**Advanced Turbine Systems Program
Conceptual Design and Product
Development**

**Annual Report
August 1993 - July 1994**

November 1994

Work Performed Under Contract No.: DE-AC21-93MC30247

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
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Section 1

INTRODUCTION

The stated objective of the project was to analyze and evaluate different cycles for the natural gas-fired Advanced Turbine Systems (GFATS) in order to select one that would achieve all of the ATS Program goals. These goals are as follows:

- Greater than 60% plant cycle efficiency (on LHV basis)
- NO_x emissions 10% lower than current values
- Cost of electricity 10% less than current values
- RAM as for current combined cycle plants
- Demonstration by year 2000

Detailed cycle performance, cost of electricity, and RAM analysis were carried out to provide information on which the final selection of the GFATS cycle would be made. To achieve the very challenging goals, innovative approaches and technological advances are required, especially in combustion, aerodynamic design, cooling design, mechanical design, leakage control, materials, and coating technologies.

The ATS engine will be the next model in the series of large heavy duty gas turbines developed by Westinghouse. Westinghouse has been involved in gas turbine design and development since 1945. During that time, Westinghouse engineers made significant contributions in advancing gas turbine technology, especially in the field of heavy duty engines used in industrial and electrical utility applications. Some of the innovations included single shaft two bearing design, cold end drive, axial exhaust, cooled turbine airfoils in an industrial engine, and floating pad bearings. The improvement in turbine performance and mechanical efficiency were both evolutionary, where small improvements were made continuously, and revolutionary, where new designs were produced to achieve a large step change in engine output power and efficiency, as well as mechanical integrity and reliability.

This report discusses the major accomplishments achieved during the first year of the project.

Section 2

PROJECT OVERVIEW

STATUS

The objectives for the first year of the program have been addressed -- on budget, on schedule and with the technical goals met, and in some instances, exceeded. Definitive Topical Reports have been completed for the following Tasks:

- Task 1.0 Management Plant
- Task 2.0 Information Required for the National Environmental Policy Act.
- Task 3.0 Selection of Natural Gas-fired Advanced Turbine System.
- Task 4.0 Conversion of a Gas Fired Advanced Turbine System to a Coal-Fueled Advanced Turbine System.

In addition to the following overview of the major project tasks, highlights of the GFATS and CFATS engineering efforts are contained in Sections 3 and 4 of this report.

PROJECT TEAM

Several changes were made in the area of sub-contractor effort. As the program tasks were developed, it became apparent that for purposes of continuity some of the effort could be combined more efficiently with other Westinghouse tasks. The cost savings were employed in companion tasks to enhance the quality of the deliverables.

NATIONAL ENVIRONMENTAL POLICY (NEPA)

The locations chosen as candidate sites for testing of materials and systems were all evaluated as being in compliance with the requirements of NEPA. Several additional testing sites have been added and are in the process of being evaluated to the same NEPA approval processes prior to release of any testing efforts.

GAS FIRED ADVANCED TURBINE SYSTEM (GFATS)

Major accomplishments were attained in the reduction of the number of cycle arrangements initially studied for GFATS. The current results, as reported in the Topical Task, reveal that the target efficiency for GFATS is greater than 60%, based on the lower heating value (LHV) of the fuel. Additional efficiency improvements are anticipated as the program progresses.

High temperature material studies reveal that the casting of ceramic alloys is feasible and the few problems encountered are not insurmountable.

Ongoing work at Clemson University is being directed to the design and testing of critical components. The current effort involves diffuser air extraction studies.

COAL-FUELED ADVANCED TURBINE SYSTEM (CFATS)

Two closely related advanced coal-based power plant technologies were selected for consideration for conversion of the GFATS to CFATS application.

The first technology is an air-blown, coal gasification process with hot gas cleaning incorporated in an Integrated Gasification Combined Cycle (IGCC).

The second technology involves a second-generation Pressurized Fluidized Bed Combustion (PFBC) combined cycle.

Each of the two studies contains estimated performance and emission levels. A development program is outlined that focused on areas of combustion, flow distribution, structural analysis and materials selection.

MARKET STUDY

The analytical computer software required to perform a competitive economic analysis has been obtained. The analysis in progress will use a screening approach against selected alternative technologies. The factors chosen will evaluate the levelized cost of electricity as a function of capacity factor. The effort will identify how the ATS will perform competitively in peaking, intermediate and baseload applications.

SYSTEM DEFINITION AND ANALYSIS

The gas turbine flange-to-flange conceptual design is nearing completion.

Section 3

CYCLE ANALYSIS

Various cycles were considered and evaluated for the ATS Program on the basis of plant efficiency, emissions, cost of electricity, reliability-availability-maintainability (RAM), and program schedule requirements. The main cycle concepts investigated were the advanced combined cycle, intercooled, recuperated, reheat, and thermochemically recuperated cycles. The advanced combined cycle was selected for the ATS Program because it was considered to have the best potential for achieving all of the ATS Program goals by the year 2000.

Detailed cost and RAM analyses were carried out on six main cycle configurations and compared to the estimates for the Reference Plant.

Table 3-1 summarizes the results of the ATS cycle evaluation in comparison to the Reference Plant cycle.

Issues critical to the successful development of the advanced combined cycle plant were identified. The achievement of the ATS plant cycle efficiency and cost of electricity goals will require extremely high engine firing temperatures with minimized cooling of hot end components. This, in turn, will necessitate the selection and/or development of new alloys, materials, casting processes, and coatings for the ATS engine. To ensure success of this program, a concerted development effort and technological breakthroughs will be mandatory in combustion, aerodynamic design, cooling design, mechanical design, leakage control, and materials/coating technologies.

Based on the investigations described in this report, the following conclusions can be drawn:

1. The advanced combined cycle plant can achieve or improve on the ATS Program goals, namely:
 - Plant cycle efficiency >60% (LHV basis)
 - NO_x emissions <8 ppm
 - Cost of Electricity 10% less than current values
 - RAM as for best currently available plants.
2. Considerable effort will be required to develop critical technologies to ensure success of the ATS Program
3. From the analyses to date, Westinghouse intends to develop an advanced combined cycle plant with the following features:

- Advanced aero/heat transfer/materials technology
- Burner outlet temperature of 2750°F
- Closed-loop steam cooling
- Single crystal and directional solidified airfoils
- Improved thermal barrier and anti-corrosion coatings
- Ceramic ring segments
- Active tip clearance control
- Brush seals
- Reduced compressor dump/combustor loss
- Increased Row 4 blade exit area
- Reduced inlet and exhaust losses
- Fuel preheating
- All rotating T-G power components on a single shaft
- Advanced steam turbine design technology
- 1800 psi/1100°F/1100°F steam cycle

INTRODUCTION

Westinghouse has developed a product line that mirrors the general trend in gas turbine inlet temperatures. To make gas turbines more competitive than steam turbine plants, it has been necessary to develop more efficient cycles. Over the years, Westinghouse has performed many engineering studies to determine optimum cycles to minimize the cost of electricity. Some of the more promising cycles (intercooled, multiple shafts, reheat, steam injected, and water injected at various locations) have been studied in detail. In the final analysis, the simple cycle gas turbine combined with a steam bottoming cycle (a synergistic combination of the Brayton and Rankine cycles) was developed to increase overall cycle efficiency.

Current large natural gas-fired combined cycle power generation systems are capable of efficiency levels in the range of 54 percent. Within the ATS program, Westinghouse has been given the opportunity to re-evaluate cycle efficiencies using older, established concepts, such as intercooling and recuperation, and newer concepts, such as thermochemical recuperation. In addition, efficiency enhancements within the ATS selected cycle are to be evaluated to determine the best approaches to raising overall thermal plant efficiencies to greater than 60 percent while adhering to the other ATS program goals. The concepts considered in the Westinghouse analyses are required to be capable of demonstration within a three to four year time frame. From a baseline cycle definition, this report discusses as to how different concepts will affect the overall plant thermal efficiency. Results given below show that a plant efficiency of greater than 60 percent is achievable.

