal state at synchronous idle. At this point, the throttle valve would have a controlled reopening in conjunction with the controlled

increase of fuel flow to its normal synchronous idle value. Coordination of fuel flow with increasing air flow can probably be accomplished by the normal action of the speed governor.

With the availability of the no-load to maximum load thermodynamic calculation results, the need for any new control elements including those discussed above, would be determined. Some transient response calculations may be required, but mainly steady-state information should suffice. It is doubtful that hardware testing will be required to determine control system requirements. An exception could be for combustor minimum fuel requirement as a function of air inlet flow conditions. Specifications for new control components found necessary, such as high temperature dump and throttle valves, would be developed and submitted to vendors for their evaluation and price quotations.

5.5 Development Program

5.5.1 Phase I - System Scoping and Feasibility Studies

The work presented in this report constitutes the effort expended the Phase I of the ATS Program. Primary emphasis was put on the evaluation of different combustion turbine system configurations to arrive at the most feasible options for achieving 60% plus system efficiencies. From these options a preferred selection was made based on being able to achieve full-scale demonstration in the year 2000. It must be noted that further analytical study during Phase II conceptual design may dictate that the selected ATS cycle be modified to incorporate modified or different system components.

5.5.2 Phase II - Critical Component Design and Testing

As discussed in Section 5.4, a conceptual ATS design should be completed in Phase II to support the development of the advanced cycle and gas turbine components. Based upon the completion of this task there may be additional hardware components that should be developed in the early stages of this proposed multi-year program. Based upon current available information there are several tasks that Westinghouse believes should be initiated in Phase II.

Specific turbine hardware components and technology areas that should be investigated in Phase II include:

- LP Compressor Diffuser Scroll and HP Compressor
- Turbine Cooling
- Row 1 Vane Swallowing Capacity
- Shell/Span Design and Construction of Row 1 Vane
- Thermal Barrier Coatings
- Investment Casting for Vanes and Blades

Each recommended development program presented in this section contains the background of the technical concern, the objective of the proposed task, a technical approach, work statement, and schedule. Each proposed task can be used as a separate independently proposed work effort.

Development Task 1 Low Pressure Compressor Diffuser Scroll and High Pressure Compressor Inlet

Background

The reference cycle for the ATS power plant utilizes an intercooler between the LP compressor and the HP compressor. The machinery arrangement requires that the space between the two compressor cylinders be as short as possible to minimize overall length of the rotors. The overall plant performance is directly effected by the amount of pressure loss that results from having air from the LP compressor cylinder exhaust to the intercooler, and re-enter the HP compressor after cooling. There must be a minimum of flow distortion in this inlet flow to the HP compressor.

Objective

The objective of this component test program is to verify that the exhaust diffuser and out flow of air to the intercooler piping has minimum pressure losses, and to further verify that the return flow to the HP compressor inlet has no flow distortions that would adversely effect the efficiency of the HP compressor.

Technical Approach

The engineering design of the LP compressor discharge diffuser and surrounding cylinder design to the discharge piping will be designed using aerodynamic rules for the diffuser and cylinder discharge space. Similarly, the inlet scroll of the HP compressor will also be designed for adequate flow area and acceleration into the compressor blade path.

To verify that the chosen design geometry is going to meet the objective for this hardware, model flow tests of the flow paths involved will be designed. The first model flow tests will include the LP compressor outlet guide vane, diffuser, and exhaust scroll to the outlet piping connections on the cylinder. The second flow model will include the return piping connections to the HP compressor cylinder, the inlet scroll and the inlet guide vanes.

Both of the models will be a scale of the initial prototype design. The models will simulate the prototype Mach number at a reduced Reynolds number. The models will be constructed of wood and/or other suitable low temperature materials. The air flow required will be approximately 10 to 15 lb./sec., at 2 atm and low temperature (less than 200° F). The flow will be blown or drawn through the models and the pressure losses and flow distortions measured. Changes will be made to the models where indicated by the tests and made to the prototype design.

Work Statement

1.1 Define Test Program

Define test program for the LP compressor exit and the HP compressor inlet flow models. The test conditions for the flow models will be defined and test conditions specified.

1.2 Design of Flow Model Test Rigs

The two flow test rigs will be designed as subscale models of the LP compressor exhaust and cylinder geometry and the HP compressor inlet cylinder and scroll geometry. The models will be matched to an existing air flow supply.

1.3 Build the Flow Model Test Rigs

The flow model test rigs will be built with suitable instrumentation and installed into the air flow supply test facility.

1.4 Complete Air Flow Model Rig Tests.

The test program plan will be completed, and the data reduced and organized for review.

1.5 Evaluate Air Flow Model Rig Test Results

The data from the rig tests will be compared to the aerodynamic design objectives of the LP compressor and HP compressor performance to determine if the design of the full scale hardware is adequate or must be modified.

1.6 Task Report

Prepare a topical report on the results of LP compressor diffuser scroll and HP compressor inlet tests and prepare a preliminary design based on available data.

Schedule

Task 1 can be completed in 12 months.

Development Task 2 Turbine Cooling Technology Development

Background

The cycle choice for the ATS system will initially operate at 2300°F turbine RIT. The 501F is currently being marketed at this temperature level. Thus the existing cooling technology in this engine is completely acceptable for the 2300°F cycle. The growth version of the chosen cycle will operate at 2500°F turbine RIT. Improvement in the cooling technology will be required for the vanes and blades in this growth version.

Objective

The objective of the turbine cooling technology development is to achieve reliable long life cooled turbine components for a 2500°F RIT cycle.

Technical Approach

Advancements in materials, construction and cooling techniques will be required for the 2500°F RIT cycle.

