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**DEVELOPMENT OF A HIGH-PERFORMANCE
COAL-FIRED POWER GENERATING SYSTEM WITH
PYROLYSIS GAS AND CHAR-FIRED
HIGH TEMPERATURE FURNACE (HITAF)**

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**Quarterly Progress Report 11
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INTRODUCTION

A concept for an advanced coal-fired combined-cycle power generating system is currently being developed. The first phase of this three-phase program consists of conducting the necessary research and development to define the system, evaluating the economic and technical feasibility of the concept, and preparing an R&D plan to develop the concept further.

Foster Wheeler Development Corporation (FWDC) is leading a team of companies involved in this effort. The team consists of:

- AlliedSignal Aerospace Company—AiResearch Division
- Bechtel Corporation
- Research-Cottrell
- TRW, Inc.
- Foster Wheeler Energy Corporation (FWEC)
- General Electric Corporation.

The power generating system being developed in this project will be an improvement over current coal-fired systems. Goals have been specified that relate to the efficiency, emissions, costs, and general operation of the system. These goals are:

- Total station efficiency of at least 47 percent.
- No more than: 0.15 lb NO_x/10⁶ Btu fuel heat input
0.15 lb SO_x/10⁶ Btu fuel heat input
0.0075 lb of particulates/10⁶ Btu fuel heat input.
- All solid wastes must be benign. Generation of solid wastes is minimized through production of usable by-products.
- Over 95 percent of the total heat input is ultimately from coal, with initial systems capable of using coal for at least 65 percent of the heat input.

- Efficient and economic baseload power generation:
 - Operation with a range of U.S. coals
 - Annual capacity factor of 65 percent
 - Load following with minimal degradation in efficiency
 - Net electrical output as low as 100 MW
 - 10-percent lower cost of electricity (COE) relative to a modern coal-fired plant conforming to NSPS.
- Safety, reliability, and maintainability to meet or exceed conventional coal-fired power plants.
- Amenable to construction using factory-assembled modular components based upon standard design.

There are two basic arrangements of our HIPPS cycle. Both are coal-fired combined cycles. One arrangement is the 35% natural gas HIPPS. A simplified process flow diagram of this system is shown in Figure 1. Coal is converted to fuel gas and char in a pyrolysis process, and these fuels are fired in separate parts of a high temperature advanced furnace (HITAF). The char-fired furnace produces flue gas that is used to heat gas turbine air up to 1400°F. Alloy tubes are used for these tube banks.

After leaving the alloy tube banks, the gas turbine air goes through a ceramic air heater where it is heated from 1400°F to 1800°F. The flue gas that goes through the ceramic air heater comes from the combustion of the fuel gas that is produced in the pyrolysis process. This fuel gas is cleaned to remove particulates and alkalies that would corrode and plug a ceramic air heater. The air leaving the ceramic air heater needs to be heated further to achieve the efficiency goal of 47%, and this is done by firing natural gas in the gas turbine combustor.

An alternative arrangement of the HIPPS cycle is called the All Coal HIPPS. With this arrangement, the char is used to heat the gas turbine air to 1400°F as before, but instead of then going to a ceramic air heater, the air goes directly to the gas turbine combustor. The fuel gas generated in the pyrolyzer is used as fuel in the gas turbine combustor. A simplified process flow diagram of this cycle arrangement is shown in Figure 2. In both cycle arrangements, heat is transferred to the steam cycle in the HITAF and a heat recovery steam generator (HRSG).

Project Work

Subtask 3.29 - Design of Gas Turbine Piping

General Electric has completed the design of a piping system to adapt their Frame 7FA gas turbine for HIPPS operation. The General Electric Frame 7FA gas turbine has 14 separate combustors for fuel injection and ignition. In the HIPPS design, the standard combustor cans are removed and replaced with a manifold. The manifold facilitates the routing of the compressed air to the HITAF and the return of the heated air to the gas turbine combustors. The system is designed for an air return temperature of 1800 F which is the temperature in the 35 percent Natural Gas HIPPS. The general arrangement of the piping system will also be suitable for the All Coal HIPPS where the air return temperature is only 1400°F.

As part of the project, General Electric has done a fairly detailed analysis of their piping system. Some of this information is included in this report. Westinghouse has also developed a design for some of their gas turbines that provides the functions required for HIPPS. The Westinghouse system was developed on another DOE project, so details will not be presented here. However, the conditions in the All Coal HIPPS are very similar to those in the other project so the Westinghouse gas turbine can also be used for the All Coal HIPPS.

Design Approach

After investigating some design options, General Electric chose what is referred to as the "torus" design. This arrangement is shown in Figure 3. It consists of two separate tori as headers, one each for the compressor discharge air and the furnace return air. The return air piping penetrates the compressor discharge torus, and then travels through the discharge piping to the turbine. This arrangement is more symmetrical and compact than other options.

In the design of the manifold, the hot air return pipe metal temperatures were a prime concern. Hot air returning to the gas turbine from the HITAF will be at 1800°F in the 35 Percent Natural Gas HIPPS. This far exceeds the allowable working temperatures for steel at design pressures. Other high temperature alloys were investigated but were ruled out because of high costs and borderline mechanical properties at the design temperatures.