BASELINE CYCLE

In order to evaluate different technologies and concepts applicable to combined cycle power generation systems, a baseline combined cycle configuration first had to be developed to provide a basis for comparison of all the cycle concepts and technologies to be considered. For this purpose, a conventionally configured combustion turbine coupled with a three-pressure level reheat steam cycle was modeled to provide a baseline cycle. The combustion turbine rotor inlet temperature (RIT) was set at 2600°F to approximate near-term temperature capabilities. The compressor pressure ratio was set at 18. High pressure steam conditions entering the steam turbine were specified at 1450 psi and 1000°F and the hot reheat steam temperature was also set at 1000°F. This cycle configuration utilizes a turbine rotor cooling air heat to produce additional low pressure steam in the steam cycle via a heat exchanger located in the heat recovery steam generator (HRSG). Also, the natural gas fuel is preheated by feedwater recirculation flow.

COMPONENT IMPROVEMENTS

Several component improvements, available through recent technological developments and advanced design techniques, were incorporated into the power generation system and results in significant efficiency gains. The application of advanced design tools will result in increased compressor and turbine efficiencies. Blade tip and seal leakages will be minimized to further enhance efficiency. Improved materials, including ceramic components, coating, and cooling designs will be incorporated to reduce cooling requirements and improve cycle efficiency.

There are two electric generators in the baseline configuration. One of these is the combustion turbine generator, while the other is the steam turbine generator. Current generator designs are capable of higher efficiency than those chosen for the baseline cycle. While the combustion turbine generator of the baseline configuration is of sufficient size to cost effectively apply this technology, the steam turbine generator is not. By utilizing a single shaft arrangement, however, the smaller steam turbine generator is eliminated and the remaining single generator may be designed at the higher efficiency.

When the component improvements listed above are all incorporated into the baseline cycle, the net plant thermal efficiency is increased by approximately 2 percentage points.

STEAM CYCLE ENHANCEMENTS

The basic reason for raising the steam pressure and temperature of the Rankine cycle is to improve the potential thermal efficiency. The first cycle variations investigated within this study were modifications to the baseline cycle in which the steam cycle was enhanced, utilizing previous Westinghouse studies to optimize steam cycle configurations for subcritical and supercritical applications. The results of these studies indicated that increasing either high pressure steam superheat temperature, or the reheat steam temperature by 50°F, results in an improvement in combined cycle thermal efficiency of 0.1 percentage point. Increasing high pressure steam pressure from 1450 psi to 1800 psi results in an increase in net plant thermal efficiency of 0.1 percentage point. A further increase in pressure to 2400 psi yields only an additional 0.05 percentage point in thermal efficiency, while adding to the cost of the high pressure steam system. Also, since the steam turbine size is set by the exhaust energy of the combustion turbine, increasing steam pressure reduces the blade heights in the high pressure steam turbine. For 2400 psig high pressure steam, the resulting blade heights are much smaller and less efficient than for the 1800 psi steam. Therefore, the optimum steam cycle was determined to be at 1800 psi with 1100°F high pressure superheat steam and 1100°F reheat steam (both 100°F above the baseline temperature). This resulted in a 0.5 percentage point increase in net plant thermal efficiency and also in a slight increase in output due to the increased efficiency of the steam cycle. The steam temperatures were limited to 1100°F for this study due to steam turbine materials, reliability, and cost considerations. The steam temperature of 1100°F plus a reasonable steam superheater approach ΔT is determined by the combustion turbine exhaust temperature and this, in turn, set the baseline cycle pressure ratio.

ROTOR AIR COOLER HEAT UTILIZATION

The Westinghouse 501F combustion turbine combined cycle provides two options for rotor air cooler heat utilization. The first option is an air-to-air cooler to cool the rotor cooling air after it exits the compressor and prior to its introduction into the rotor. The rotor air heat is rejected to the atmosphere via an air-to-air cooler. The other option is to cool the rotor air via an air-to-exhaust gas heat exchanger located in the heat recovery steam generator upstream of the low pressure evaporator, as was done in the baseline cycle. With this configuration, the rotor air cooler heat is recovered by the steam cycle, which produces low pressure steam. This results in higher plant efficiency than that of the air-to-air cooler method, since the rotor air cooler heat is recovered by the low pressure steam system.

Another concept involves removing the HRSG rotor air cooler used in the baseline configuration and installing a rotor air cooler which exchanges heat with the incoming natural gas fuel (after the fuel has been preheated). This returns the rotor air heat back to the combustion turbine, which then requires less fuel to achieve the desired rotor inlet temperature. Therefore, the rotor air heat is recovered at the combustion turbine efficiency (typically about 40 percent), which is much higher than the low pressure steam system efficiency. This is, therefore, a much more effective recovery of the heat than is obtained via low pressure steam production, and results in an increase in net plant thermal efficiency of 0.4 percentage point over the baseline configuration.

CLOSED-LOOP STEAM COOLING

Most current technology gas turbine engines utilize air to cool the turbine vanes and rotors. This allows the turbine inlet temperature to be increased beyond the temperature at which the turbine material can be used without cooling, thus increasing the cycle efficiency and power output. However, the cooling air itself is a detriment to cycle efficiency in four ways. First, it is ejected from the turbine airfoils, thereby causing a disruption in the surrounding flow field. This increases the airfoils' irreversible pressure losses and results in a reduction in turbine efficiency. Secondly, since the cooling air is ejected from the airfoil into the gas path, the resulting mixing of the cooling air into the gas path results in irreversible pressure losses due to the non-ideal mixing of the streams, which have very different velocity vectors. The third loss mechanism is caused by the reduction in gas path temperature that accompanies the mixing of the cooling air into the gas path. This reduction in temperature reduces the work output of the turbine and, therefore, impairs cycle efficiency. Finally, the turbine cooling air must be pumped to pressures significantly higher than that of the gas path pressure at the location it is injected. This is done to assure that the cooling flow rate will be sufficient during certain operating conditions where the ratio of coolant pressure to gas path pressure drops below its design level. While some of this pressure is recovered by the turbine, there are internal losses as the cooling air passes from the compressor to the turbine gas path. The additional pumping work required to raise the cooling air to the required pressure is the associated loss.

The effect of reductions in cooling air results in a potential increase in combined cycle thermal efficiency for fractional reductions in cooling or leakage flows. Although turbine leakage generally carries a larger efficiency penalty than turbine cooling per given amount of flow, the fact that the amount of leakage flow is far less than the amount of cooling flow results in far less efficiency benefit from reducing turbine leakage flows a given fractional amount compared to reducing turbine cooling flows by the same fraction of their baseline value.

By using closed-loop steam cooling, the loss mechanisms described above can be largely eliminated, while still maintaining turbine material temperatures at an acceptable level. In combined cycles, the steam used for cooling the combustion turbine hot parts is usually taken from the steam bottoming cycle. This steam is then returned to the bottoming cycle after it has absorbed heat in the closed-loop steam cooling system. For an advanced bottoming steam cycle, closed-loop steam cooling would route cold reheat steam from the exit of the high pressure steam turbine to the combustion turbine vane casing and rotor. The steam is passed through passageways within the vane and rotor assemblies and through the vanes and rotors themselves,

then collected and sent back to the steam cycle intermediate pressure steam turbine as hot reheat steam. This approach to turbine cooling relies on convective and conductive heat transfer. Since no steam or cooling fluid is ejected from the airfoils, aside from a small amount of steam leakage through the rotor seals, there is very little influence of the cooling steam on the airfoil flow fields, and hence minimal mixing losses. Also, the reduction in gas path temperature is minimized, since the convective heat flux across the airfoils is relatively small. Typically, first vane cooling air mixing reduces the gas path temperature approximately 100 to 150°F. For closed-loop steam cooling however, the reduction in gas path temperature is only about 10 to 15°F, or one tenth of the reduction of conventional cooling techniques. Application of closed-loop steam cooling to the baseline configuration yields a 2 percentage point increase in combined cycle efficiency.

Closed-loop steam cooling of the transitions, which will be used on the ATS engine, will result in a small (about .05 percentage point) penalty in cycle efficiency due to heat extraction from the combustion gases. However, both closed-loop exhaust gas cooling and impingement air cooling of the transitions will result in a higher loss in cycle efficiency. The closed-loop exhaust gas cooling will be less efficient because of the work of compression. The impingement air cooling will be less efficient because of the higher combustor pressure loss requirement for this type of transition cooling scheme.