Work Statement

2.1 Cooling Design Studies

Heat transfer studies will be conducted which lead to selection of a cooling configuration for all turbine rows including the endwalls (shrouds). For the Row 1 vane, this will include both a Shell/Spar and a thermal barrier coated design.

For the Row 1 blade, introduction of a new proprietary Westinghouse material will decrease the amount of incremental cooling required. More importantly, elimination of the transverse grain boundaries will tremendously increase transverse thermal-mechanical fatigue resistance and resistance to environmental embrittlement and possible transverse fracture when coating failure occurs. The incremental cooling required is expected to be achieved by the incorporation of enhancements to the existing design. Because the life of these parts will be largely governed by degradation from the environment (oxidation), consideration will also be given to the application of a thermal barrier coating to increase life by lowering the temperature of the protective MCrAlY bond coating, as well as temperatures of the internal passage surfaces, which may also have to be coated by a chemical vapor deposition process for protection from oxidation. However, no credit will be taken for the thermal barrier coating in the design of the cooling scheme, (i.e., the thermal barrier coating will not be relied upon to meet minimum creep and oxidation life requirements). In addition, the effect of thermal barrier coatings on turbine performance will be evaluated and considered in a life cycle cost analysis of the thermal barrier coating option.

For the second stage, it is probable that configurations employing convection/impingement/film cooling of the type currently used in the 501F can be used to

meet design requirements. Use of the new proprietary Westinghouse alloy will diminish somewhat the incremental cooling required for the blade, however, as for row 1 this will be governed by coating limitations and hence surface temperature, not bulk temperature which governs creep.

Various approaches to cooling stage 3 airfoils will be identified and the most promising selected for development effort, as required. Substitution of the new proprietary Westinghouse material for the row 3 rotating blade would offset to a considerable extent the need for additional cooling of this row. If this proves not to be feasible for this size blade, it will be conventionally cast, which will require considerably more cooling. A special concern here is shroud creep because of the high bending stresses associated with shrouds and the difficulty in cooling them. This will be investigated in detail and impingement cooling from the stationary shroud (ring segment) employed, if necessary.

IN-738 or the new material in conventionally cast form will be substituted for the current bill-of-material alloy U-520 in the last stage rotor blade. Since IN-738 and the new alloy have approximately 60° and 95° F better creep resistance than U-520, respectively, it is expected that cooling of this row will not be required. Longer range increases in firing temperature may require casting of this row, as well. Another potential solution is light weight titanium aluminide exothermic dispersion strengthened alloys. The last stage vane segment will be cooled in a manner similar to that currently employed in the third stage vane segment, if required.

Steam cooling studies will also be undertaken as part of this task.

2.2 Definition of Development Programs

Development program tasks include:

2.2.1 Bench Scale Heat Transfer Tests

There may be uncertainty in heat transfer and flow information for some features of the row 1 blade cooling design. Following past Westinghouse practice, the required data will be obtained at low temperature and atmospheric pressure on scaled up configurations to effect Reynolds number simulation of the design features being tested. Heat flux is applied electrically. Measurement of electrical power, metal and gas temperature and pressure will supply the desired information.

2.2.2 Stage 3 Development

An approach will be developed for manufacturing of cooled stage 3 parts.

2.2.3 Tapered Hole Development

Problems in the manufacture of tapered holes in cast rotor blades will be investigated. It is probable that tapered glass cores can be used, as constant diameter glass cores are currently used, in the manufacture of cooled cast rotor blades. For a given constant diameter core, the tapered core should be stronger and less susceptible to breakage during the casting process.

2.2.4 Turbulated Cooling Hole Development

Development of turbulated cooling holes will be conducted with the tapered hole development in 2.2.3.

2.3 Detail Design

Heat transfer input to multidiscipline engineering efforts for design, drafting, and manufacture of all cooled parts will be provided utilizing results as required from the various development programs. Input will also be provided for design of the engine cooling system. In addition to external piping, this involves design of the internal flow path which routes coolant throughout the engine.

2.4 Row 1 Vane Verification

In the past, Row 1 vane temperatures have been verified by test at full pressure and temperature of instrumented full scale production hardware in a cascade incorporating four vanes. These tests have consistently verified that the tools available for the calculation of both hot side and cold side heat transfer using bench scale testing, where applicable, are adequate. The test rig is not designed for the higher pressure level envisioned for the ATS machine. Considering the success of past design predictions and the high cost of a new facility, a program of cascade testing of the Row 1 vane is not considered cost effective.

2.5 Task Report

Prepare a topical report on the turbine cooling technology task.

Schedule

Task 2 can be completed in 18 months.

Development Task 3 Row 1 Vane Swallowing Capacity

Background

It has been learned from experience with previous engines that the flow swallowing capacity of the row 1 vane cannot be determined precisely from drawing dimensions and quality control checks of a machined vane. The effective thrust area is a function of not only the geometric configuration, but also the interaction of the main flow entering the vane with its associated boundary layer and with the cooling air being discharged along the suction and pressure surface of the vane. Thus, there must be a flow calibration test carried out on a sector of the row 1 vane assembly.

Objective

The objective of the row 1 vane swallowing test will be to determine the flow coefficient of the row 1 vane at the design mach number in a cold flow rig.

Technical Approach

A full scale annular sector of the prototype row 1 vane assembly will be installed in a specially designed flow test rig that has calibrated flow measuring nozzles to determine the exact air flow passing through the vane assembly as a function of the vane pressure ratio. The sector of vanes will include the prototype cooling configuration, and the blowing rate of the cooling flow to mainstream flow will be simulated. The temperature ratio of the cooling flow to main flow will be approximated. A minimum of at least four vanes must be used to make up the sector in the test rig. The air flow requirements will be approximately 20 lb/sec., at 2.5 atms, and about 900°F. The cooling flow will be at a similar pressure, but at a temperature of about 100°F.