The hot air piping metal temperatures were reduced by using internal refractory insulation. This allowed for the use of steel for all of the pressure piping in the system. Any uninsulated piping was constructed of a high temperature alloy. These sections are not designed to be exposed to long term pressure loads. Shrouded internal refractory has been used to insulate pressure piping in other applications. In this type of design, the internal

Figure 3 Torus Design Layout

insulation is covered with an Inconel shroud. The shroud is supported by a welded cone system and is designed with joints to facilitate the relative thermal expansion between the shroud and the outer pipe.

Hardware

The torus piping design consists of the following components:

1. Header leg piping
2. Burner assembly flange
3. Compressor discharge supply torus
4. Hot air return torus
5. Supply and return piping
6. Bypass piping
7. Bypass valve system

Figure 4 shows a side-view of one leg of the system. The compressor discharge air flows up through the outer annulus of header leg, and into the inner torus. The air then flows through two exit ports into the supply piping. The supply piping connects to the furnace headers.

After heating within the furnace, the air flows through the return piping to the two ports on the hot air return torus. The torus distributes the air to the 14 header-legs that carry the flow to the combustion chamber for supplemental firing before entering the turbine section.

Figure 5 shows the components of each leg of the torus design. The first section of piping, working from the turbine to the furnace, is the header leg. This section is made up of two concentric pipes. The inner section is a 16 inch schedule 20 HS-188 pipe with slip fit expansion joints. This section is constrained by slip or compression fits with adjoining sections of pipe made with similar metals.

The outer piece is a 20 inch 0.75 inch wall SA-312 pipe modified to accept the burner-diffuser assembly. Details of this section are shown in Figure 6. The burner assembly bolts to a special rectangular flange on the top of the pipe. The lower flange is designed to bolt to the Frame 7FA combustor flange. The upper flange is a 300 pound flange that connects to the compressor discharge torus section. The next section is the

Figure 4 Torus Design Air Flow Pattern

Figure 5 Torus Design Components

Figure 6 Burner Port Details

compressor discharge torus. This section is constructed of 40 inch schedule 40 SA-312 piping, with a 0.75 inch wall thickness.

Perpendicular to the compressor discharge torus are the connections for the header-legs. Between the turbine and the torus is a 22 inch schedule 30, 0.5 inch wall, SA-312 pipe. This pipe has a 22 inch weld-neck 300 pound SA-240 flange to connect to the header leg section. On the outer half of the 40 inch torus pipe is a 30 inch schedule 30, 0.625 inch wall, SA-312 pipe. This pipe has a 30 inch weld neck, 300 pound, SA-182 flange to connect to the hot air torus piping. The area where each of these pipes penetrate the torus are supported by 0.75 inch, SA-240 reinforcing pads.

Passing through the center of the header-legs is the hot air return piping. This is another section of 16 inch schedule 20 HS-188 pipe which flares out to 20 inches as it enters the outer header leg. This pipe has a 50 degree bend to match the angle of the header-legs, and is centered by supports between the two pipes. The 50 degree angle was added to move the hot air return torus away from the gas turbine's air intake as shown in Figure 7. If the piping had remained straight, the hot air return torus would obstruct the inlet air path, and the inlet air would absorb heat transferred from the piping. This would result in decreased output and efficiency because of increased compressor work or decreased compressor efficiency.

The outer header leg is a 30 inch pipe with 4 inches of internal cast refractory insulation. The insulation protects the SA-312 steel piping from the 1800°F air temperature returning from the furnace. The insulation is not used in the concentric piping where there are low pressure loads and lower average metal temperatures because excess heat is carried away by the compressor discharge air.

The hot air return torus is constructed of 48 inch, schedule 40, 0.75 inch wall SA-312 piping. The necessary metal temperatures are obtained by using 4 inches of insulation on the inside of the hot air return torus. A 0.75 inch, SA-240 reinforcing pad was used at the connection of hot air return torus to the header leg piping.

The individual torus sections will be welded together into four sections as shown in Figure 8. Two four-leg sections with supply ports, and two three-leg sections will be joined by bolted flanges. Each section will be equipped with lifting lugs to facilitate assembly and disassembly of the piping.

The supply and return piping layout, and bypass piping and valve system is shown in Figure 9. This set-up allows the compressor discharge air to be routed around the furnace at start-up and in case of a load disturbance. Without the bypass piping, the energy added by the furnace, which can not be quickly shut-down, could cause an overspeed incident on the unloaded gas turbine.

Figure 7 Design Cross-Section

Figure 8 Assemble Layout

Figure 9 Piping Layout

The piping with internal insulation was sized large enough to allow for internal human inspections, when necessary, as shown in Figure 10.

Several valves are provided in the system for control and protection purposes. For fast operation, all of the valves are butterfly valves with hydraulic actuators. The furnace supply valve (normally closed) is opened when the compressor air is ported to the furnace. The furnace bypass valve is slowly closed during start-up, and opened quickly during an emergency trip. Blow-off valves are provided to reduce the possibility of compressor surge.

Vendor inquires regarding valve construction and operational capabilities concluded that a 1 second cycle time is possible for butterfly valves up to 48 inches in diameter.

Design Analysis

Optimization of the piping design had two basic goals in mind:

1. The design had to have as small a pressure drop as possible.
2. The design had to limit the heat losses.

All heat transfer and pressure drop calculations were performed using the "General Electric Heat Transfer and Fluid Flow Data Books", published by Genium Press, Schenectady, New York.

These data books were originally developed by the General Electric Corporate Research and Development Center. Additional concerns included material and dimensional constraints, as well as the method of construction.

