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Generic Turbine Design Study

Final Report

June 1993

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

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

By
Westinghouse Electric Corporation
Power Generation Business Unit
Orlando, Florida

MASTER

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Office of Fossil Energy
Morgantown Energy Technology Center
P.O. Box 880
Morgantown, West Virginia 26507-0880**

**By
Westinghouse Electric Corporation
Power Generation Business Unit
4400 Alafaya Trail
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June 1993

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EXECUTIVE SUMMARY

The purpose of Task 12, Generic Turbine Design Study, was to develop a conceptual design of a combustion turbine system that would perform in a pressurized fluidized bed combustor (PFBC) application. A single inlet/outlet casing design that modifies the W251B12 combustion turbine to provide compressed air to the PFBC and accept clean hot air from the PFBC was developed. Performance calculations show that the net power output expected, at an inlet temperature of 59°F, is 20,250 kW.

Since the selected application for this study required a reduced airflow requirement for the compressor, certain modifications to the compressor were required. These modifications are not necessary for a standard W251B12 application.

Removing the first two stages of compressor blading, reshaping the inlet scroll, modifying the blade design and repositioning the IGV is a feasible design approach to obtaining the necessary reduction in mass flow from the modified W251B12 combustion turbine. The compressor inlet flow was reduced from 375 ft/sec to 229 lb/sec, the pressure ratio was decreased from 15.5 to 11.6, and the overall thermal efficiency is nearly the same for both compressors.

A single inlet/outlet design with the piping located on the lower right-hand side of the casing is recommended to reduce the design impact on the bedplate. The total pressure loss from the compressor diffuser exit to the pipe leading to the fluidized-bed combustor is approximately 2.9 psi, which compares favorably to the unmodified unit. An internal manifold was designed to route inlet air from the PFBC to the current transitions and into the turbine. The casing penetrations satisfy ASME Boiler and Pressure Vessel Code rules.

The current WDPF control system offered by Westinghouse for the W251B12 is acceptable for the PFBC application. Modified versions of standard Westinghouse fuel skids will be required for either fuel oil or natural gas. A combination heating and cooling system must be installed in the intake system, and the exhaust system will contain a heat recovery steam generator.

For this PFBC application the combustor baskets were removed from the engine. Off-engine combustors are required for turbine and fluidized bed startup. An off-engine startup combustor concept was developed along with a detailed bill of materials. Many of the current combustor components can be used in this application, including the combustor basket, fuel, nozzles, igniters, and steam and water injection system (if necessary).

The pipe sizes defined are reasonable for power plant piping. The velocities and pressure drops are not excessive. Pipe wall stresses are acceptable and insulation material recommendations are based on prior successful operating experience.

The result of this successful study is a versatile conceptual design capable of meeting PFBC requirements.

INTRODUCTION

The purpose of Task 12, Generic Turbine Design Study, was to develop a conceptual design of a combustion turbine that would perform in a pressurized fluidized bed combustion (PFBC) application. For the purposes of this study the pressurized circulating fluidized bed (PCFB) system designed by Pyropower was the reference bed application. To develop a reference plant configuration, the study was further refined by using the current demonstration project under development for the Des Moines Energy Center (DMEC-1), Pleasant Hills, Iowa.

A series of tasks were accomplished to develop the combustion turbine system design for the PFBC application chosen. These tasks included

- Turbine and plant performance
- Blading and cooling arrangements
- Casing arrangement with an off-engine startup combustor
- Turbine system analysis (including controls, fuel handling, and inlet and exhaust systems)
- Off-engine startup combustor
- Interconnecting piping

Figure 1 shows the detailed task and subtasks performed under this study. The sections of this report correspond to these tasks. Each section includes an introduction or overview of the task; review of the analysis performed; and any results, conclusions, or recommended design approaches. Section 7 highlights various conclusions and major design issues, and summarizes the overall activity. Appendixes A and B provide details of some of the project design data.

The Westinghouse W251B12 Combustion Turbine, shown in figure 2, is used as the baseline engine for the PFBC application. The standard W251B12 combustion turbine performance includes

- Power Output - 48 MW (nominal)
- Pressure Ratio - 15.5:1
- Total Air Flow - 382 pounds per second
- Heat Rate - 10,600 Btu/kWh (LHV)
- Thermal Efficiency - 32.2 percent.