Work Statement

3.1 Definition of Test

Based on the final cycle conditions established in Phase II, the test requirements for the row 1 vane swallowing test will be specified. Both the main stream flow and the cooling air flow and the operating pressure ratio and stream temperatures will be established for the test rig. The number of vanes to be included in the sector will be specified. The amount and quality of instrumentation will be determined. The number of test vanes will be specified.

3.2 Design and Build Row 1 Vane Swallowing Test Rig

Based on the specification for the test rig developed in 3.1, a rig will be designed to accommodate the full scale row 1 vane segment, and adapted to the air supply test facility. Instrumentation will be located in the test rig system.

The vane swallowing test rig will be manufactured, and set up in the test facility. Instrumentation will be installed and checked out.

3.3 Run the Row 1 Vane Swallowing Test Rig

The test agenda for the swallowing test will be run, and data verified as complete and correct.

3.4 Evaluate

The test data will be reduced and compared to the required values for the prototype row 1 vane swallowing capacity to meet the engine pressure ratio requirements. If the design goal is not met, the row 1 vane must be re-staggered to correct the deficiency and be re-tested. This may seem time-consuming, but it is much more cost effective to determine the flow swallowing capacity by rig testing than to learn about it when the prototype engine is operating. Then the correction process is very costly.

3.5 Task Report

Prepare topical report on the row 1 vane swallowing capacity task.

Schedule

Task 3 can be completed in 14 months.

Development Task 4 Shell/Spar Design and Construction of Row 1 Vane

Background

In a joint program with EPRI, Westinghouse and the Allison Gas Turbine Division of General Motors Corporation, the design and fabrication techniques for this unique construction of a turbine vane was established. A shell/spar vane is formed by bonding a thin non-porous airfoil shaped metal sheet (the shell) to an inner hollow cast support member (the spar). Cooling air for the assembly is supplied from alternate cavities within the cast spar to chord wise cooling channels between the spar and shell. The spent cooling air is then routed from the discharge cavities through exit holes in the shrouds or airfoils back into the main gas stream.

Objective

The objective is to design a shell/spar first stage vane for the 501F turbine and develop the necessary manufacturing technology to produce such a vane suitable for operation at 2500°F RIT.

Technical Approach

A very solid base of shell/spar design and fabrication technology was developed in the EPRI/Westinghouse program. The first stage vane for the Westinghouse 251B12 turbine was chosen as the candidate in this program. All of the tooling required to produce the spar, shell and shrouds was designed and built, as well as the encapsulation techniques for the diffusion bonding of the shell to the spar.

This existing base of design and fabrication technology will be the starting point for producing a first row shell/spar vane for the 501F. The physical size of the 501F vane is not greatly different in size from the 251B12 vane designed in the previous program.

Work Statement

4.1 Detailed Design of the 501 First Row Vane

From the cycle data for the ATS, the inlet gas stream temperature distribution will be defined for the vane, as well as the hot spot temperature. From these data, the maximum surface metal temperature for the vane will be determined. The design goals for high-cycle and low-cycle fatigue and the blade life will be specified. The inner and outer shrouds of the vane will also be of shell/spar construction. These end walls and the vane itself will be fabricated as three separate pieces, which will then be fabricated by brazing into a complete vane assembly. The aerodynamic design of the vane and the cooling flow circuitry will be based upon the design experience gained in the EPRI program for the 251B vane. Similarly, the cooling flow circuitry for the inner and outer shroud will also draw on this design experience.

4.2 Material Selection for the Shell/Spar Vane

The design factors that must be considered in the selection of the shell material are corrosion resistance, metallurgical stability, formability, rupture strength and coefficient of thermal expansion. Several candidate shell materials are available. Low cycle fatigue properties impose an additional requirement.

The factors to consider in the selection of the spar material are weldability, castability, corrosion resistance and thermal fatigue resistance. Alloys available and used in gas turbines today will be prime candidates for the spar.

With the clean fuel environment, the primary purpose of a protective coating on the vane surface would be for oxidation resistance, and a suitable one will be chosen from the many available candidates. A prime reason for the need for oxidation protection of the shell material is because it is very thin material (.020 in.), and cannot tolerate surface damage due to oxidation.

4.3 Fabrication Development for the Shell/Spar Vane

Success of the shell/spar technology depends on the ability to achieve a quality diffusion bond between the shell and spar. To achieve complete conformance of the shell surface to the spar surface, with no compressing or stretching of the shell material, the airfoil geometry must be developed with a straight line relationship between points of equal slope at the hub and tip sections. This is not a significant restraint in achieving a highly efficient vane profile. Detailed procedures for the diffusion bonding process itself, and the associated tooling to produce the shell, the spar, the shroud pieces and the containment bag for the pressing are major areas requiring development. The final assembly of the vane and its two shroud pieces by brazing require additional tooling and manufacturing development.

4.4 Inspection of the Shell/Spar Row 1 Vane

A procedure for inspection of the vane will need to be developed along with the fabrication development. First the spar must be qualified by inspection from the casting vendor and inspections for checking the final fabrication will be required. Some in process inspections may also be required. Such items as metal bonding of the shell to the spar and shell contour (deformation) will need to be verified.

4.5 Task Report

Prepare a topical report on the shell/span design and construction of Row 1 vane.

Schedule

Task can be completed in 24 months.