Heat Transfer. The heat transfer design goals were basic: lower the metal temperature to levels where the material properties are acceptable, and reduce the heat loss from the internal air to the surroundings. The most straight-forward means of attaining these goals is through the application of insulating materials. To lower the metal temperatures, the insulation had to be installed between the hot air flow and the metal, or internal to the pipe. As insulation is added to the inside of the pipe, either the velocity increases, or the pipe diameter must be increased. As the pipe diameter increases, the pipe wall thickness must increase due to stress concerns. Therefore, the optimum selection of the insulation thickness and pipe diameters must account for heat losses, pipe metal temperatures, pressure drops, and pipe stresses.

Figure 10 Internal Sizing

As insulation is added to the inside of the hot air return torus, the amount of heat loss decreases rapidly, as illustrated in Figure 11. This figure also shows that there is little difference in the amount of heat loss with insulation thicknesses above 4 inches.

Table 1 lists the air properties that were used in the heat transfer analysis. The transfer of heat from the hot air return flow (1800°F) to the atmosphere considered the following heat transfer media:

1. Forced convection on the inside of the pipe.
2. Conduction through the insulation.
3. Conduction through the pipe wall.
4. Free Convection on the outside of the pipe.
5. Radiation from the pipe wall to the ambient.

The procedure for calculating the pipe metal temperatures and heat transfer to the ambient necessitated several iterations around the average insulation and metal temperature. The outer diameter of the piping was increased to allow for a constant cross-sectional flow area. It was assumed that the ambient temperature would be in the range of 70°F to 200°F. Table 2 and 3 summarize the pipe temperatures and heat losses for an ambient temperature of 70°F and 200°F. Figure 12 shows the average pipe metal temperature as a function of inside insulation thickness. Table 4 shows the expected temperature profile between the hot air return flow and the ambient.

The effect of radiation probably has the highest degree of uncertainty in the heat transfer calculation. The emissivity varies between 0.09 for a shiny aluminum jacket to 0.9 for a flat matte finish. Most high temperature piping in gas turbine applications are painted with a high temperature silver paint. This paint usually ends up peeling off after several hours of operation. For the hot air return torus, the design point emissivity was assumed to be 0.2. Figure 13 illustrates the effect of the emissivity on the pipe wall metal temperatures. Calculated pipe temperatures as a function of the emissivity are shown in Table 5. With 4 inches of inside insulation, 1 inch of outer insulation, and assuming an emissivity of roughly 0.2, the pipe metal temperature will be in the range of 275°F to 475°F, depending on the ambient temperature and the actual emissivity of the outer pipe surface.

Figure 11 Heat Losses vs. Insulation Thickness

Table 1 Air Properties

Parameter	Symbol	Units	Cold Compressor Discharge	Hot Heater Out
Total Flow	m	lb/hr	2514300	2514300
Pressure	p	psia	238.07	216.47
Temperature		°F	796	1800
Density	rho	lb/ft ³	0.5119	0.2587
Viscosity	mu	lb/hr-ft	0.0808	0.1157
Thrm Conduct	k	Btu/hr-ft-F	0.0295	0.0451
Specific Ht.	Cp	Btu/lbF	0.2569	0.2825
Prandtl No.	Pr		0.6948	0.7230
Heat Flow	q	kBtu/hr	468,941	1,228,801

Table 2 Pipe Temperatures and Heat Losses. (70°F ambient)

Inner Insul. Thickness in	Pipe Temp. °F	Skin Temp. °F	q kBtu/hr	(Heat Loss/ Heat Added) *100%
0.0	1579.7	1565.6	14971	1.97%
0.5	891.7	887.6	3550	.46%
1.0	678.4	675.9	2051	0.27%
2.0	473.9	472.5	1105	0.14%
3.0	370.2	369.2	751	0.09%
4.0	307.7	306.9	568	0.07%
5.0	266.2	265.6	456	0.06%
6.0	236.8	236.3	381	0.05%
8.0	198.1	197.7	287	0.03%
10.0	173.7	173.4	231	0.03%

Table 3 Pipe Temperatures and Heat Losses (200°F ambient)

Inner Insul. Thickness in.	Pipe Temp. °F	Skin Temp. °F	q kBut/hr	Heat Loss/ Heat Added *100%
0.0	1582.0	1568.0	14818	1.95%
0.5	921.9	917.9	3481	0.45%
1.0	729.4	727.0	2010	0.26%
2.0	553.5	552.1	1087	0.14%
3.0	466.6	465.6	743	0.09%
4.0	414.0	413.3	564	0.07%
5.0	378.7	378.1	454	0.06%
6.0	353.2	352.7	380	0.05%
8.0	318.9	318.5	288	0.03%
10.0	296.8	296.6	232	0.03%

Figure 12 Pipe Temperature vs. Inside Insulation Thickness

Table 4 Temperature Profile (4 inch inside insulation, 200°F ambient)

Temperature (°F)	Location
1800.0	Center of Pipe
1791.5	Insulation hot side
1103.1	Insulation Average
414.8	Insulation cold side/pipe hot side
413.3	Pipe cold side
200.0	Ambient

Table 5 Pipe Temperatures as a Function of Emissivity (4 inch inside insulation, 1800°F pipe)

	Emissivity	Avg Pipe Temperature (°F)	
		70°F ambient	200°F ambient
Design:	0.01	435.7	607.4
	0.1	354.3	475.9
	0.2	306.9	414.0
	0.5	238.9	334.7

Flow Velocities. The air flow velocity through the piping system was calculated. These velocities were important for two reasons. First, the pressure drop through the piping is a function of the air velocity, and secondly, air velocity will have an effect on erosion of any internal insulation.