INCREASED COMPRESSOR PRESSURE RATIO

Commercial aircraft gas turbine engines are designed with high overall pressure ratio. This is done to maximize the simple cycle efficiency. For the ideal Brayton gas turbine cycle, the cycle efficiency is a function solely of the cycle pressure ratio and increases with cycle pressure ratio. In real cycles, the effect of non-ideal components causes the peak efficiency pressure ratio to decrease significantly from that of the Brayton cycle. Simple cycle efficiencies are relatively flat above a pressure ratio of approximately 40. This indicates that it is nearing the peak simple cycle efficiency. The steam cycle efficiency is seen to decrease with increasing combustion turbine pressure ratio. This is due to the reduction in combustion turbine exhaust temperature, which in turn reduces the maximum steam temperature and pressure and the steam's availability, and results in lower steam cycle efficiency. The effect of all of this on combined cycle efficiency is that it peaks around a pressure ratio of about 20, but remains approximately constant for a relatively large increase in compressor pressure ratio.

Reducing the compressor pressure ratio below the baseline value of 18 results in a significant decrease in combined cycle efficiency. This is due to the fact that, since the maximum steam temperature considered for this study was 1100°F, any decrease in pressure ratio below the baseline value of 18, where the maximum steam temperatures are reached, will increase the turbine exhaust temperature while maintaining the steam temperatures at 1100°F. This results in a much smaller increase in steam cycle efficiency than that obtainable by allowing the steam temperatures to rise with the turbine exhaust temperature. The significant reduction in the slope of the steam cycle efficiency line at this point causes the combined cycle efficiency decrease.

The maximum work output of the Brayton cycle occurs at a significantly lower value of pressure ratio than that for the peak efficiency. The simple cycle peak specific output occurs at a pressure ratio of approximately 18. Also, the steam cycle specific output is reduced as the pressure ratio

is increased, since there is less exhaust energy available to the steam cycle. The combined cycle specific output, which is merely the sum of the simple cycle and steam cycle specific outputs, decreases for increasing compressor pressure ratios.

Therefore, it is desired to select the lowest pressure ratio at which the peak combined cycle efficiency level is obtained. In this way both efficiency and output are maximized, and the cost of electricity is minimized (the baseline cycle is designed in this fashion).

The results of the above analysis apply to air cooled combustion turbines. To determine the optimum pressure ratio for closed-loop steam cooled turbines, the effects of the closed-loop steam cooling must first be considered. Since the turbine gas path temperature is reduced far less for closed-loop steam cooling than for air cooling, the combustion turbine exhaust temperature is higher for closed-loop steam cooled turbines than air cooled turbines operating at the same pressure ratio. Application of closed-loop steam cooling to the baseline configuration results in an increase in turbine exhaust temperature of approximately 100°F. This then indicates that the cycle pressure ratio should be set so that turbine exhaust temperature is equal to the maximum steam temperature plus margin for the superheater approach; and the cycle pressure ratio increased to reduce the turbine exhaust temperature 100°F. In the baseline cycle, this corresponds to increasing cycle pressure ratio from 18 to 25. The steam cycle will still have the same efficiency as the optimum steam cycle, but the combustion turbine simple cycle efficiency has been increased by approximately 3 percentage points. This results in a further increase in combined cycle efficiency of 1.6 percentage points, bringing the total combined cycle efficiency increase due to application of closed-loop steam cooling to 3.6 percentage points.

INCREASED TURBINE INLET TEMPERATURE

Since thermal efficiency increases with increasing turbine inlet temperature, the potential benefits of increased turbine inlet temperature were investigated. The RIT for the baseline cycle was increased 300°F to 2900°F. This resulted in a cycle output increase of 10 percent, and combined cycle thermal efficiency increase of slightly more than 1 percentage point. The reason that the performance increase was relatively small for such a large increase in turbine rotor inlet temperature is that, since the cooling technology remained constant as the temperature was increased, large amounts of additional turbine cooling air were required to maintain turbine material operating temperatures at an acceptable level. This increase in cooling flow decreased cycle efficiency by the three mechanisms described earlier, and this significantly offset the benefit of increasing the turbine rotor inlet temperature. Therefore, since efficiency and output would have been increased much more if turbine cooling was held at the same level as in the baseline cycle, increased turbine operating temperature must be accompanied by corresponding advancements in turbine cooling. However, even if cooling technology advancements were available to allow operation at much higher rotor inlet temperatures, the formation of NO_x at these higher temperatures would result in unacceptable emissions characteristics.

COMPRESSOR INTERCOOLING

The most typical arrangement for compressor intercooling involves removing the compressor air flow halfway through the compressor temperature rise, sending it through an air-to-water heat

exchanger, and then returning it to the compressor for further compression to combustor inlet pressure. The heat removed from the compressor air flow by the intercooler is rejected to the atmosphere, because, at the pressure ratios considered in this study, the heat is of too low a quality to be of use to the cycle.

Another intercooling concept is to spray water droplets into the compressor. As the air is compressed and increases in temperature, the water evaporates and absorbs heat. This results in a continuous cooling of the compressor. Note that for this concept the heat absorbed by the water is also rejected to the atmosphere, since this water is never condensed by the cycle but instead exhausted with the stack gases as low pressure steam.

Compressor intercooling reduces the compressor work, because it compresses the gas at a lower average temperature. Since the combustion and steam turbines produce approximately the same output as in the non-intercooled case, the overall cycle output is increased. However, since the compressor exit temperature is lowered, the amount of fuel that must be added to reach a given turbine inlet temperature is greater than that for the non-intercooled case. The ratio of the amount of compressor work saved to the amount of extra fuel energy added is about equal to the simple cycle efficiency. It can therefore be said that intercooling adds output at approximately the simple cycle efficiency. Since combined cycle efficiencies are significantly greater than simple cycle efficiencies it would be expected that the additional output at simple cycle efficiency would reduce the combined cycle net plant efficiency for the intercooled case. This trend is the same for a wide range of cycle pressure ratios. The simple cycle shows almost no change in efficiency for intercooling, which is expected since output is added at approximately the simple cycle efficiency.

Since the compressor work requirement is reduced while the gas and steam turbine work outputs remain approximately the same as in the non-intercooled case, the net power output of both the simple and combined cycles is increased. Also, since the turbine exhaust temperature is increased slightly due to the aforementioned exhaust gas composition effects, the steam cycle specific output is also increased slightly.

RECUPERATION

In recuperative cycles, turbine exhaust heat is recovered and returned to the combustion turbine combustor, usually via a heat exchange between the turbine exhaust gases and the compressor exit air flow. The discharge from the compressor exit is piped to an exhaust gas-to-air heat exchanger located aft of the combustion turbine. It is then heated by the turbine exhaust and returned to the combustor. Since the resulting combustor air inlet temperature is increased above that of the non-recuperated cycle, less fuel is required to heat the air to a given turbine inlet temperature. Because the turbine work and the compressor work are approximately the same as in the non-recuperated cycle, the decrease in fuel flow results in an increase in thermal efficiency. This is especially true for the simple cycle, since the heat recovered by recuperation is otherwise rejected to the atmosphere in the non-recuperative case. For combined cycles the efficiency is also increased, because the combustion turbine recovers the recuperated heat at the simple cycle efficiency, which is larger than the 30 to 35 percent thermal efficiency of the bottoming steam cycle, which recovers this heat in the non-recuperated case.

Installation of a recuperation system on the baseline configuration results in an increase in thermal efficiency of 1 percentage point. The steam cycle in this recuperated cycle has a lower efficiency than the steam cycle in the baseline configuration, because the recuperator exit temperature is significantly lower than the turbine exhaust temperature. However, the effect of reduced steam cycle efficiency is smaller than the effect of recovering the recuperated heat at the combustion turbine efficiency.

The thermal efficiency of the recuperated cycle can be increased further as follows. Since the combustor inlet flow is less than the turbine exhaust flow (due to the removal of the turbine cooling air prior to combustion) and has a higher specific heat (due to the combustion of the fuel), the heat capacity of the turbine exhaust flow is somewhat higher than that for the burner inlet flow. This means that the recuperated cycle described above does not fully utilize the quantity of heat available in the turbine exhaust. By placing a steam superheater in parallel with the recuperator, the remainder of the available turbine exhaust heat can be recovered at its maximum quality. The maximum steam temperature can then be raised to that of the baseline cycle (1000°F), and the cycle efficiency is increased by an additional 0.1 percentage point.

Since recuperative cycles return exhaust energy to the combustion turbine, less energy is available to the steam cycle, and the resulting steam turbine output is lower than that of the baseline configuration. However, the combustion turbine output is approximately the same as in the baseline cycle (minus losses in the recuperation system). This means that recuperative cycles carry a significant output penalty, with this penalty being proportional to the amount of recuperation performed.