Note:

ISO conditions
Natural gas
No injection
4" H₂O inlet and exhaust loss

TASK #12
Generic Turbine Design Studies

Figure 1. Work Breakdown Structure

Figure 2. Westinghouse W251B12 Combustion Turbine

The W251B12 combustion turbine is a single shaft, two-bearing, solid coupling, simple cycle unit containing:

- Multi-stage (19 stages) axial-flow air compressor featuring
 - Variable inlet guide vanes
 - Horizontally split casing giving access to internal parts
 - Individually removable stainless steel blading
 - Accessible pressure-lubricated, pivoted-pad journal bearing
 - Double-acting Kingsbury-type thrust bearing
 - Cold-end drive with solid coupling to main reduction gear
- Combustion system including
 - Eight can-type combustors in a circular array
 - Combustors removable with cylinder cover in place
 - Optional multiple fuels capability
 - Ignition system with retractable ignitors

- Three-stage reaction-type turbine featuring
 - Horizontally split casing giving access to internal parts
 - Individually removable alloy turbine blades
 - Cooled by air-to-air cooler with cooling air filtered
 - Individual first-stage vanes removable with cylinder cover in place
 - Accessible pressure-lubricated, pivoted-pad journal bearing
 - Low loss axial-exhaust system ideal for waste heat applications

Several modifications are required to this engine to enable it to perform as a prime mover in a PFBC system. These modifications will be identified throughout this report.

The W251B12 combustion turbine is typically sold as a packaged unit called the ECONOPAC. The W251B12 ECONOPAC, shown in figure 3, is a complete pre-engineered packaged generating system, designed for ease of operation and maintenance. All components and subsystems are carefully selected and optimized to form a compact, housed within enclosures, designed to comply with environmental requirements as well as showing Westinghouse's increasing concern for aesthetics. The basic ECONOPAC system includes

- W251B12 Combustion Turbine on bedplate assembly
- Generator
- Exciter
- Auxiliaries
 - Starting package
 - Inlet air system including filter (2-stage pad)
 - Exhaust system
 - Electrical/control package
 - Compressor water wash system
 - Fuel-gas/steam injection skid
 - Cooler assemblies
 - Generator switchgear
 - Medium voltage motor starter
 - Fire protection

The ECONOPAC package will need to be modified slightly for these applications. Along with the W251B12 combustion turbine modifications, the inlet air system, the exhaust system, and the fuel gas and steam injection skids must be redesigned. These design changes are described in section 4.

Figure 3. Typical W251B12 ECONOPAC

1. TURBINE AND PLANT PERFORMANCE

1.1 INTRODUCTION

Performance calculations were performed applying a W251B12 combustion turbine, with the first two compressor stages removed, to a first generation PFBC application. Included in this section is a description of the analysis performed and the results of these analyses.

1.2 ANALYSIS

The application of PFBC technology being incorporated by Pyropower at Midwest Power, DMEC-1, was used as the basis for the performance calculations. The approach was to fix those parameters associated with this application and vary the following:

- Inlet temperature
- Pressure drop in PFBC system
- Composition of vitiated air from the PFBC
- Vitiated air temperature into turbine

Variation of these parameters, would give the insight necessary to understand the turbine requirements associated with this application.

1.3 RESULTS

Table 1-1 shows the results associated with a variation in the inlet temperature. Although higher power results from a lower ambient temperature, the analysis indicates that the necessary modifications to the W251B12 limits the lower inlet limit to approximately 59°F.

Table 1-2 gives the results associated with various pressure drops of the PFBC system. As shown, the overall design of the bed, piping, and valves should minimize the pressure drop in the system. This is also true of the combustion turbine in that an objective should be to minimize the pressure drop where the vitiated air enters the casing through its transition to the turbine.

Performance calculations were performed using two different types of coal. Fuel A had products of combustion that were higher in moisture content and contained less carbon dioxide than produced by fuel B. The result, as shown in table 1-3, is that fuel A produced greater power output due to the moisture having higher specific heat than carbon dioxide.

From a net power perspective, as indicated in table 1-4, the system should operate at the highest temperature it can accommodate.