The velocity of air flow through each section of pipe is a function of the cross sectional area and the volumetric flow rate. Table 6 lists the calculated air speeds through different sections of the piping design.

Figure 13 Pipe Temperature vs. Emissivity

Table 6 Air Velocity by Section

Section	Velocity (ft/s)	Air Temp (°F)
1. Compressor Discharge	207	735
2. Discharge Entering Torus	97	735
3. Piping to Furnace	91	735
4. Return Piping	99	1800
5. Entering Return Headers	80	1800
6. Entering Combustors	150	1800

The maximum velocity of 207 ft/s at the compressor discharge piping is constrained by the sizing of the outer pipe flanges, the stress levels in the combustor access flange area, and the velocity requirements for air entering the combustor. These velocities are considered acceptable because there is no internal insulation in this area, and the length of piping over which the high velocities occur is relatively short. The velocities in all of the insulated pipe sections, namely sections 4 and 5, are all relatively low.

Pressure Drop. One of the goals for the piping design was to minimize the pressure drop. The pressure drop in the piping contributes to the system losses.

The pressure drop analysis was performed on the system by sections. Each section pressure drop was calculated based on its individual dimensions. The following effects were considered when calculating the piping system pressure drop:

1. Length effects
2. Expansion from smaller to larger areas
3. Separation of flows in cold header
4. Mixing of flows in cold header
5. Separation of flows in the hot header
6. Bends and elbows
7. Reduction of larger to smaller areas

It was assumed that the total mass flow is distributed equally among the 14 connections to the torus headers. It was also assumed that the total flow is split equally between the

two header pipes that connect between the torus piping and the furnace.

Table 7 provides a summary of the pressure drops of the manifold piping, as well as estimates of the remainder of the piping system. The combustor pressure drop shown in Table 7 is the pressure drop between the compressor discharge to the compressor air torus. Pressure drop due to the fuel nozzles was not considered.

Table 7 Summary of Pressure Drops

	Piping Pressure Drop (psi)
Manifold Piping	2.64
Combustor	3.04
Piping to/from heater	1.90*
Furnace	10.00*
Overall Pressure Drop	17.58

*From Plant Analysis

Stress Analysis. Pipe stresses were addressed by both static stress studies (ASME Section VIII Division I Pipe Code) and finite element analyses. The static stress calculations were used to develop the minimum thicknesses required for each section of pipe. Once the overall dimensions were known, the finite element analysis provided refinements to the design, and determined shape specific stress levels.

The finite element model of the piping design was developed using the COSMOS/M finite element package developed by Structural Research and Analysis Corporation of Santa Monica, California. The model geometry was generated from AutoCAD files. The geometry consisted of a half model of the torus system. The model included all of the external piping, but ignored the internal concentric piping such as the hot air distribution piping to the combustor. Due to model and element size restrictions, the following features were not modeled:

1. Flanges
2. Torus-to-header connections
3. Burner insert flanges

The flanges and connections were designed based on the ASME Section VIII Code B31.1

Series. The thickness of the header leg between the turbine and the compressor air torus was increased to simulate the additional stiffening provided by the burner flange and assembly.

The element mesh consisted of two-dimensional thick-shell elements. This type of element is appropriate for modeling pipes and other geometry where the thickness of the piece is small in comparison to its other dimensions. Both three-node triangular, and four-node rectangular elements were used in the mesh. These shell elements have the full three translational and three rotational degrees of freedom at each node.

The material properties used in the analysis were the properties of 403 stainless steel. These properties were included in a standard material property library supplied with the finite element program.

The boundary conditions that were used with the model included an axial constraint at the gas turbine combustor flange, symmetric restraints across the X-Y plane to model the second half of the system, and constraints to prevent rotation around the central axis.

Steady state thermal and linear static analysis were conducted. Thermal loads were considered for a preliminary analysis, and both thermal and pressure loads were considered for the secondary analysis. The thermal load cases included:

Case 1. Temperatures of 796°F on the compressor air torus and a temperature of 484°F on the hot air return torus. The 484°F temperature simulates 1800°F air with 4 inches of internal refractory insulation and no outer insulation. This also assumes a 70°F ambient temperature.

Case 2. Temperatures of 796°F on the compressor air torus and a temperature of 772°F on the hot air return torus. The 772°F temperature simulates 1800°F air, with 4 inches of internal refractory insulation and 1 inch of outer insulation. This also assumes a 70°F ambient temperature.

Case 3. Temperatures of 796°F on the compressor air torus and a temperature of 832°F on the hot air return torus. The 832°F temperature simulates 1800°F air with 4 inches of internal refractory insulation and 1 inch of outer insulation. This also assumes a 200°F ambient temperature.

Case 4. Further points were considered in an attempt to determine the maximum temperature difference between the two tori that would generate acceptable stress levels.

Figure 14 shows the steady-state temperature distribution for the first thermal load case.

The acceptable stress levels adopted for these analyses were taken from the ASME Section

Figure 14 Piping Temperature Distribution

VIII, Division 2, Part D allowable stress tables for SA-312 stainless steel. The allowable stress level was set at 15,300 psi, assuming welded pipe at 484°F for the first case. Allowable stresses for the higher temperature components were set at 13,500 psi assuming welded pipe at approximately 800°F.