INTERCOOLING WITH RECUPERATION

From a simple cycle standpoint, the combination of intercooling with recuperation eliminates the problem of the reduced combustor inlet temperature associated with intercooled cycles. The simple cycle then gets the benefit of the reduced compressor work and, at all but high pressure ratios, actually has a higher burner inlet temperature than the corresponding non-intercooled, non-recuperated cycle. This results in a dramatic increase in the simple cycle efficiency.

However, the bottoming cycle receives even less energy than in the recuperated cycle, since the recuperator removes much more heat from the turbine exhaust than in the recuperated, non-intercooled cycle. The additional heat removed corresponds to the heat rejected to the atmosphere by the intercooler. This means that we have merely displaced the lost energy of the intercooler by taking energy from the bottoming cycle. This is in addition to the energy already removed from the bottoming cycle in the non-intercooled, recuperative cycle. This results in a very low recuperator exit temperature, which in turn translates into a low generated steam pressure (low thermal availability) and a low efficiency steam bottoming cycle. The result is that, while simple cycle efficiency can be increased to over 50 percent, the combined cycle efficiency is reduced for the entire useful range of compressor pressure ratios. For conventional air-to-water heat exchanger intercooling of the compressor, the combined cycle efficiency is reduced approximately 1.9 percentage points at the baseline cycle pressure ratio. For continuously cooled compressors utilizing water droplet spraying into the compressor, the combined cycle efficiency is reduced by 0.4 percentage point.

Since the continuously cooled compressor with recuperation is not far below the baseline level of combined cycle thermal efficiency, it is worth investigating further optimization of the steam cycle in this case. As mentioned earlier, steam may be superheated in parallel with the recuperator, yielding more efficient recovery of the heat available to the steam cycle. Utilizing this approach, along with a two pressure level steam cycle, results in an increase in net plant efficiency of 0.8 percentage point over the baseline efficiency level.

Another approach to optimizing the intercooled recuperative cycle is to place a saturator between the compressor exit and the recuperator entrance. This saturator, also called an aftercooler, evaporates water into the compressor exit flow, resulting in a lower temperature, higher mass flow entering the recuperator. While this may be seen as a way to better balance the heat capacities of the hot and cold streams in the recuperator and thereby increase the amount of heat recuperated by the combustion turbine (in addition to the fact that the air flow is at lower temperature), it is important to realize that all of this additional recuperation is accomplished by the evaporation of water in the saturator. Since this water is never condensed by the cycle but instead is rejected through the stack as low pressure steam, the additional amount of energy recuperated is not recovered anywhere in the cycle. Furthermore, the recuperator exit temperature is reduced even further than in the intercooled recuperative cycle, resulting in an even lower efficiency steam cycle. Finally, note that, since the heat capacity of the cold side recuperator flow now closely matches that of the hot side flow, parallel steam superheat cannot be utilized to increase the steam cycle efficiency. The application of this concept to the baseline cycle results in a decrease in thermal efficiency of 2 percentage points.

REHEAT COMBUSTION TURBINE

Reheat combustion turbines utilize a sequential combustion process in which the air is compressed, combusted, expanded in a turbine to some pressure significantly greater than ambient, combusted again in a second combustor, and finally expanded by a second turbine to near ambient pressure. For a given turbine rotor inlet temperature, the simple cycle efficiency is increased for a reheat combustion turbine operating at a pressure ratio corresponding to the second combustor's operating pressure. This is because the reheat cycle performs some of its combustion and expansion at a higher pressure ratio, which increases simple cycle efficiency (see section on increased compressor pressure ratio). From a purely thermodynamic standpoint, the average temperature at which heat is added is raised, thus raising the Carnot efficiency of the cycle. For combined cycles the turbine exhaust temperature can be controlled by the selection of second turbine inlet temperature and expansion ratio. This in turn allows control over the efficiency of the steam bottoming cycle.

Another beneficial feature of reheat combustion turbine cycles is that they exhibit higher output for a fixed compressor flow rate and turbine inlet temperature than non-reheat cycles. This is due to the fact that they burn sequentially with an expansion between the combustors. This allows for the addition of more fuel (in the second combustor) without violation of the turbine rotor inlet temperature limit. Also, since this results in a higher fuel-to-air ratio than in non-reheat cycles, they burn closer to stoichiometry and exhaust lower concentrations of excess oxygen.

Applying combustion turbine reheat to the baseline cycle, with the second combustor operating at the exit pressure and temperature equal to those of the baseline cycle, results in nearly identical turbine exhaust temperatures, and therefore the steam cycle of the combustion turbine reheat case is not compromised and is identical to the steam cycle of the baseline cycle. However, the compressor pressure ratio has been increased to 36. This necessitates the addition of 6 compressor stages, and an additional combustor and turbine stage located upstream of the second combustor. The simple cycle efficiency is increased nearly 2 percentage points from the baseline simple cycle level. Since steam cycle efficiency remains at the level of that in the baseline cycle, the combined cycle efficiency is increased 1.3 percentage points.

An investigation was made into the application of intercooling and recuperation to reheat combustion turbine cycles with the same, reduced efficiency results as for non-reheat combined cycles.

THERMOCHEMICAL RECUPERATION

In a thermochemical recuperation power plant, a portion of the stack exhaust (flue) gas is removed from the stack, compressed, mixed with natural gas fuel, heated with exhaust heat from the combustion turbine, and mixed with the compressor exit air as it enters the combustor. As the mixture of natural gas and flue gas is heated by the combustion turbine exhaust, an endothermic reaction occurs between the methane and the carbon dioxide and water in the flue gas. This reaction occurs in the presence of a nickel-based catalyst, and results in the production of hydrogen and carbon monoxide. For complete conversion of the methane, the effective fuel heating value is increased approximately 30 percent. Therefore, the natural gas / flue gas mixture absorbs heat thermally (as it is heated) and chemically (via the endothermic reaction), resulting in a larger potential recuperation of exhaust energy than could be obtained by conventional recuperation, which recovers energy by heat alone. In fact, with full conversion of the natural gas fuel to hydrogen and carbon monoxide, up to twice the energy recuperated by the standard recuperative cycle may be recovered.

The endothermic reaction described above is accelerated for low excess oxygen in the reacting mixture, low pressures, and high mass ratios of recirculated flue gases to methane. Therefore, in order to take full advantage of this concept, the engine is controlled by running the combustor at near stoichiometric fuel-to-air ratios (10 percent excess air at the combustor exit) and using flue gas recirculation to quench the combustion products down to the desired turbine inlet temperature. This maximizes flue gas recirculation and minimizes excess oxygen in the flue gas. For typical cycles utilizing this control philosophy, the resulting recirculation rate of flue gas is over 50 percent of turbine flow. This means that both the air compressor flow rate and stack exhaust flow rate are less than half that of conventional cycles with the same turbine size.

Another advantage of the thermochemical recuperation / flue gas recirculation (TCR/FGR) concept is that, because the fuel has a low adiabatic flame temperature, it contains a significant amount of hydrogen and operates with very low levels of excess oxygen in the exhaust, the resulting emissions of NO_x and CO are much lower than those for conventional design power plants.

When applied to the baseline configuration, thermochemical recuperation yields a combined cycle thermal efficiency over 2 percentage points greater than that of the baseline cycle for several different TCR/FGR configurations. There are many possible cycle configurations, other than those described above, which can utilize thermochemical recuperation.

STEAM INJECTION

To apply steam injection to the baseline configuration, both the high pressure and low pressure steam systems are eliminated, leaving only the intermediate pressure steam system. The intermediate pressure superheated steam is then routed into the combustion turbine combustor inlet. Therefore, all steam turbines are eliminated, along with the condenser and the cooling tower. This results in a cycle which is significantly less expensive to build than the baseline cycle and it provides a means to reduce NO_x emissions via the large amount of steam injection. However, the cost associated with demineralization of the large cycle make up flow must also be considered.