Development Task 5 Thermal Barrier Coatings

Background

The presence of a thermal barrier coating consisting of a metallic bonding layer and a ceramic outer layer, can substantially reduce structural metal temperatures in cooled combustion turbine components. Heat transfer and design considerations are covered elsewhere in this report. Ceramic thermal barrier coatings have been used for some time on selected combustion turbine components to solve particular overheating problems, especially combustors and transition pieces. When used in advanced high temperature land based combustion turbines, thermal barriers must resist chemical and mechanical deterioration due to the actions of high temperature combustion gas and the stresses of cycling and steady state operation. Currently used ceramic thermal barrier coatings have not been proven under these conditions for periods approaching normal turbine inspection intervals. There is a need to assess the as applied quality and durability of available thermal barriers in thicknesses suitable for use in the 2500°F RIT engine. Development of material and processing improvements will be necessary to achieve reasonable service life targets.

Objective

The objectives of the ceramic thermal barrier program are to ensure that the appropriate materials technology necessary to support the ATS 2500°F RIT turbine is available and that the coating will function as designed.

Technical Approach

Performance of this task will depend heavily on an evaluation and careful selection from the options available within existing thermal barrier coating technology. Problems which must be addressed in adapting thermal barrier coatings to engine hardware are thermal stress driven spallation, oxidation of the metallic bond coat, and hot corrosion of the bond coat. Process and production questions include the ability to produce uniform and consistent coating microstructure, thickness, and bonds, the ability to retain cooling hole geometry, and identification of reconditioning procedures. The approach proposed combines process development work with experienced vendors, laboratory studies, and field evaluation of thermal barrier coatings on representative hardware in existing engines. The intent will be to identify and evaluate candidate thermal barrier coatings, bond layers, and application processes, to evaluate effects on performance, and to develop life prediction methodology for service degradation of the thermal barrier coating systems.

Work Statement

5.1 Laboratory Testing and Coating Selection

A preliminary screening of available ceramic coatings, bond coats, and application methods will be performed to identify those most likely to be capable

of meeting the overall program goals. Laboratory test samples will be prepared and testing performed to assess critical properties, including: thermal shock resistance, oxidation resistance, and hot corrosion resistance.

5.2 Preliminary Process Development

Based on the results of the laboratory testing, thermal barrier coatings will be chosen for each component in question. The plasma spray and the electron beam-physical vapor deposition processes will be considered for application of the coating to each specific part and the preferred process will be chosen. In conjunction with selected vendors, the processes will be modified and refined to achieve optimal thickness control, surface roughness, and cooling hole geometry. A primary consideration will be the establishment of procedures which ensure a consistent product of known quality.

5.3 Performance Analysis

The expected effect of the thermal barrier coating on aerodynamic performance will be assessed, taking into account the change in surface finish, increased trailing edge thickness, and the possible alteration of film cooling hole geometry.

5.4 Field Testing

Field tests should be performed on critical engine components in later stages of the ATS program for: transition piece, row 1 vane, and stage 1 blade. These tests will be performed to characterize the long term engine performance of the candidate thermal barrier coatings, and will be performed in existing engines of a state-of-the-art design. Special attention will be given to selection of components where locally high temperatures and oxidation have been experienced and where the bond coat temperature is likely to be as high as that in the ATS application. This is an important requirement because bond coat oxidation and subsequent spalling appears to be a key component of life. The target for the testing program should be a period of at least 5000 hours and 200 starts.

5.5 Evaluation and Development of Life Prediction Model

Laboratory and field test materials will be periodically inspected and destructively tested as necessary to evaluate the performance of the thermal barrier coatings. A model will be developed to predict failure mode and the service life of the coatings in ATS service.

5.6 Final Process Development

Based on the foregoing results, vendors will be selected for each of the components requiring a thermal barrier coating. Tooling and the production process will be developed. Quality assurance procedures will be established.

5.7 Task Report

Prepare a topical report on the thermal barrier ceramic coating work performed in Phase II.

Schedule

Total task time is 30 months, with a 10-month effort within Phase II.

Development Task 6 Investment Castings for Vanes and Blades

Background

Investment castings incorporating a variety of internal flow passages for the passage of air to cool vanes and blades have been employed for several years. Some of the early cooled designs even used solid cast or forged vanes and blades with small holes drilled in the spanwise direction and/or chordwise through the trailing edge using the electrochemical machinery technique. As large amounts of cooling were required for the higher firing temperatures of later engines, higher cooling side heat transfer was required which resulted in more complicated internal cooling passages. Investment castings with integrally cast internal passages became the accepted design.

The current state-of-the art technology consists of cast vanes with internal spanwise cavities in which inserts are installed. High heat transfer is achieved by impingement of air on the vane walls. Cooling is further enhanced by gill holes around the vane walls to provide film cooling and trailing edge holes or slots for trailing edge cooling.

For blades, two approaches to cooling have been used. One approach is to integrally cast a complicated triple-pass or serpentine arrangement. The cooling air enters radially through the blade root or dovetail and flows through the blade by making two or three passes radially from hub to tip before exiting through trailing edge holes, the blade tip, and even through holes on the leading edge in a shower head configuration. Internal cooling enhancement can be further achieved by using turbulators which are ridges on the wall passages perpendicular to the flow that forces the flow to remain turbulent. A second approach is to integrally cast a number of spanwise holes in the blade. These holes can be tapered from the blade hub to tip and strategically located to achieve maximum cooling benefit. Turbulators located circumferentially around the hole and spaced periodically along the hole can also be included to enhance cooling.

Objective

To develop investment cast air cooled vanes and blades with sufficient cooling capacity to satisfy the cooling requirements for the turbine operating at a rotor inlet temperature of 2500°F.