Analysis of the first thermal load case without pressure forces, resulted in a maximum stress of 41,400 psi on the inner torus (Figure 15, Figure 16) This greatly exceeded the allowable stress level of 13,500 psi. The high stresses were apparently due to the relative displacement differences caused by the thermal expansion of the same material to two temperatures that were roughly 300°F apart.

The subsequent load cases were developed in response to the above results. The high stress levels were caused by the large temperature difference between the hot air return and compressor air tori. With four inches of insulation on the inside of the hot air return torus piping, the metal temperature of the hot air return torus is less than the compressor air-torus piping. To reduce the stresses due to thermal loading, the temperature of the hot air return pipe should be controlled to be close to the compressor air torus metal temperature in one of three ways:

1. Reduce the amount of insulation on the inside of the hot air return piping
2. Add insulation to the outside of the hot air return piping
3. A combination of both of the above.

Since option 1 increases the amount of heat transfer, it was decided to add insulation to the outside of the hot air return piping. This practice was not encouraged from a maintenance standpoint, since outer insulation limits the use of an optical pyrometer, which can be used to inspect the integrity of the inside insulation.

Simulation of outer insulation was added to the analysis in an attempt to match the metal temperatures and thermal expansion more closely.

The analysis of cases 2 and 3 gave maximum stress levels of 6,000 to 8,000 psi. The patterns of the maximum stresses were similar to those revealed in the first analysis. The maximum temperature difference between the tori with acceptable stress levels was calculated to be between 75°F and 85°F.

Figure 15 Thermal von Mises Stress Levels at Original Design Point

Figure 16 High Thermal Stress Regions at Original Design Point

After the thermal load cases were completed, pressure forces were added to the model. Table 8 summarizes the pressure forces applied to the model. The analysis of case 2, with temperatures of 772°F and 796°F, gave a maximum stress value of 14,000 psi (Figure 17, Figure 18). The high stress region was localized to an area of the model around the gas turbine combustion chamber flanges, that may have had boundary condition effects.

Table 8 Finite Element Model Pressure Forces

Location	Pressure
Hot air return piping	211 psi
Compressor discharge torus	233 psi
compressor discharge header leg	238 psi

The maximum stress value in the body of the torus was roughly 13,000 psi. This indicates that the stress levels in service may be marginal, based on the allowable stress limits. The steady state stresses should be acceptable, but any transient, including start-up and shut-down may cause localized yielding of the material.

Figure 19 summarizes the peak stresses in the torus design as a function of the temperature difference between the two tori. The temperature difference shown on the x-axis is the hot air return-piping temperature minus the compressor air-piping temperature. The peak stress is minimized when the hot air return piping is about 20°F colder than the compressor discharge (cold) piping.

The initial results show that the torus piping is above the allowable stress limits under all conditions. It is important to remember that the stress limits are from the ASME Section VIII Piping code which use a 50% joint efficiency factor. By utilizing x-ray inspections of all of the welds, the joint efficiency factor can approach 100%. This would result in an increase in the allowable stress levels for this material.

Areas for Further Study. During the gas turbine start-up, the temperature of the compressor air pipe rapidly approaches the compressor discharge temperature (750°F). The hot air return piping is insulated and therefore does not approach its operating temperature until a short time after coming on-line. This temperature differential causes a stress concentration between the hot air return and compressor air tori. Although this over-stressing only occurs during the start-up or shut-down, it is cause for concern. An effort will be made to find ways to limit these peak transient stresses.

Figure 17 Combined Von Mises Stress Levels at Optimum Point

Figure 18 High combined Stress Regions at Optimum Point

Figure 19 Peak Stresses vs. Torus Metal Temperature Difference

Another area that needs further investigation is the plant control logic. Many factors are critical to the safe operation of the plant. For example, during a generator trip, to protect the gas turbine from overspeeding, the bypass valves open and the furnace supply valves close. The thermal dynamics of the HITAF and the gas turbine piping need to be analyzed in more detail to provide protection for both of these systems.

Summary. General Electric has developed a feasible design adapt a Frame 7FA gas turbine for HIPPS operation. The design was based on the conditions of the 35 Percent Natural Gas HIPPS with an air temperature of 1800°F from the HITAF. The stress analysis of the initial design indicates that the stress levels are marginal, but the mechanical design has not been optimized based on the initial stress analysis. Also, a joint efficiency of only 50 percent was used in the analysis. With x-ray inspection of the welds this efficiency can approach 100 percent.

The operation of this type of system with the All Coal HIPPS was not investigated in the present study, but in the All Coal HIPPS, the air from the HITAF will only be at 1400°F. This situation should have beneficial affects on the design.

More work needs to be done on the thermal dynamics of the entire system to ensure that all possible transients have been considered. There are design options in the valving and control of the system that can be used to minimize the effects of upset conditions. In this area also, the situation should be better with the All Coal HIPPS. With the HITAF outlet air at 1400°F, there is an opportunity for valving at the outlet of the HITAF. Even if valves at this temperature are not used to protect the turbine from overspeed, they can be used to protect the HITAF from damage.