Table 1-1. Effect of Change in Inlet Temperature with 1600°F Burner Outlet Temperature

Ambient Temperature		59°F		80°F		90°F	
Ambient Pressure (psia)		14.3		14.3		14.3	
Relative Humidity (%)		65		65		65	
Inlet Pressure Loss (H ₂ O)		9.0		8.6		8.3	
Exhaust Pressure Loss (H ₂ O)		12.0		11.1		10.7	
Fuel Lower Heating Value (Btu/lb)		2203		2187		2188	
Expected Performance:							
Net Power Output (kW)		20,250		17,870		16,780	
Heat Rate (Btu/kWh) (LHV)		14,000		14,830		15,270	
Exhaust Flow (lb/sec)		265		253		248	
Exhaust Temperature (°F)		801		815		821	
PFBC Pressure Drop (psi)		18.85		18.85		18.85	
Scroll Pressure Drop (psi)		1.84		1.68		1.62	
Station	Pres. (psia)	Temp. (°F)	Flow (lb/sec)	Pres. (psia)	Temp. (°F)	Flow (lb/sec)	Temp. (°F)
Inlet Duct	14.3	59	229	14.3	80	219	90
Compressor Inlet	14.0	59	229	14.0	80	219	90
Compressor Exit	180.8	687	211	173.7	712	202	723
Combustor Inlet	178.6	687	211	171.6	712	202	723
Combustor Exit	158.0	1,600	246	151.1	1,600	235	1,600
Turbine Exit	14.7	801	265	14.7	815	253	821
Exhaust Duct	14.3	801	265	14.3	815	253	821

Table 1-2. Effect of Change in PFBC Pressure Drop with 1600°F Burner Outlet Temperature

PFBC Pressure Drop		1.3 Bars			0.9 Bars			0.5 Bars		
Ambient Pressure (psia)		14.3			14.3			14.3		
Ambient Temperature (°F)		59			59			59		
Relative Humidity (%)		65			65			65		
Inlet Pressure Loss (H ₂ O)		9.0			9.0			9.0		
Exhaust Pressure Loss (H ₂ O)		12.0			10.8			9.3		
Fuel Lower Heating Value (Btu/lb)		2203			2,217			2230		
Expected Performance:										
Net Power Output (kW)		20,250			20,700			21,240		
Heat Rate (Btu/kWh) (LHV)		14,000			13,770			13,520		
Exhaust Flow (lb/sec)		265			265			265		
Exhaust Temperature (°F)		801			801			800		
PFBC Pressure Drop (psi)		18.85			13.05			7.25		
Scroll Pressure Drop (psi)		1.84			1.74			1.67		
Station	Pres.	Temp.	Flow	Pres.	Temp.	Flow	Temp.	Pres.	Temp.	Flow
Description	(psia)	(°F)	(lb/sec)	(psia)	(°F)	(lb/sec)	(°F)	(psia)	(°F)	(lb/sec)
Inlet Duct	14.3	59	229	14.3	59	229	59	14.3	59	229
Compressor Inlet	14.0	59	229	14.0	59	229	59	14.0	59	229
Compressor Exit	180.8	687	211	174.9	678	211	678	169.2	670	211
Combustor Inlet	178.6	687	211	172.7	678	211	678	166.9	670	211
Combustor Exit	158.0	1,600	246	157.9	1,600	246	1,600	158.0	1,600	246
Turbine Exit	14.7	801	265	14.7	801	265	801	14.7	800	265
Exhaust Duct	14.3	801	265	14.3	801	265	801	14.3	80	265