Technical Approach

Based on results and review from the latest state-of-the art technology, vane and/or blade cooling configurations will be selected that meet the cooling requirements for the ATS combustion turbine. With the cooling configurations selected, reviews with the casting vendors to discuss materials and fabrication capabilities will be conducted. If these results are fruitful, preliminary designs can be completed.

Work Statement

6.1 Detail Design of Vanes and Blades

If the cooling objective can be achieved and the vendors can meet the material and fabrication requirements, designs will be finalized and detail drawings made. With the exception of row 1 vane, all other rows will be investment cast with the cooling configuration identified.

6.2 Material Selection for Vanes and Blades

As previously discussed in 5.4.3 Turbine Cooling Requirements and Task 2, Turbine Cooling Technology Development, materials will be selected consistent with vendors capabilities and considering minimal usage of cooling air for all rows of vanes and blades.

6.3 Fabrication Processes

In cases where new materials such as directional solidification physical part size, and more complex cooling configurations are required, work with vendors will be initiated early to allow for their tooling and fabrication development.

6.4 Inspection Techniques

During vendor discussions on casting capabilities and prior to order placement, inspection procedures must be agreed upon so that necessary tooling for the final inspection process can be developed to achieve part verification.

6.5 Task Report

Prepare a topical report on the investment castings for vanes and blades.

Schedule

Task can be completed in within Phase II in 18 months. Additional work in Phase III is required.

5.5.3 Phase III - Testing of Large Scale Components and Integrated Subsystems

Detailed engine designs, layouts, and manufacturing drawings will be completed for all the components not finalized in Section 5.5.2 Phase II - Critical Component Design and Testing. Results from testing and vane/blade fabrication development conducted in Section 5.5.2 will be included. Some additional areas where design studies need to be conducted and finalized are the cylinders and support system, bearings, shafting and couplings, seals and thrust management, and rotors for the separate components.

Design of the seals after the LP compressor and inlet to the HP compressor are important for minimizing leakage. In addition, if the unbalanced thrust exceeds available thrust bearing capacity, a balance piston will need to be designed and strategically located in the rotor system.

Final design reviews will be conducted on all key system components as well as the complete engine. Upon completion of these reviews, all manufacturing drawings will be released.

Tooling development and manufacture of parts will be initiated. As vendor produced parts become available, inspections for part approval will be conducted.

In conjunction with prototype engine tests, thermal barrier coated parts will be instrumented to verify the component design, including the coating. The durability of thermal barrier coatings on the various parts will be evaluated through periodic observation while in the engine and destructive testing after the instrumented components are removed. Modifications to the bond coat, ceramic layer, and/or application process will be made as necessary.

For the prototype engine, instrumentation will be designed and installed during assembly or prepared for installation later. The instrumentation will be used to verify proper operation of the combustion turbine from starting to synchronous speed and over the complete load range. The instrumentation could consist of 1300 to 1500 individual measuring points. When the unit is assembled for initial testing, part of the special instrumentation will be used since initial tests will likely be run at no load to verify starting, control functions, rotor dynamics, air flow checks, etc. Evaluation of the initial test results will determine if any design or control adjustments will be required before complete system demonstration in Phase IV - System Demonstration.

5.5.4 Phase IV - System Demonstration

The remainder of the special instrumentation will be installed in the combustion turbine and other parts of the ATS. Upon completion of plant construction and assembly of the ATS, the plant will be ready for full system demonstration.

It is intended to demonstrate the ATS plant in two discrete steps.

- At 2460°F TIT (2300°F RIT) or at the highest firing temperature obtained through continued developmental evaluation, and
- At 2680°F TIT (2500°F RIT).

This approach will allow the earliest demonstration (1998) of the proposed ICRCC ATS concept. A test program will be defined and during the system demonstration, test measurements will be obtained and evaluated to verify the plants operational and performance objectives.

Upon successful completion of the first stage testing, the high temperature components of the hot gas path will be incorporated into the combustion turbine with testing again being conducted to verify operational and performance objectives.

5.5.5 Program Schedule and Costs

A proposed schedule to perform ATS Program Phases II-IV is shown in Figure 5.1. It is possible for industry to begin full-scale demonstration of an ATS, as described in this study, by 1998. Because of the time required for development of high temperature materials, such as ceramics, components made of these materials would most likely have to be retrofitted to the combustion turbine during the 2001-2002 time-frame for system testing.

The financial limitations of this feasibility study and the wide array of developmental options within the selected ICRCC reference cycle precluded the developing of accurate costs for performing the work required to achieve an ATS by the year 2000. This task will best be accomplished during the conceptual design and product development stage of Phase II.

The question of potential levels of private sector cost sharing and the program's benefits versus costs are interdependent. The benefits of pursuing ATS are obvious. In order for our industrial and utility gas turbines to continue to be competitive in both domestic and world markets we must keep pace with our foreign competitors. The benefits of developing these advanced systems from an economic and environmental standpoint are clear as discussed in Section 5.6. The most reasonable way to fund these efforts within the U.S. to assure both industry and the country as a whole benefit, is for the U.S. Government and industry to cost-share in the development. In past development programs of a similar nature, where industry and Government have shared in the developmental effort, the range of cost-share for industry in the 20-35% range for the initial stages and 40-50% in the full-scale demonstration of the technology seems to be acceptable for both parties.

5.6 ATS Market Opportunities

In evaluating the potential market for the natural gas-fired ATS beyond the year 2000 and the potential for switching to a coal-fired system at some later date, possibly as early as 2005, one has to look at what's driving the market.