For a given turbine flow area, the compressor flow size must be reduced significantly, due to the addition of a large amount of steam, which now must pass through the turbine in addition to the compressor exit air flow. Since the steam is generated with exhaust heat, which is not used in a simple combustion turbine cycle, this concept provides a significant portion of the combustor inlet gas flow with very little work requirement (a small amount of energy is needed to run the feedwater pumps which pump the water to the intermediate steam pressure). Therefore, from a simple cycle standpoint, the work of compression is much reduced due to the fact that some of the combustor inlet gas flow is compressed as a liquid. Also, the turbine output is increased significantly, since the average specific heat of the working fluid is increased considerably by the presence of the steam. The simple cycle efficiency and output are therefore increased by steam injection.

Compared to combined cycles, however, elimination of the high pressure steam system results in generation of more intermediate pressure steam, but this steam is at significantly lower availability due to its lower pressure. Also, since the low pressure steam system has been eliminated as well, the exhaust stack gas temperature is much higher, as is the associated heat loss. Finally, since the steam injected into the gas turbine is effectively throttled to its partial pressure upon mixing with the compressor exit air without doing any work, and is only expanded to its partial pressure in the exhaust stack (which is significantly higher than typical low pressure steam turbine exit pressure), the resulting steam expansion ratio is much smaller than that of the conventional steam turbine cycle. These losses result in a reduction in combined cycle efficiency in the range of 5 to 8 percentage points. Net plant output, while much higher than the baseline configuration simple cycle output, is less than the combined cycle output.

Section 4

CONVERSION TO COAL-FIRED ATS

A number of advanced, coal-fired power generation technologies have been under development that could be applied to the GFATS. These include a broad range of coal gasification technologies (fixed bed, fluid bed, and entrained bed), Second-Generation PFBC, and Direct Coal-Fired Turbines (DCFTs). Two advanced, coal-fueled technologies have been selected for consideration as CFATS: Air-blown IGCC with hot gas cleaning, based on the KRW fluidized bed gasifier; and Second-Generation Pressurized Fluidized Bed Combustion (PFBC). The rationale for these selections is summarized in this section.

The selection of a coal-fired reference system for the conversion of the GFATS to a CFATS has been made based on performance potential (power conversion and emissions), cost potential, and state of development. Table 4-1 summarizes estimated thermal conversion performance, cost potential, and indicates the status of development of several advanced, coal-fueled technologies that could achieve the desired turbine inlet conditions. These are compared to natural gas-fired turbine cycles, as well as to conventional coal-fired power plants -- Atmospheric-pressure Fluidized Bed Combustion (AFBC) and pulverized-coal combustion boilers with flue gas desulfurization (PC/FGD). The estimates represent performance using currently available combustion turbine technology and standard combined-cycle design conditions. The values in the table are representative, and should be applied for general comparison purposes only. Such results are sensitive to coal type, site location and conditions, and process design specifics. The air-blown IGCC technologies with hot gas cleaning, and Second-Generation PFBC appear to have the greatest thermal performance and cost potential. Their respective status of development is categorized as "early demonstration," which is suitable for the ATS program.

Table 4-2 summarizes the environmental emissions performance of these same technologies. Shown are emissions of NO_x , SO_2 , CO_2 , particulate, and solid waste that the technology can economically achieve, and they are presented on the basis of the plant net generating output (lb/hr per MWe), rather than on the basis of fuel input (lb/MBtu). A typical bituminous coal composition was applied to make the estimates (ash content 10 wt%, sulfur content 3 wt%, LHV 12,940 Btu/lb).

All of the advanced technologies are estimated to have the capability of economically improving on the performance required cleaning, and Second-Generation PFBC have outstanding environmental performance. by current and projected future emissions standards. Air-blown IGCC with hot gas The solid waste from these technologies is higher than that from several other advanced technologies, but is lower than that from conventional, coal-fired power plants on the basis presented in the table. Solid waste disposal, and its potential by-product uses are currently developing issues.

Table 4-1 - Performance of Alternative Power Generation Technologies

	Heat Rate Btu/kWh (LHV)	Cycle Eff. % (LHV)	Capital \$/kW	COE mills/kWh	Status of Development
<u>Advanced Technologies</u>					
IGCC (O ₂ -blown) ^a	7618	44.8	1500	76.9	Near commercial
IGCC (Air-blown) ^b	7380	46.7	1350	71.7	Early demonstration
Direct-Fired ^c	7170	47.7	1300	68.5	Developmental
2nd-Gen. PFBC	7580	44.6	1250	68.0	Early demonstration
<u>Conventional Technologies</u>					
AFBC	9600	35.2	1550	83.5	
PC/FGD	9600	35.2	1530	82.5	
Gas Simple-Cycle	9350	36.5	390	72.6	
Gas Combined-Cycle	6320	54.0	550	68.0	

a: Entrained, oxygen-blown gasifier with conventional coal gas cleaning

b: Fluidized bed, air-blown gasifier with hot gas cleaning

c: Slagging, air-blown gasifier with hot gas cleaning

Table 4-2 - Emissions of Alternative Power Generation Technologies

	NO _x	Emission (lb/hr/MW _e)		Particulate	Solid Waste
		SO ₂	CO ₂		
<u>Advanced Technologies</u>					
IGCC (O ₂ -blown) ^a	1.7	0.40	1774	6.7	79.0
IGCC (Air-blown) ^b	1.5	0.68	1506	5.7	226.9
Direct-Fired ^c	1.4	0.67	1464	5.6	166.0
2nd-Gen. PFBC ^d	1.5	1.76	1547	5.9	215.3
<u>Conventional Technologies</u>					
AFBC ^e	2.8	4.45	1959	22.2	256.3
PC/FGD ^f	5.7	4.45	1959	22.2	244.6
Gas Simple-Cycle ^g	0.8	---	1061	---	---
Gas Combined-Cycle ^g	0.7	---	843	---	---

a: NO_x 0.2 lb/MBtu, sulfur removal 99%, particulate 0.01 lb/MBtu

b: NO_x 0.2 lb/MBtu, sulfur removal 98%, Ca/S 2.0, particulate 0.01 lb/MBtu

c: NO_x 0.2 lb/MBtu, sulfur removal 98%, Ca/S 1.8, particulate 0.01 lb/MBtu

d: NO_x 0.2 lb/MBtu, sulfur removal 95%, Ca/S 1.8, particulate 0.01 lb/MBtu

e: NO_x 0.3 lb/MBtu, sulfur removal 90%, Ca/S 2.5, ESP particulate 0.03 lb/MBtu

f: NO_x 0.6 lb/MBtu, sulfur removal 90%, Ca/S 1.1, ESP particulate 0.03 lb/MBtu
These represent current NSPS for coal-fired power plants.

g: Low-NO_x burners with 25 ppmv NO_x (at 15% oxygen)

Based on the tabulated comparisons, the air-blown and oxygen-blown coal conversion processes with hot gas cleaning, and Second-Generation PFBC, have the greatest thermal efficiency potential, and have adequate emissions control potential. Air-blown, fluidized bed coal gasification processes with hot gas cleaning (for example, the KRW gasifier, Sierra Pacific, Pinon Pine Project) and Second-Generation PFBC (Air Products, Four Rivers Energy Modernization Project) are scheduled to be demonstrated in Clean Coal Technology Programs, and are appropriate for consideration in the CFATS. Both of these technologies are considered as CFATS because they pose very similar conditions and constraints on the combustion turbine and are both involved in major demonstration programs that will lead to commercialization.

These selected technologies also provide greater challenges to the combustion turbine than other advanced technologies, because: 1) air-blown, hot gas cleaning systems deliver a high-volumetric flow of low-Btu fuel gas to the combustion turbine at a high temperature, requiring large manifolding/combustion systems adapted to high-temperature duty; and 2) hot gas cleaning produces fuel gas conditions with potentially higher contaminant contents than do low-temperature cleaning systems. Successful development of the selected CFATS ensures that adaptation to the other coal-fueled technologies can be achieved with relative ease.

ESTIMATE OF CFATS PERFORMANCE AND EMISSIONS

CFATS performance and emissions estimates for IGCC (air-blown, fluidized bed gasifier with hot gas cleaning) and Second-Generation PFBC have been made using prior process studies for these technologies. The power cycle conditions in the prior studies were modified to the conditions of the CFATS, and process material and energy balances were developed to estimate the modified power plant performance. In both cases, significant increases in plant performance were obtained, while environmental emissions were maintained at low levels.