**Table 1-3. Effect of Change in Fuel Composition 1600°F Burner
Outlet Temperature**

	Fuel A			Fuel B		
Ambient Pressure (psia)	14.3			14.3		
Inlet Temperature (°F)	59			59		
Relative Humidity (%)	65			65		
Inlet Pressure Loss (H ₂ O)	9.0			9.0		
Exhaust Pressure Loss (H ₂ O)	12.0			11.5		
Fuel Lower Heating Value (Btu/lb)	2,203			2,427		
Expected Performance:						
Net Power Output (kW)	20,250			18,300		
Heat Rate (Btu/kWh) (LHV)	14,000			14,760		
Exhaust Flow (lb/sec)	265			260		
Exhaust Temperature (°F)	801			807		
PFBC Pressure Drop (psi)	18.85			18.85		
Scroll Pressure Drop (psi)	1.84			1.70		
Station	Pres.	Temp.	Flow	Pres.	Temp.	Flow
Description	(psia)	(°F)	(lb/sec)	(psia)	(°F)	(lb/sec)
Inlet Duct	14.3	59	229	14.3	59	229
Compressor Inlet	14.0	59	229	14.0	59	229
Compressor Exit	180.8	687	211	178.8	681	211
Combustor Inlet	178.6	687	211	174.6	681	211
Combustor Exit	158.0	1,600	246	154.0	1,600	242
Turbine Exit	14.7	801	265	14.7	807	260
Exhaust Duct	14.3	801	265	14.3	807	260

Table 1-4. Effect of Change in Burner Outlet Temperature

	1,600°F			1,950°F		
Ambient Pressure (psia)	14.3			14.3		
Inlet Temperature (°F)	59			59		
Relative Humidity (%)	65			65		
Inlet Pressure Loss (H ₂ O)	9.0			9.0		
Exhaust Pressure Loss (H ₂ O)	12.0			12.0		
Fuel Lower Heating Value (Btu/lb)	2,203			2,987		
Expected Performance:						
Net Power Output (kW)	20,250			27,980		
Heat Rate (Btu/kWh) (LHV)	14,000			13,150		
Exhaust Flow (lb/sec)	265			264		
Exhaust Temperature (°F)	801			1,000		
PFBC Pressure Drop (psi)	18.85			18.85		
Scroll Pressure Drop (psi)	1.84			1.81		
Station	Pres.	Temp.	Flow	Pres.	Temp.	Flow
Description	(psia)	(°F)	(lb/sec)	(psia)	(°F)	(lb/sec)
Inlet Duct	14.3	59	229	14.3	59	229
Compressor Inlet	14.0	59	229	14.0	59	229
Compressor Exit	180.8	687	211	180.9	687	202
Combustor Inlet	178.6	687	211	178.7	687	202
Combustor Exit	158.0	1,600	246	158.0	1,950	236
Turbine Exit	14.7	801	265	14.7	1,000	264
Exhaust Duct	14.3	801	265	14.3	1,000	264

2. BLADING AND COOLING ARRANGEMENTS

2.1 BLADING ARRANGEMENT ANALYSIS

The reference plant PFBC requires a steady-state flow at the design point of 211 lb/sec of clean compressed air in the range of 125 to 180 psig. The bed will in turn produce 252 lb/sec of hot gas that will pass through a ceramic filter and be piped into the combustion turbine casing. The standard configuration for the W251B12 combustion turbine has a compressor discharge mass flow on the order of 375 lb/sec at a pressure of 201 psia. To adapt the W251B12 compressor for this reduced flow, the turbine and the compressor must be modified. The following sections discuss the changes necessary to achieve the modified performance required for installation in the baseline plant.

2.1.1 Turbine Vane / Blade Analysis

2.1.1.1 Introduction

The compressor and turbine are matched when the mass flow and the compressor discharge pressure allow the turbine and the compressor to operate at or near their maximum efficiencies. To match the turbine and compressor at the reduced mass flow that the PFBC requires, we had initially planned to restrict the throat area of the first-stage turbine vane by turning the airfoils on the shrouds by several degrees based on a scaled compressor discharge pressure. After we calculated the compressor map for the modified compressor and used it as input to the performance program, we determined that the compressor discharge pressure was lower than our first estimate. To increase the compressor discharge pressure to an acceptable value, we restricted the throat area of the 1st-stage turbine vane by approximately 1 degree.