There are three major forces that are currently influencing the power generation market place. In all three cases these forces are positive with respect to ATS development. These three market forces are:

- Total power generation capacity needed for expansion and replacement between now and the year 2015. It is estimated that the U.S. will add over 140 gigawatts of new power generation capacity (this approximately 20% of total world capacity addition) from

ATS Development Program

Schedule for Phases II, III, & IV

Figure 5.1 ATS Program Schedule

1992-2001 (Figure 5.2), with 80 gigawatts being combustion turbines fueled by natural gas. The combustion turbine domestic sales trend of more than 10 gigawatts per year can be expected from the year 2000 through 2010. Overall, another 200 gigawatts of new capacity can be expected to be added by 2015 based on projected U.S. load growth (Figure 5.3). In addition to the new capacity additions there will be an estimated 150 gigawatts of existing domestic capacity that will have to be retired and replaced by the year 2015 (Figure 5.4). The potential for the use of advanced coal technologies grows from 100 gigawatts in 2005 to 250 gigawatts by the year 2015 (Figure 5.5).

- The availability and the price of power generation fuels both now and to the year 2010. Oil and gas prices from 1995 through the year 2010 are predicted to rise with an average annual price growth of approximately 4.0% (Figures 5.6 and 5.7). Coal is expected to maintain an annual price growth of about 0.1%. Although world oil reserves and domestic gas reserves are expected to remain plentiful through the year 2010, reliance of these premium fuels will decrease as the U.S. continues its policy of decreased oil dependency and as the price of natural gas continues to grow as compared to the relatively flat price of coal. Also, with an estimated U.S. coal reserve of 250 to 300 years and with the continued advancement of innovative clean coal technologies, coal should continue to play a major role in the fueling of U.S. power generation from now until the middle of the next century.

Figure 5.2 U.S. New Power Plants 1992-2001 Orders

Figure 5.3 New Electric Power Capacity Growth in the U.S.

Figure 5.4 Capacity Retirements in the U.S. After 1998

Figure 5.5 Potential Market for Advanced Coal-Based Technologies After 1998

Figure 5.6 Oil Price Projections

Figure 5.7 Gas Acquisition Price Trends

- **Environmental regulation.** In recent years a clear environmental bias has emerged in the evolution of public policy and in its interpretation at the federal and, especially, the state and local levels of government. This bias has been shaping energy-using decisions and will continue to do so even more in the future. These environmental concerns are forcing increased energy conservation and efficiencies. Acid rain legislation has already put financial and technical strains on the utility industry with emissions limits having been placed on SO_x , NO_x and particulates. Also, with carbon dioxide contributing to half the greenhouse gases and electric power generation being attributable for 35% of those gases, it won't be long before utilities are limited in these emissions.

The ATS program addresses all these factors and therefore the market for a high efficiency, low cost electric power generation system with reduced emission levels and fuel flexibility should be very favorable.

5.7 Commercialization Plan

As with any commercialization plan, Westinghouse's ATS commercialization plan is to bring the ATS technology to the market place and advance it to commercial use in a time frame that meets the needs of the market. Utility acceptance and confidence in the technology and equipment are the keys to ATS commercialization.

The plan will provide the framework for follow-on activity, including additional development, demonstration and marketing. With respect to the proposed Westinghouse ICRCC ATS, a majority of the proposed technology already exists. It will be the demonstration of combining the proposed technologies such as the intercoolers/aftercoolers and recuperator with the LP

compressor, HP compressor and power turbine, and the steam bottoming cycle as a highly efficient integrated system that will be the task. In addition, the improvements to today's existing advanced combustion turbine with new high temperature materials and advanced aero and mechanical designs will have to be demonstrated.

The commercialization process consists of six steps:

- Concept exploration and screening
- Technology plan
- Market survey and analysis
- Commercialization plan
- Product development
- Commercial introduction and product acceptance

An initial ATS market survey is contemplated while proceeding with the development of the ICRCC concept during the second phase of the program. Developing the market for ATS can best be done while the ATS program proceeds. Westinghouse meets on a regular basis with utility members of the Edison Electric Institute and committees of the Association of Edison Illuminating Companies. These contacts plus Westinghouse's network of global customers, will provide the opportunities to keep utility executives apprised of results and progress of the ATS program.

Product development will involve component and materials development along with component and subsystem demonstrations in Phase II and III of the overall ATS program. Introduction of the ATS technology will be best accomplished with direct involvement and risk sharing between the OEMs, utilities and government. This combined participation should result in full scale demonstration of the complete system in Phase IV of the ATS program by the year 2000. A detailed commercialization plan for a Westinghouse ATS will be developed in Phase II of the ATS program.

6. Conversion to Coal-Fueled Reference System

6.1 Identification of Coal-Fueled System

The National Energy Strategy recognizes that coal is a fuel source that must be considered for future power generations needs. With a coal reserve in the United States of more than 300 years, it is absolutely essential that we develop clean, efficient, coal utilization technologies that can be joined with the ATS program.

Westinghouse has led in the development of advanced coal-fueled programs for over 30 years and has supported over 55 individual research and development projects. Continuous effort has been devoted to the use of coal-fueled gases in Westinghouse turbines since the early 1970s. Today, Westinghouse's focus on coal-fueled systems include:

- Integrated Gasification Combined Cycle (IGCC)
- Pressurized Fluidized Combustion (PFBC)
- Direct Coal-Fired Turbines (DCFT)
- Indirect Coal-Fired Turbines (ICFT)

Westinghouse will continue to support and monitor the coal-fueled system programs mentioned above relative to developmental improvements, economics, successful demonstrations, and commercialization potential.

Each of the coal-fueled systems has its advantages and disadvantages. If the indirect coal-fired turbine system is ruled out because of the inability to provide timely development of high temperature heat exchanger technology, that leaves advanced IGCC, DCFT, and PFBC as the remaining candidates for the coal-fueled ATS. All three of these technologies are similar in that each process produces a low-to-medium Btu coal gas to be combusted in the turbine. This coal gas fuel is the primary concern when addressing the conversion of the advanced turbine from natural gas fuel.