In addition to the GE design, Westinghouse has developed a design for their 501 gas turbine that is intended for use in Second-Generation PFB systems. In these systems, the conditions at the gas turbine are similar to the All Coal HIPPS. The main difference is that the combustion air in HIPPS does not need to be cleaned. This situation can only help matters, and it is one benefit of the HIPPS approach.

Subtask 3.31 - Char Combustion Laboratory Testing

Overall Objectives

The overall objectives of the char combustion laboratory test activity are as follows:

- Determine FWDC char flame ignition and flame anchoring characteristics as a function of precombustor design and operating parameters, including air preheat temperature and oxygen content, stoichiometry, fuel assist, char particle size, burner swirl, injector configuration, injection velocity, and chamber residence time. Compare with baseline parent coal results.
- Characterize FWDC char combustion rates, carbon burnout, and char NO_x emission characteristics under high temperature conditions representative of the char combustor. Compare with parent coal results.

The purpose of the first objective is to provide information to help evaluate the feasibility of a char-fired precombustor. The baseline char combustor design concept consists of a coal-fired precombustor and a char-fired main stage. This is a conservative approach. Using char as the precombustor fuel will likely simplify the fuel preparation system and reduce costs. The test results from this task, along with supporting analytical model calculations, will help determine whether char can be used as the precombustor fuel, and will help identify necessary changes to the precombustor design to ensure successful operation.

The char combustion tests will also provide essential information on char combustion rates and NO_x emission characteristics (relative to the parent coal). These tests represent the next logical step following bench-top char characterization tests at BYU, and represent an intermediate development step prior to proceeding with pilot-scale char combustor tests at the 20-40 MMBtu/hr level. Data from the tests will be used to update the existing TRW char combustor analytical model, which is used for combustor performance predictions and scaling purposes.

Technical Approach

The char combustion experiment will be located at TRW's M1-J Combustion Technology Laboratory. A schematic of the experimental apparatus is presented in Figure 20. The test set-up includes a coal/char feed system, a gas-fired vitiator (or air heater), a coal/char burner assembly, a refractory-lined combustion chamber, a water tempering chamber, and a high efficiency baghouse for particulate capture. Air preheat temperatures of up to 1150°F will be used. Flame ignition and anchoring will be characterized based both of

Figure 20 Schematic of The Experimental Apparatus

visual observation and gas composition and temperature measurement. Char combustion rates will be determined based on gas composition and temperature measurements along the combustor length. Emissions monitoring equipment will measure CO, CO₂, O₂, and NO_x.

Three test phases are planned. Initially, the basic test hardware (combustion chambers, exhaust duct, and baghouse) will be hot-fired tested with natural gas. This will also provide a hot-fired checkout of the natural gas burner, cooling water system, air system, tempering system, gas sampling probes, and related instrumentation. Once the basic operation of the system is verified, the coal feed system, gas-fired vitiator, and coal-fired burner will be installed and checked out under hot-fired conditions. These tests will provide an opportunity to obtain baseline combustion data for the parent coal. Once coal-fired operation is verified, tests will be conducted with pyrolyzer char obtained from Foster Wheeler. Test parameters that can be varied during char-fired operation include air preheat temperature and oxygen content, burner and overall stoichiometry, natural gas assist, char particle size, burner swirl, char injection velocity, and overall fuel firing rate.

Detailed test activities for the gas-fired checkout testing are listed in Table 9, along with detailed objectives of each test activity. Test activities and objectives for the coal-fired and char-fired phases of the test program are listed in Table 10.

Description of Experimental Apparatus

An elevation view of the HIPPS char combustion laboratory test set-up is presented in Figure 21. The test hardware consists of the combustion chambers, the start-up gas burner, the gas-fired vitiator, and the coal/char burner. The test support equipment includes the natural gas delivery system, the coal/char feed system, the combustion air and oxygen systems, the tempering system, the sodium bicarbonate feed system, the baghouse and exhaust system, the forced and natural cooling systems, and the gas sampling system. Each system is described in the following two subsection sections.

Test Hardware

Combustion Chamber. The combustion chamber assembly consists of five refractory-lined chambers. Each chamber is constructed with two concentric pipes (18 inch and 24 inch), forming a 3-inch wide cooling annulus. The chamber internal diameter is 10 inches, with a refractory thickness of approximately 4 inches.

The first two sections are identical in construction and contain 10 diagnostic ports each for either temperature measurement, gas sampling, or visual observation. The third section is designed for secondary air injection, and contains six air injection ports at the upstream

Table 9 HIPPS Char Combustion Test Activities - Gas-Fired Config

TEST ACTIVITY	TEST OBJECTIVE(S)
. Functional Checkout of Gas-Fired Configuration	<ul style="list-style-type: none"> . Verify function capabilities of the following systems: <ul style="list-style-type: none"> . Compressor air system . Blower Air system . Baghouse system . Tempering system . Forced cooling system . Natural cooling system . Obtain all necessary correlations for flow conditions . Calibrate all instruments . Leak check all components . Natural gas system . Burner control system . Emissions monitoring system . Data acquisition system
. Refractory Curing	<ul style="list-style-type: none"> . Verify light-off procedures and low temperature burner operation . Slowly heat refractory to 1250°F to remove all chemically bounded moisture. Use gas burner.
. Hot-Fire Checkout of Gas-Fired configuration	<ul style="list-style-type: none"> . Verify start-up burner operation up to 500,000 Btu/hr . Verify adequacy of cooling system . Verify tempering system operation . Verify baghouse operation up to 300°F . Verify gas sampling and temperature measurement systems . Verify secondary air injection . Verify/modify gas-fired operating procedures . Verify burner system safety shut-off . Verify data acquisition system . Verify/modify manual alarms