IGCC CFATS DESCRIPTION

An IGCC, CFATS process was chosen as being representative of several types of air-blown, coal gasification plants with hot gas cleaning being developed in Clean Coal Projects, such as the Sierra Pacific, Pinon Pine Project (KRW fluid bed gasifier), the Tamco Power Partners, Tom Creek Project (Institute of Gas Technology fluid bed gasifier), and the ABB CE, Springfield Project (Combustion Engineering entrained gasifier). Estimates of plant cycle performance and emissions were made for a specific KRW fluid bed gasification process using process design information reported in a recent DOE/METC study [Southern Company Services, "Assessment of Coal Gasification/Hot Gas Cleanup Based Advanced Gas Turbine Systems, Dec. 1990," DOE/MC/26019-3004].

In this process, coal and a sulfur sorbent, limestone or dolomite, are prepared (sized and dried) for feeding using conventional handling and preparation equipment. The coal and sorbent are pressurized and distributed to the fluid bed gasifier by a commercial feeding system in either dry or slurry form. The gasifier generates a low-Btu fuel gas that has been partially desulfurized by the in-bed sorbent.

The fuel gas is pre-cleaned by conventional cyclones and the captured ash, char and sorbent particles are either recycled to the gasifier or are removed to solid waste processing. The fuel gas is cooled by indirect, high-pressure steam heat exchange, to a temperature of 800° to 1000°F, as is dictated by the zinc-based, sulfur removal process. After cooling, the fuel gas sulfur content is reduced to an acceptable level by a regenerative, zinc-based, fluid bed sulfur removal process. The sulfur removal process may either produce a sulfur by-product, or may recycle the regenerator sulfur stream to the gasifier, or to solid waste processing. The desulfurized fuel gas then passes through a ceramic barrier filter for final particle removal before entering the combustor. Fuel gas particle content is limited to a total solids content and maximum particle size acceptable to the turbine expander, as well as to environmental standards.

Air provided by the turbine compressor combusts the fuel gas to the desired temperature of 2700°F. The combustor is a dry, low-NO_x design that is closely integrated with the turbine. A portion of the turbine compressor air also is boosted in pressure for gasifier oxidant supply. The combusted fuel gas is expanded in the turbine, and steam is generated in the heat recovery steam generator (HRSG) for the steam bottoming cycle. A portion of the high-pressure, cold reheat steam is consumed in the gasifier, assisting the coal gasification reactions. High-pressure steam is also heated in the fuel gas cooler. The stack gas temperature exiting the HRSG is limited by the gas dew point at its prevailing acid-gas content.

Waste solids from the gasifier, the cyclones and the ceramic barrier filter are treated in a waste process to utilize the waste solids fuel content and sensible heat content, and to render the waste environmentally acceptable.

The coal gasification process chosen for study is the KRW process. It is carried out in a bed of char and ash with added limestone sorbent for fuel gas desulfurization and volatile product cracking. The bed is fluidized by the air and steam required for carbon conversion and temperature adjustment. The ash is agglomerated in the bed to facilitate its removal. The fuel gas leaving the fluidized bed is cleaned hot. The possible cleaning steps include:

- absorption of residual H₂S and alkalis (Na and K compounds carried in the gas) by solid sorbents,
- reduction of the NH₃ content by catalytic decomposition,
- removal of particulates by a filter.

The temperature of the fuel gas between the gasifier and the turbine is lowered for cleaning and for delivery to the turbine at 1000°F. This reduction is accomplished either by a steam quench or by heat interchange preferably, for best efficiency, with the steam and/or air fed to the gasifier.

The ash and sorbent solids removed from the bed are oxidized with air to recover their carbon content as heat and to convert the sulfided limestone based sorbent to a sulfate, suitable for disposal. This process can be carried out at ambient pressure; and the hot combustion product gases, used for coal preheating and drying and/or for steam generation. The temperature of the process must be carefully controlled to prevent the release of sulfur from the sulfided sorbent as

SO₂. An alternative process carries out the oxidation of the ash and sorbent at pressure with the return of the hot gases to the gasifier. In this instance, the heat is efficiently recovered in fuel production; and the escape of SO₂ is effectively prevented.

SECOND-GENERATION PFBC CFATS DESCRIPTION

Specific plant performance and emissions estimates were prepared for the specific configuration of the Second-Generation PFBC technology being developed by the Foster Wheeler Corporation under a DOE/METC supported program [Phase 1, Task 1, Topical Report, DOE/MC/21023-2825, Vol. 1, September 1989].

In this process, coal and a sulfur sorbent, limestone or dolomite, are prepared (sized and dried) for feeding using conventional handling and preparation equipment. The coal and limestone are pressurized and distributed to the carbonizer with a commercial feeding system in either dry or slurry form. The carbonizer generates a low-Btu fuel gas that has been substantially desulfurized by the in-bed sorbent. The carbonizer differs from a gasifier in that it produces a large flow of char that must be combusted separately. The carbonizer fuel gas acts as a topping fuel for the char combustor. Char drained from the carbonizer and captured by the cyclones and ceramic barrier filter is circulated to a fluidized bed combustor. In this combustor, char is burned, sulfur released from the char is captured by the sorbent contained in the solids from the carbonizer, and steam is generated in the fluid bed heat exchanger (FBHX).

The fuel gas and combustion gas streams are pre-cleaned by conventional cyclones. Neither the fuel gas or the combustion gas is cooled before final particle control is performed by ceramic barrier filters, and additional sulfur removal is not required. Particle penetration is limited to total content and particle sizes acceptable to the turbine expander. Alkali removal may be required from the fuel gas or vitiated air streams, using, for example, packed bed absorbent.

"Vitiated air" issued from the fluid bed combustor combusts the fuel gas to the desired temperature of 2700°F. The combustor is a dry, low-NO_x design specially configured to utilize the hot fuel gas and the hot, vitiated air. A portion of the turbine compressor air is boosted in pressure for carbonizer oxidant supply. The combusted fuel gas is expanded in the turbine, and steam is generated in the heat recovery steam generator for the steam bottoming cycle. A small portion of the high-pressure, cold reheat steam may be consumed in the carbonizer, preventing coking from occurring. The stack gas temperature in the HRSG is limited by the gas dew point at its prevailing acid-gas content.

Waste solids from the fluid bed combustor, the cyclones and the ceramic barrier filter are cooled to utilize the waste solids sensible heat content. No treatment is required to render the waste environmentally acceptable.

PERFORMANCE AND EMISSIONS ESTIMATES

The performance and emissions of the selected CFATS has been estimated by modifying the process designs found in the Reference Studies for these two technologies. Southern Company Services evaluated, with the M.W. Kellogg Company, Tennessee Technological University, and

the General Electric Company, several IGCC process configurations of the KRW gasifier using hot gas cleaning. Case 3 from this study, a nominal 200 MWe plant having bulk sulfur removal with limestone in the KRW, fluid bed gasifier, and zinc ferrite sulfur polishing, was used as the basis for the performance estimates. The Foster Wheeler Development Corporation (FWDC) with several subcontractors, has developed commercial plant designs and performance evaluations for Second-Generation PFBC. A recent evaluation of Second-Generation PFBC, performed by FWDC with a high-temperature turbine, was applied as the basis for the CFATS performance emissions.

Table 4-3 lists the power plant thermal performance for both the Reference Studies and the CFATS cases, along with the steam cycle conditions and the combustion turbine conditions.

The fluidized bed gasification system presented in the reference study has the following unique features:

- The moist coal is dried and preheated at atmospheric pressure prior to feeding into the fluidized bed gasifier. Hot combustion product gases from a combined carbon burnup/sorbent sulfation operation are used in this process and exhausted to the stack.
- The coal feed and the char recycle streams are transported with cooled, cleaned, and compressed fuel product gases. The recycle fraction is not quantified in the flow sheet, but calculations indicate that between 30 and 45% of the fuel product is recycled for coal and other transport. The recycle stream also provides most of the water vapor, steam, required to moderate the operating temperature of the gasifier.
- The product fuel gas is cleaned -- H_2S , NH_3 , Na, K, and Cl; and particulate removal or reduction -- and cooled to 1000°F in preparation for recycle and for feed to the turbine. (The order of the various cleaning and cooling operations is not completely specified.) Cooling of the product fuel gas is carried out by a combination of heat exchange, generating steam and hot boiler feed water for the bottoming Rankine cycle, and of steam quench, enhancing the steam content of the product gas for recycle to the gasifier and feed to the turbine.