6.2 Design Changes to Gas Turbine of Natural Gas-Fired Reference System

For the coal-fueled combustion turbine arrangement design modifications are centered about the combustion system, piping, and controls with tests being required for the combustion system and the manifold/inlet scroll that ducts the hot gases into the turbine.

One modification required of the combustion turbine is to change the combustion system from 16 baskets located within the combustion cylinder, equally spaced circumferentially, to two combustion chambers, eight baskets each, located externally to the main engine cylinder as shown in Figure 6.1. These two combustion chambers are located, one on each side of the turbine. Figure 6.2 shows the detail of transmitting the high temperature combustion exhaust gases from the combustor outlet to the turbine inlet.

The engine change as described above will involve a major design change to the combustion turbine cylinder from the compressor discharge to the turbine inlet. Again, as described in Section 5.5.2 Critical Component Testing, the critical structural problem is thermal stress and

Figure 6.1 Conceptual Dual Topping Combustor Arrangement

Figure 6.2 Topping Combustor Outlet to Turbine Inlet Gas Flow Concept

distortion caused by temperature differences between the compressor and combustor legs and the mating leg from the turbine cylinder. Also, a similar problem exists for the hot gas duct containment cylinder, annular duct, and inlet scroll that guide the hot gases into the turbine inlet. An extensive finite element heat transfer analysis will be required to redesign the cylinders, flanging, and the interior hot gas flow system to insure that stress and distortion limits are met.

It is expected the material for the main cylinder, horizontal joint, and vertical flange bolting will be the same as used in the natural gas-fired gas turbine in Section 5.5.2. The hot gas containment system could present the most critical problem for this design. This containment system represents a large area to maintain at acceptable metal temperatures, control leakages at the juncture to the turbine inlet, and minimize distortion to stay within acceptable limits. Materials would need to be the same as used in state of the art gas turbine combustors and transition pieces or possibly even using new concepts such as ceramics in some areas.

This arrangement of the combustor system into two off-board combustor housings, as opposed to the conventional in line on-board combustors of the Westinghouse combustion turbine, presents some major design and development problems. The extended surface of the hot gas flow path is much greater in area than would be the surface of the transition pieces for the conventional system. All of this surface must be cooled by compressor discharge air. The cooling required will be much more than what can be achieved by surface convection which has been the conventional way of cooling the transitions. Impingement cooling of ceramic tiles lining the interior of the extended hot gas flow path means a significant amount of compressor discharger air will by-pass the combustion process within the combustors, thereby reducing the amount of air available for the combustion process, cooling of the combustor wall surface, and the profiling of combustion products for uniform turbine inlet temperature to the first row vanes. It is difficult in an off-board combustion system arrangement, to have any real final control of turbine inlet temperature profile, because of lack of control on the mixing process in the large volume of space between the exit of the combustor and the inlet to the turbine.

The piping to and from the coal to fuel processing system must be designed to reduce and control thermal expansion stresses and pipe reactions on the turbine cylinder. The requirement for the use of by-pass valves to prevent turbine over speed in the event of sudden load loss will be studied and the piping arrangement will be evaluated to minimize piping cycle flow pressure losses.

Assuming the mass and energy stored in the coal gas system is similar to that stored in the recuperator for the natural gas-fired engine, the control system functions should also be similar except where changes are required to account for the considerably larger fuel flow. The control concept will be reviewed to identify any modifications.

6.3 Development Work for Coal-Fueled System

Conversion of the natural gas-fired reference system to a coal-fired system will require development in two areas, namely, the combustion system and the inlet manifold and scroll ducting from the combustor outlet to the turbine inlet.

CFATS Development Task 1 - Combustion System And Turbine Inlet Scroll

Background

The program activity described here is necessary to convert the natural gas-fired advanced turbine system plant to the coal-fueled system selected. The work scope described below covers the fuel nozzle/topping combustor area and the inlet scroll to the turbine.

The combustor outlet gas temperature of each combustor should be approximately 2650°F and is ducted into a scroll which brings the gas to the turbine inlet. Some of the combustion tests will need to include this scroll as part of the test configuration. Design requirements for the inlet scroll area are described in Section 6.2.

Combustor aerodynamics can be designed for lean-burn or rich-lean operation. The latter mode of operation can be used where the fuel gas contains significant quantities of fuel bound nitrogen. The use of the topping combustor permits the attainment of the high turbine inlet temperatures required for high efficiency.

The main combustor elements consisting of cylindrical sections, lend themselves to use of ceramic cylinders using ceramic materials developed in the parallel materials development section of the ATS program.

Downstream from the exit of the burners is a large volume of space that must have all of its surfaced cooled to protect the metal housings that separate the compressor discharge air from the hot gases exiting the combustor and entering the turbine. With a rotor inlet temperature of 2500°F, this means keeping the internal surface temperature of 2500°F or higher from penetrating into the metal support structure. A ceramic tile will be required, and the major cooling requirement will not be for the tile, but for the supporting structure that holds the tiles to the supporting surface. Ceramic tiles to protect the metal supporting structure have been used extensively by European manufacturers for off-board combustor systems but have been offered only for 2000°F turbine inlet systems.

Objective

Provide for the efficient combustion of coal fuel, while meeting all the emission targets and providing for satisfactory life and maintenance of components. Provide for satisfactory pattern factor of hot gas delivered to the turbine vanes with a minimum of scroll pressure drop and leakage loss. Utilize advanced ceramic materials as they are developed in the materials portion of the ATS program, working with ceramic manufacturers during material development activity.