2.1.1.2 Analysis

As a first approximation, we attempted to match the compressible flow function, S , of the reduced flow PFBC turbine to that of the baseline W251B12. The compressible flow function, which is a measure of the "swallowing capacity" of the turbine, is expressed in simplified form as

$$S = \frac{G \sqrt{T}}{P},$$

where G = mass flow (lb/sec)
 T = absolute temperature, °F
 P = pressure, psia

2.1.1.3 Results

The compressible flow function at the stage 1 turbine vane inlet for the baseline W251B12 is 81.1. For the reduced-flow PFBC turbine, with no setting angle correction, the compressible flow function is 77.8. This agreement shows that the existing vane setting angle is about correct for the reduced flow, temperature, and pressure. Final adjustment of the setting angle is determined by the arrangement that gives the lowest incidence angles at the design point and acceptable values at turndown conditions. Analysis of the velocity triangles in the turbine, however, shows some departure from the baseline case. Incidence angles for the reduced flow PFBC turbine with 8-percent cooling were acceptable following current Westinghouse design practices.

The match of incidence angles and the turbine efficiency can be somewhat improved by closing the row 1 vane by approximately 1 degree. There is no technical difficulty in closing the vane setting angle, but the additional cost of producing manufacturing drawings, requalifying the casting process and first-piece inspection, and maintaining a spare parts inventory would probably offset the modest improvement in turbine performance.

2.1.2 Compressor Vane/Inlet Guide Vane (IGV) Development

2.1.2.1 Introduction

For the conceptual design of the 17-stage PFBC compressor, we had to modify baseline W251B12 inlet flow path contours. We also repositioned the IGV closer to the new 1st stage (the original W251B12 3rd stage) since two compressor stages were removed from the baseline compressor. These changes resulted in a reduction of compressor inlet mass flow from 375 lb/sec to 229 lb/sec, a reduction in pressure ratio from 15.5 to 11.6, and a revised compressor map.

2.1.2.2 Analysis

The original 19-stage B12 compressor was modified to remove the first two stages of compressor blading. The IGVs needed to be moved closer to the new 1st stage compressor blades to maintain their effectiveness. Several options were analyzed to determine whether the IGVs would fit into the location of the 2nd-stage stator vane and if the externally located operating linkage could be relocated. Once the location of the IGVs was determined, tip and hub contours were changed to improve the inlet flow path.

2.1.2.3 Results

Figure 2-1 shows the new 17-stage compressor inlet and inlet guide vane configuration. The changes to the compressor section to convert it to a 17-stage compressor includes

- Compressor flow path contour modification featuring
 - Reshaped inlet inner scroll
 - W251B10 inlet outer scroll
 - Reshaped IGV tip and hub surfaces
- IGV alterations
 - Modified casing for new IGV position
 - Modified blade design (profile and shaft)
 - Reversed IGV actuating linkage

To locate the IGV at the 2nd-stage stator vane location in the baseline W251B12 compressor, the conical tip and hub surface contours had to be made cylindrical. This change in contours will enable the IGVs to operate normally as they rotate from closed to open as they control flow through the compressor during startup. To provide this cylindrical surface for the IGV, we combined the B12 and B10 tip and hub contours. A layout of the 17-stage compressor contours was developed. The B10 contour provides a cylindrical surface for the IGV at the tip but is slightly tapered at the hub. To provide cylindrical surfaces for the IGV at both the tip and the hub, we removed the slight taper at the hub and blended the cylindrical and tapered section of the hub with a radius. We used the modified B10 contour for the front of the compressor and the B12 contours for the unmodified portion of the B12 compressor.

The compressor analysis program uses the straight lines between the tip and the hub contour calculating stations. We improved the location of the B10 calculating stations at the inlet of the compressor by locating the calculating stations more uniformly. The revised location of the calculating stations improved the convergence of the compressor analysis. The final flow path modifications include a revised inlet inner scroll shape, the B10 inlet outer scroll, and a revised IGV blade area.

Since the IGV was relocated, we needed to modify the IGV actuating linkage and the compressor casing for the new location (see figure 2-1). The IGV linkage was originally in a position that did not interfere with the compressor casing support. We needed to modify the casing support by reducing its length from 15.67 inches to 14 inches. This still should provide enough space for adequate support of the compressor casing. We also shifted the location of the center point of the IGV shaft 21 inches toward the compressor blades (with the direction of flow). The IGV operating linkage was also reversed from the B12 design so that the linkage mechanisms are inward from the compressor inlet. This modification should enable us to use most of the same linkage parts for the new 17-stage compressor design.

During the detailed design the new IGV will need to be designed. Once this design is developed the casing support, casing IGV penetrations, and the linkage designs can be completed.