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Table 10 HIPPS Char Combustion Test Activities—Coal-Fired and Char-Fired Configuration

TEST ACTIVITY	TEST OBJECTIVE(S)								
. Functional Checkout of Coal-Fired Configuration	<ul style="list-style-type: none"> . Verify function capabilities of the following systems: <ul style="list-style-type: none"> . Vitiator system . Coal feed system . Coal-burner cooling circuits . Sodium bicarbonate feed system . Calibrate all additional instruments . Leak all new components 								
. Baseline Coal Firing	<ul style="list-style-type: none"> . Verify hot-fire start-up burner operation in new configuration . Verify/characterize vitiator operation . Verify coal feed system operation under hot-fired conditions . Verify/modify coal-fired operating procedures . Verify burner control operation while firing coal . Characterize baseline coal-fired burner performance in terms of gas temperatures, carbon burnout, and no_x emissions. 								
. Baseline Char firing	<ul style="list-style-type: none"> . Verify coal feed system operation with char . Verify burner control operation while firing char . Characterize baseline char-fired burner operation in terms of gas temperatures, carbon burnout and no_x emissions 								
. Parametric coal and Char Tests	<ul style="list-style-type: none"> . Characterize burner performance as a functions of some or all of the following parameters: <table> <tr> <td>. Air preheat temperature</td><td>. Burner swirl</td></tr> <tr> <td>. Burner stoichiometry</td><td>. Injection velocity</td></tr> <tr> <td>. Firing rate</td><td>. Fuel assist</td></tr> <tr> <td>. Particle size</td><td></td></tr> </table> 	. Air preheat temperature	. Burner swirl	. Burner stoichiometry	. Injection velocity	. Firing rate	. Fuel assist	. Particle size	
. Air preheat temperature	. Burner swirl								
. Burner stoichiometry	. Injection velocity								
. Firing rate	. Fuel assist								
. Particle size

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Figure 21 Elevation View of Char Combustion Test Set-up

end. The last two sections, are designed for post combustion diagnostics and/or tempering water injection. The baseline location for the tempering water injector is the fifth section.

Each chamber is flood-cooled with ordinary industrial water from the main water tank. During steady state operation, a bulk boiling condition exists within the chamber cooling annulus, maintaining chamber metal temperatures at approximately 200°F. Steam generated in the cooling passages is vented to the atmosphere through fifteen 12 inch vents (3 per chamber).

Start-up Burner. The start-up burner will be used both during gas-fired operation and coal-fired operation. During initial gas-fired checkout testing, the start-up burner will be installed in the burner headend plate, and will be used for hot-fired checkout of the combustion chambers, tempering system, exhaust ducting, baghouse, and the instrumentation and control systems. During coal-fired operation, the start-up burner will be used both to warm up the chamber refractory prior to coal light-off, and as a pilot burner during coal light-off and initial operation.

Gas-Fired Vitiator. The vitiator is used to preheat the primary combustion air to up to 1150°F prior to entering the coal burner. The vitiator assembly consists of a pilot burner, main burner, flame rod, observation port, and air check valve. The burner maximum rating is 500,000 Btu/hr, with a 40:1 turndown. Nominal firing rate is approximately 90,000 Btu/hr.

Coal/Char Burner. The coal/char burner assembly consists of a refractory-lined, water-cooled outer chamber, an uncooled burner combustion can, and a refractory-lined endplate. Two tangential air inlets are located at the upstream end of the burner can. The air inlets are equipped with individual swirl vanes, which can be adjusted to vary the burner swirl as desired. The start-up burner is positioned in the center of the burner can, surrounded by the six individual coal/char injectors. The final section of the injectors can be removed and modified if necessary. The burner throat is formed with poundable refractory, and can also be modified as deemed necessary during testing.

Support Systems

Natural Gas Delivery System. The natural gas system has been designed based on guidelines provided in NFPA Publication 85A-1982, "Standard for Prevention of Furnace Explosions in Fuel- and Natural Gas-Fired Single Burner Boiler-Furnaces". Automatic

shut-off valves are installed in both the start-up burner and vitiator lines to ensure safe burner operation. High and low pressure switches are also provided in each line which will shutdown burner operation in the event of regulator failure (over-pressurization) or loss of gas flow (under-pressurization).

Coal/Char Feed System. The coal/char feed system consists of a coal run tank and screw feeder, a electronic scale and weigh platform, coal transport lines, an automatic fire valve, and a six-way coal splitter. Filtered air from the compressor is used to transport the coal or char to the burner.

Air Systems. The primary air to the start-up burner is provided by a 10-hp compressor, while the remaining combustion air is provided by a 5-hp blower. Differential pressure switches are located in each line to ensure that adequate combustion air is continuously supplied during operation. Oxygen from a gas bottle is also injected into the vitiator air stream as needed to control the initial air oxygen content in the coal/char burner.

Tempering System. The tempering system is used to lower the temperature of the burner combustion products to approximately 300°F. This is required for safe baghouse operation. The system consists of a air-assisted water atomizing nozzle, a temperature controller, and a pneumatic flow control valve.