Technical Approach

Perform the necessary combustion, heat transfer, and mechanical analysis to produce a design of the combustor and related components. Define development tests and test conditions for the necessary design evaluations. Install and evaluate performance in the prototype tests.

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Some of the technical design areas to be addressed are:

- **NO_x Control in the Combustor**

Two factors in the design will tend to cause high NO_x levels, the high inlet temperatures and the high combustor outlet temperatures. Both these factors tend to increase flame temperature, which drives NO_x up. Countering this trend will be the use of low-to-medium Btu fuel which acts to reduce flame temperature. These effects will have to be analyzed and accounted for in the design process. If NO_x levels are excessive, water injection can be utilized. If the fuel contains significant levels of fuel bound nitrogen (FBN), a rich-lean mode of operation will be selected which defeats the formation of FBN NO_x.

- **Combustor Wall Cooling and Downstream Cooling of Turbine Inlet Scroll**

Combustor wall cooling air temperature can be as high as 1600°F, but this temperature begins to approach allowable wall metal temperatures. Massive combustor wall cooling may be sufficient protection, but use of thermal barrier coatings may be necessary. The inlet scroll may also require use of thermal-barrier coatings or ceramic wall cylinders.

- **Fuel Gas Handling**

The fuel gas temperature entering the combustor can range from 1200° to 1600°F. The piping and valving controlling the fuel gas will have to be specially designed for these temperatures and for resisting coal gas trace contaminants. Carbon deposition on the piping walls may be a problem. A double walled arrangement may be necessary.

- **Combustor Transition and Turbine Scroll Entry**

One option for cooling this structure is to use regenerative cooling from the compressor discharge air. The combustor outlet temperature is so high that there may be local hot spots that require special design treatment. This structure lends itself to application of high temperature ceramics.

The flow distribution to the combustors and then into the turbine needs to be designed to minimize the pressure losses while giving the desired pattern factor of temperature to the turbine. For minimum risk, the entire system will be tested at low pressure (about 2 atmospheres) and full temperature conditions with modifications made, as required.

As previously discussed, the turbine inlet scroll may involve the use of ceramic tile lining the entire flow path from the burners to the turbine inlet at the first row vanes. Cooling the ceramic tile supports and fitting of these tiles into curved surfaces for good aerodynamic flow is another major design and manufacturing development problem. The support of the inner surface of the inlet scroll around the turbine shaft is a major design challenge, because all of the surface must be cooled sufficiently to protect the metal surface underneath the ceramic tiles providing the flow guiding surface into the turbine.

- Turbine Corrosion/Erosion/Deposition

Cleanup of the coal gas is projected to be such that erosion of the turbine is not expected to be a problem. Deposition of trace coal fuel elements may be a problem because the high combustor outlet temperature will leave these materials in a sticky or molten state.

Work Statement

1.1 Select Combustor Design

Perform preliminary design and select combustor module design choice(s) for further evaluation. Task duration - 4 months

1.2 Combustor Design and Test Rig

Design selected combustor modules and laboratory test rigs. Construct combustor hardware and rigs for laboratory test and evaluation. This test rig would be a small scale (single basket) configuration to evaluate basic combustion parameters. Task duration - 8 months

1.3 Laboratory Tests

Conduct laboratory tests of combustor modules, analyze and evaluate results from the rig designed in Task 1.2. Task duration - 12 months

1.4 Combustor Subsystem Test Design

Design combustor subsystem test apparatus and rigs. Construct combustor hardware and rigs for subsystem test and evaluation. This test rig would be a full scale configuration of one of the combustion chambers including an 180 degree section of the containment manifold and turbine inlet scroll to minimize pressure losses, provide proper temperature pattern factor to the turbine, and minimize leakage at the transition mouth. Task duration - 18 months

1.5 Subsystem Tests

Conduct subsystem tests of combustor modules, analyze and evaluate results from the rig designed in Task 1.4. Task duration - 21 months

1.6 Combustor Modules Design

Design prototype combustor modules and instrumentation for installation into the prototype engine. Task duration - 12 months

1.7 Combustor Manufacture

Construct prototype combustors and instrumentation for the prototype engine. Task duration - 18 months

1.8 Prototype Tests

Conduct prototype tests of combustor modules as part of demonstration testing. Analyze and evaluate results. Task duration - 18 months

6.4 Estimated Performance and Emissions

Ideally no degradation in performance or emissions is desired when switching from a natural gas-fueled system to a coal-fueled system. Emissions from the use of synthetic gas will depend on the cleanliness, chemical composition, and heating value of the gas. Two Westinghouse 501D5, 100 MW class combustion turbines at the Dow Chemical commercial demonstration coal gasification unit at Plaquemine, LA, were converted from operation on natural gas to a combination of natural gas (20%) and synthetic gas (80%) in 1987. Operational data has shown that when operated on 80% synthetic gas (239 Btu/scf LHV), and compared to baseload operation with natural gas, NO_x decreased by 45%, SO_x was below 10 ppm, CO was below 5 ppm, and power output actually increased by 10% due to the increased mass flow. Provided that when the ATS is converted from natural to coal gas, the gasifier is of adequate size, the synthetic gas produced is of sufficient quality, and the system component changes to accommodate the synthetic gas are made, it follows that emissions and performance could be expected to be as good or better than operation on natural gas.

6.5 Estimated Costs and Schedule for Coal-Fueled System Gas Turbine Development Work

The sum of all estimated task durations for the development work presented in Section 6.3 is 111 months. Allowing for certain effort to be done in parallel, the total elapsed time would be approximately 86 months. The associated costs for performing this development work was not addressed for the same reasons as presented in Section 5.5.5.

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