The air/coal ratio is 3.4 lb/lb. The steam/coal ratio is not directly specified and cannot be accurately calculated because of a lack of information on both the fuel gas recycle and the temperature of the gasifier; it appears to be in the range 0.4 to 0.7 lb/lb. The Ca/S mol ratio for the sorbent feed is 3.0.

Table 4-3 - CFATS Thermal Performance

	IGCC (KRW gasifier)	Second-Generation PFBC
REFERENCE STUDY		
Steam conditions		
Temperature (F):	1000	1000
Pressure (psia):	1600	2400
Combustion turbine conditions		
Combustion temperature (F):	2350	2800
Expansion Ratio:		12
Stack gas temperature (F):	280	280
Plant net efficiency - LHV (%):	41.5	48.7
Gasifier cold gas eff. - LHV (%):	72.1	---
CFATS PERFORMANCE		
	Modified Process*	
Turbine Frame	ATS	ATS
Steam conditions		
Temperature (F):	1100	1100
Pressure (psig):	1800	1800
Combustion turbine conditions		
Combustion temperature (F):	2700	2700
Expansion Ratio:	18	18
Stack gas temperature (F):	280	280
Plant net efficiency - LHV (%):	>52	>52
Gasifier cold gas eff. - LHV (%):	86.8	---
Gasifier hot gas eff. - LHV (%):	95.8	---

* Process was modified to incorporate coal drying, carbon burnup, and sorbent processing within the gasifier. Steam used for coal and char transport rather than recycle gas. Fuel gas cooling produced steam fed directly to the gasifier.

A modified fluidized bed coal gasification process has been proposed and evaluated for the CFATS, having these distinct features.

- The moist coal and recycle char are fed to the gasifier in superheated steam. No cooled, cleaned, compressed fuel gas is recycled.
- Burnup of carbon from the ash and sulfation of the spent sorbent are carried out with air at system pressure and the hot gases are routed to the gasifier.
- The product fuel gas is cleaned as previously, but cooling is accomplished in exchangers which provide superheated steam (and perhaps also heated air) for gasification. No steam is exported to the bottoming Rankine cycle.

The air/coal ratio is 3.0 lb/lb; the steam/coal ratio, 0.8 lb/lb. The Ca/S mol ratio is 2. The gasification temperature is 1806°F; the product fuel gas temperature, 1000°F. Material and energy balances have been carried out for this process based on the Illinois No. 6 coal feed selected for the reference study described above. These simplified calculations assume equilibria are attained in the gasifier for the reactions:

The fuel gas quantity and composition have been calculated for this modified fluidized bed coal gasification process. Both the hot and the cold gas efficiencies are significantly increased by the process modifications. Further increases in efficiency, to the extent practical in the design and operation of the plant, might be obtained by:

- increasing the temperature of the product fuel gas fed to the gas turbine;
- reducing the air/coal, steam/coal, and sorbent/coal ratios; and,
- increasing the pressure and reducing the temperature of the gasification process.

The overall performance of a CFATS plant in power generation has been estimated based on the modified fluidized bed coal gasification process described above and on the selected GFATS cycle selected by Westinghouse for development. The gasification study has defined the coal, air, and water input streams and the fuel gas and steam output streams. Combustion calculations have been carried out to determine the quantities of fuel gas and air to reach the specified combustor outlet temperature of about 2700°F. The Westinghouse GFATS cycle calculations have been used to define four work terms:

- compression of the air used in combustion, dilution, and gasification -- assumed proportional to the mols of air;

- compression of the air used in turbine vane, blade, and rotor cooling -- assumed proportional to the mols of hot gases entering the turbine;
- expansion of the air diluted combustion products from the turbine combustor -- assumed proportional to the mols entering the turbine times the quantity $[(\gamma)/(\gamma-1)]$ where γ is the specific heat ratio for the diluted combustion product gases; and,
- generation and expansion of steam in the bottoming Rankine cycle - assumed proportional to the mols of diluted combustion product gases times their specific heat times the temperature difference between inlet and outlet gases.

The gasification and combustion calculations for the gasification process described above have been used to determine the proportionality factors described above and in turn to estimate the work quantities associated with the compressor, expander turbine, and bottoming cycle of a CFATS plant from those of the Westinghouse GFATS. The work of the booster compressor for air feed to the gasifier from the turbine compressor has been calculated based on an assumed efficiency and inlet temperature. Mechanical, electrical, and windage losses and auxiliary power for the generation section have also been estimated from the Westinghouse GFATS study and auxiliary power for the gasifier and coal feed sections, from the Reference Study, previously cited.

The resulting net overall heat rates and plant efficiencies for the CFATS coal gasification power plants, based on the lower heating value of the fuel, are presented in Table 4-1. The overall plant efficiencies for both the KRW and the revised fluidized bed coal gasification power plant with the Westinghouse proposed advanced gas turbine cycle are 10 to 25% above those of the conventional coal fired power plants.

The Second-Generation PFBC Reference Study was a recent evaluation performed by Foster Wheeler and Gilbert/Commonwealth Associates of a commercial plant using a high temperature turbine. A major modification to this study was a notable reduction in the turbine air cooling needs. The CFATS conditions result in significant performance improvements over the Reference Studies.

Environmental performance estimates are listed in Table 4-4 for the CFATS cases. The same reference coal has been used for the estimation of environmental emissions as was used for the tabulated information in Table 4-2. Comparison is also made with conventional, coal-fired power plants and with conventional, natural gas-fired power plants in the table. The basis of reporting the emissions in this table is "per unit of net power output", lb/hr/MWe, rather than the usual "per unit of fuel energy input," lb/MBtu. A large advantage for the CFATS relative to commercial technology, and relative to the performance of the advanced technologies using current combustion turbines (Table 4-2) is shown in this table.

Table 4-4 - CFATS Emissions Performance

	NO _x	Emission (lb/hr/MW _e)		Particulate	Solid Waste
		SO ₂	CO ₂		
<u>CFATS</u>					
IGCC (air-blown) ^a	1.4	0.64	1409	5.4	212.3
2nd-Gen. PFBC ^b	1.3	1.55	1366	5.2	190.2
<u>Conventional</u>					
AFBC ^c	2.8	4.45	1959	22.2	257.3
PC/FGD ^c	5.7	4.45	1959	22.2	244.6
Gas Simple-Cycle ^c	0.8	---	1061	---	---
Gas Combined-Cycle ^c	0.7	---	843	---	---

a: NO_x 0.2 lb/MBtu, sulfur removal 98%, Ca/S ratio 1.8, particulate 0.01 lb/MBtu

b: NO_x 0.2 lb/MBtu, sulfur removal 95%, Ca/S 1.8, particulate 0.01 lb/MBtu

c: See Table 2 comments

IDENTIFICATION OF DESIGN CHANGES TO GFATS

While the two coal-based power systems differ significantly in their major process conditions, power cycles, process designs and equipment arrangements, both pose a similar set of requirements on the fuel gas combustion and the combustion gas expansion equipment. Both require effective use of the turbine compressor air supply with minimal compressor modifications. With both, a hot, low-Btu fuel gas is generated that must be efficiently combusted, with acceptable NO_x and CO emissions, producing the designated 2700°F combustor outlet temperature, uniformly distributed over the inlet of the turbine expander.

The GFATS consists of several functional components that must be modified for the CFATS: the air compressor, the combustor inlet fuel gas and oxidant gas manifolds, the flow controls, the combustor, the turbine inlet scroll, and the turbine expander. The GFATS compressor can be modified by standard techniques to provide the air requirements for the coal processing steps and for the low-Btu fuel gas combustor. Some 20 to 30% of the air leaving the gas turbine compressor must be boosted in pressure and ducted to the coal gasifier. Customarily, the booster compressor, with associated cooling and heat recovery exchangers, is a device independent of the

gas turbine. Low pressure loss takeoffs for gasification air, as well as for turbine cooling air, are required. Also, because higher combustion product flows and higher gas temperatures in the turbine result from replacing natural gas fuel by coal gas, slightly higher air flows may need to be extracted from the compressor for turbine cooling. This is a modification that can be accomplished by available engineering design and evaluation techniques.