Sodium Bicarbonate System. Sodium bicarbonate is injected in the exhaust stream to reduce SO₂ emissions. The system consists of a powder feeder, a pneumatic transport (GN₂) line with a control valve, pressure gauge, and sonic metering orifice.

Baghouse / Exhaust System. The baghouse is used to remove approximately 99.8% of the particulate from the burner exhaust stream. Baghouse was manufactured by C.P. Environmental, and is rated for use for up to 700 ACFM. High temperature fire-resistant Ryton bags are used, which allows operation up to 350°F. A 20" x 30" pressure relief door, designed to open at 20" H₂O, is located on the north side of the baghouse (pointed away from the test area). A 55 gallon drum is installed below the baghouse to collect solids. The exhaust stack is 25 feet high with a 3 inch diameter outlet, in order to meet AMQD stack velocity requirements.

Cooling Systems. The forced cooling system is used to provide cooling water for the gas sampling probes, the start-up burner endplate, and the outer chamber of the coal burner. The natural cooling system is used to cool the five refractory-lined combustion chambers.

Under steady state and cool-down conditions, steam will be generated within the cooling annulus of each combustion chamber and is exhausted into the atmosphere.

Gas Sampling System. The gas sampling system has the capability to determine O_2 , CO , CO_2 , and NO_x concentration. Up to five different sampling locations may be used during a given test, with samples taken sequentially from each location during steady state operation.

Operating Conditions

A process flow diagram is provided in Figure 22 which includes all process and cooling water flows during char-fired operation. Table 11 lists process flows, temperatures and pressures during nominal char-fired operation. Key operating conditions that will be varied during coal-fired and char-fired tests include air preheat temperature and oxygen content, burner stoichiometry, and coal/char firing rate.

The burner and combustion chambers have been designed to simulate the temperature-time profiles within both the precombustor and slagging stage of the char combustor. Figure 23 is a plot of expected gas temperatures during operation at 500,000 Btu/hr for various burner stoichiometries ($\phi = 0.6, 0.75, 0.9, \text{ and } 1.2$). By adjusting either the stoichiometry, or firing rate, or both, temperature-time profiles in either the precombustor or slagging stage can be simulated. This will allow investigation of both flame stability/anchoring and carbon burnout.

Figure 22 Process Flow Diagram of Char Combustion Test Set-up

Table 11 Heat and Material Balance - Hipps Char Combustion Experiment

	NOMINAL
Char Thermal Input (Btu/hr)	5000000
Pilot NG thermal Input (Btu/hr)	0
Vitiator Ng Thermal Input (Btu/hr)	82019
Total Thermal Input (Btu/hr)	582019
Primary Stoichiometry	0.75
Overall Stoichiometry	1.2
(O/F°) Char	1.92
(O/F°) Natural Gas	4
Char Ash Content (%)	26.13
HHV Char (Btu/lb)	10557
HHV Natural Gas (Btu/lb)	24000
Solids/Gas Ratio	1
Vitiator Preheat Temperature (°F)	1100
Vitiated Air O ₂ Content (%)	19
Temperature following Secondary Combustion (°F)	1998.7
Temperature at Baghouse (°F)	300
Stack Gas Temperature (°F)	200

Table 11 Heat and Material Balance - Hipps Char Combustion Experiment (continued)

Stream No.	Stream Name	Mass Flow (PPH)	Temp (°F)	Pres (*H2O)	Stream Comp	Additional Info.
1	Char	47.36	70	30	C	
2	Carrier Air	47.36	70	30	A	
3	Char and Carrier Air	94.72	70	30	C + A	
4	Natural Gas Total	3.42	70	5 PSIG	NG	1.40 SCFM
5	Vitiator Natural Gas	3.42	70	5 PSIG	NG	1.40 SCFM
6	Pilot Natural Gas	0.00	70	5 PSIG	NG	0.00 SCFM
7	Primary Air	276.47	70	20	A	62.73 SCFM
8	NO _x	3.00	70	20	O	36.77 SCFH
9	O _x enriched Air	279.47	70	20	A + O	
10	Vitiated Air	282.89	1100	15	NG + A + O	
11	Total Primary Flow	377.61		10	C + NG + A + O	
12	Secondary Air	202.03	70	10	A	45.84 SCFM
13	Primary and Secondary Flow	579.64	1998.7		C + NG + A + O	570.85 ACFM
14	Tempering Water	250.17	70	10 PSIG	W	0.50 GPM
15	Atomizing Nitrogen		70	10 PSIG	N	
16	Total Exit Flow	829.81	300	5	C + NG + A + O	300.37 ACFM
17	Stack Flow	817.44	200	0	C + NG + A + O	256.95 ACFM
18	Ash Flow	12.38	200	0	ASH	
19	Makeup Water	286.19	70	0	W	0.57 GPM
20	Steam Formed, Primary Sect.1	66.29	212	0	W	0.13 GPM
21	Steamed Formed, Primary Sect.2	75.94	212	0	W	0.15 GPM
22	Steam Formed, Secondary Sect.1	64.67	212	0	W	0.13 GPM
23	Steam Formed, Secondary Sect.2	52.87	212	0	W	0.11 GPM
24	Steam Formed, Water Tempering Sect.**	26.44	212	0	W	0.05 GPM
25	Total Steam Flow	286.21	212	0	W	

*C - Coal, A - Air, NG - Natural Gas, N - Nitrogen, O - OX, W-Water

**Estimate as Half of Spool 4

Figure 23 Expected Burner Temperature Profiles