A major divergence from the GFATS is the combustor design. The IGCC, low-Btu fuel gas can be efficiently combusted by internal, rich-lean combustor baskets located within the turbine cylinder, much like the GFATS, but adapted to the high temperature fuel gas conditions. The flow of air to the gas turbine combustor is reduced by some 20 to 30% due to the air flow extracted for the coal gasification process; thus, there is less air available for combustor cooling. But the maximum stoichiometric, adiabatic flame temperature associated with coal gas is much lower than that with natural gas. For this reason, the reduction in the quantity of combustor cooling air may not be critical. The lower flame temperature will reduce the formation of thermal NO_x. The volumetric flow rate of the coal gas is higher than that of the natural gas fuel by a factor from 6 to 9. The temperature of the coal gas is 1000°F; that of the natural gas, 670°F. The volumetric flow of the fuel is thus greatly increased, and redesign of the fuel nozzle and mixing arrangements is required. Flame stability in the combustor does not appear to be a problem because of the high H₂ content of the coal gas fuel. If this larger cylinder exceeds the maximum shipping dimensions, the engine would have to be shipped without the top half.

The Second-Generation PFBC combustor conditions may require that the multiple, internal combustor baskets of the GFATS be replaced by external combustor chambers, due to the large size of the combustor chambers required. Two combustion chambers, each with eight combustor modules, would be located externally to the main engine cylinder, one on each side. Hot, low-Btu fuel gas, and combustion vitiated air, would be distributed to these combustors uniformly to achieve the required combustion performance. The external combustion requires that an inlet scroll be designed, with an appropriate cooling scheme, to introduce the combustion gas into the turbine.

The combustors must be designed with minimal structural cooling, to combust the low-Btu fuel gas efficiently and with acceptable pressure drop, acceptable outlet temperature uniformity, acceptable metal surface temperatures, low CO, soot and unburned hydrocarbons, low conversion of fuel-bound nitrogen to NO_x, acceptable turndown performance, and with multiple fuel capability. (Combustors are being developed by Westinghouse under DOE/METC study Second Generation PFBC Systems R&D contract with Foster Wheeler, DE-AC21-86MC21023.)

The combustion turbine cylinder from the compressor discharge to the turbine inlet would require major design changes for the Second-Generation PFBC case. The casing inlet, manifold/inlet scroll and outlet arrangement would be modified from the GFATS to duct the combustion products by transition liners into a uniform annular flow. Engineering analysis must be performed, considering heat transfer analysis, stress analysis, pressure drop analysis and computational fluid dynamic analysis. Engine thrusts must be reviewed and thrust bearings redesigned, if required. The combustion turbine bedplate mounting will also be substantially modified, in the Second-Generation PFBC case, from the GFATS. Similar considerations of the combustion turbine casing for the IGCC case must be made, but under much less challenging

conditions. The substitution of coal gas from the KRW or the modified fluidized bed process for natural gas fuel increases the mol flow of the combustion product gases by some 13 to 24% depending on specifics of gasifier operation. This flow increase and the higher specific heat of the coal gas combustion products, with their higher CO₂ content, increases both the work output from the turbine by 15 to 28% and the temperature of the expanding gases within the turbine -- by as much as 80°F at the turbine exit. The blade design and blade cooling arrangements must be adequate to cover these increased loads and temperatures.

Special consideration must be given to the operation and control of the integrated coal gasification, hot gas cleaning, coal-fired advanced turbine system, CFATS. Operation includes start-up, load follow, and shutdown -- planned and emergency. Control includes maintaining steady optimal operation. Start-up of the plant will probably require special provisions for natural gas firing of the gas turbine(s) -- fuel nozzles and perhaps auxiliary combustors. Emergency shut down, for example on load loss, will require quick acting valves to isolate the gasification and gas cleaning systems from the turbine to prevent overspeed and blow back into the compressor. Load follow and control of the turbine with its integrated coal gasification, hot gas cleaning fuel supply will require coordinated adjustment of the coal, air, and steam feed to the gasifier. A single fuel flow valve, customary for a natural gas fired turbine, is not adequate to provide load follow and to protect the turbine from excessive combustion temperatures. A fuel flow control may need to be omitted due to the high temperatures and flows involved. In this case the coal, air, and steam valves on the gasifier feeds will be required to provide both the turbine and gasifier control functions. Flow controls may be placed in differing positions in the CFATS plant, so that they may provide control over lower-temperature streams located upstream of the major coal processing steps. For example, in Second-Generation PFBC, vitiated air control to the combustor may be achieved by air control to the fluid bed combustor, providing valves that operate at commercially available temperatures. Similarly, over speed protection valves must be supplied for the CFATS that are placed in acceptable locations that may differ substantially from the those of the GFATS. Compressor air bypass, through the turbine start-up burner, to the combustor may be used to cool the expansion gas and promote turbine speed reduction.

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 overspeed in the event of sudden load loss must 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 GFATS, the control system functions should also be similar except where changes are required to account for the considerable larger fuel flow. The control concept must be reviewed to identify modifications. Many of the control issues are being evaluated within the current advanced, coal-fuel technology development programs and do not need to be considered in the ATS program.

The turbine airfoil geometry and casing sections themselves will not be substantially modified from the GFATS, although the row 1 vane spacing must be increased to accommodate differences in flow between the GFATS and the CFATS, providing acceptable pressure reduction. The airfoil materials may also be substantially modified. The combustion products

will contain vapor and solid/liquid contaminants that are known to induce turbine aerofoil material corrosion: sodium, potassium, vanadium, lead, calcium, lithium, phosphorous, silica, chlorine, iron, copper, and nickel. These may reduce the life of key turbine components and increase system maintenance requirements. Not only must hot corrosion, induced by deposition of slag-like particles and condensable vapors, be considered, but cold end corrosion due to acidic mists, condensates and slurries must also be considered. While corrosion may be a concern in the CFATS, it should not be any more severe than it is in the current coal-fired turbine systems because the processing steps are at identical conditions, and the turbine aerofoils are cooled to the same temperature levels in both cases.

The effectiveness of hot gas filters should make turbine aerofoil erosion a secondary concern. Particle deposition may be more severe in the CFATS than in the current coal-fired turbine cases because of the higher firing temperature and resulting adherence of particles. Available on-line and off-line airfoil cleaning methods may need to be optimized for the CFATS. Specific turbine airfoil base materials and thermal barrier coatings will need to be specified for the CFATS that are tolerant of the gas contaminants both during steady operation, during off-design conditions, turndown, startup and shutdown transients, as well as during off-line conditions.

It is expected that the material for the main cylinder, horizontal joint, and vertical flange bolting will be the same as used in the GFATS. The hot gas containment system in the Second-Generation PFBC case 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.

The possible arrangement of the Second-Generation PFBC combustor system into two off-board combustor housings, as opposed to the conventional in line, on-board combustors of the GFATS present some major design and development problems. (As applicable, reference should be made to a Westinghouse study on the generic turbine design to develop a conceptual design of a combustion turbine that would perform in a PFBC, DOE/METC Contract DE-AC21-86MC23167.) The extended surface of the hot gas flow path is much greater in area than would be the surface of the transition pieces in the GFATS. All of this surface must be cooled by compressor discharge air. The cooling required will be much more than 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 discharge air will by-pass the combustion process within the combustors, thereby reducing the amount of air available for the combustion process, cooling the combustor wall surface, and the profiling of combustion products for uniform turbine inlet temperature for 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 the lack of control of the mixing process in the large volume of space between the exit of the combustor and the inlet to the turbine.

Critical structural problems exist that differ from those of the GFATS. Both stationary and rotational parts must consider short-term loading, fatigue, creep, and embrittlement failure

mechanisms. These considerations especially must be made for the combustors (primarily thermal fatigue due to overheating and shocking), and the turbine disks, blades, and nozzle vanes. The critical structural problem in the CFATS is thermal stress and 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, and inlet scroll that guides 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.

Section 5

SUMMARY

During the year covered in this report, the project team successfully completed the target goals and are well on the way to meeting and or exceeding the performance guidelines.

The following points have been developed as a result of the work performed to date:

- All of the candidate testing sites have received NEPA approvals.
- The final configuration chosen for the GFATS cycle has the potential of exceeding the required 60% efficiency, based on the LHV of the fuel.
- The CFATS studies identified two candidate cycle processes and presented performance and emissions data. A four year development program was outlined for further consideration.
- The blading materials testing program was initiated and the results of the ceramic casting processes to date have been documented.
- Clemson University completed the test rigs and instrumentation locations and probes in anticipation of near term testing. The flow field calculation programs have had some minor application anomalies, but expect to be validated without problem.
- The gas turbine flange-to-flange conceptual design is nearing completion and will be used in component and plant arrangement studies.