Figure 2-1. 17-Stage Compressor Inlet Details

2.1.3 Compressor Aerodynamic Analysis

2.1.3.1 Introduction

This section describes the analysis performed to verify the performance of the W251B12 compressor after it was modified by removing the first two stages and relocating the IGV. An analytical compressor map in the vicinity of the design point was produced to support plant performance calculations.

2.1.3.2 Analysis

The aerodynamic analysis was performed using Westinghouse proprietary software that analyzes the performance of multistage axial-flow compressors.

Using the 17-stage compressor layout described in section 2.1.2, the compressor geometries were entered into the compressor analysis computer program. The analysis was performed at three constant speeds of 5,200, 5,400 and 5,600 rpm to generate the compressor map near the design point. At these speeds, the program converged at pressure ratios below 11.7. Program convergence was improved using the calculating stations of the modified inlet contour as described in section 2.1.2. The (Westinghouse proprietary) thermal cycle analysis program uses the compressor map generated to determine turbine performance.

The aerodynamic analysis approach was to calculate the diffusion factors and the incidence and deviation angles to determine if the blade and stator vane aerodynamic loadings were acceptable. Current Westinghouse design practice is to keep the diffusion factors below 0.5 and to keep the incidence and deviation angles low to prevent flow separation and blade stall.

2.1.3.3 Results

The results of the aerodynamic analysis of the 17-stage PFBC compressor showed that the compressor map and the aerodynamic loading on the blades and vanes were acceptable.

Performance of the standard 19-stage and the modified 17-stage W251B12 compressor is summarized in table 2-1.

The compressor inlet flow was reduced from 375 lb/sec to 229 lb/sec, and the pressure ratio was decreased from 15.5 to 11.6. The overall thermal efficiency is nearly the same for both compressors. The maximum ambient operating temperature is 120°F. The minimum inlet operating temperature for the 17-stage compressor is limited to 55°F, since the compressor aerodynamic loadings on the rear stages are higher than on the baseline 19-stage W251B12 compressor. If the compressor were to be operated at a compressor inlet temperature lower than 55°F, the compressor might stall or surge.

Table 2-1. Performance of 19- and 17-stage W251B12 Compressors

	19 STAGE	17 STAGE
Inlet Flow, lb/sec	375	229
Speed, rpm	5400	5400
Pressure Ratio	15.5	11.6
Overall Efficiency	0.87	0.86
Maximum Ambient Temperature, °F	120	120
Minimum Ambient Temperature, °F	-40	55

The turbine performance was calculated using a Westinghouse proprietary thermal cycle analysis program. Turbine performance for 59°F, 80°F, and 90°F ambient inlet temperatures is summarized in section 1.

2.2 COOLING SYSTEM ANALYSIS

2.2.1 Turbine/Rotor Cooling Analysis

We reviewed the existing turbine and rotor cooling requirements for the modified engine. The blade path temperatures in the PFBC are lower than the metal temperatures in the baseline W251B12, indicating that little or no cooling will be required for the airfoils. The existing turbine disk material, however, has a maximum operating temperature limit of 800°F, and the disk runs at approximately 650°F in the baseline W251B12 engine. To maintain a disk temperature of 650°F, cooling of the disk rim and cavity is required. Without substantially modifying the current cooling configuration, approximately 8 percent of the compressor flow will be required to cool the disks adequately. Redesign of the cooling system during the detailed design phase could reduce the cooling flow requirement further.

2.2.2 Scroll/Center Section Flow Analysis

2.2.2.1 Introduction

Because of PFBC cycle requirements, we had to analyze the toroidal manifold/casing cooling requirements to develop a cooling scheme.

2.2.2.2 Analysis

The toroidal manifold is supplied with 1600°F air from the PFBC. The strength of the metal alloys used in high-temperature applications is strongly temperature dependent, so the average wall temperatures of the manifold should be kept as low as possible. The outer diameter of the manifold is also close to the inner diameter of the casing, and cooling must be provided to keep the casing temperatures at acceptable levels. Compressor discharge air, finally, must be drawn from the casing to provide rotor cooling air. This air passes through an off-engine cooler and returns through pipes that penetrate the casing and supply cooled air to the turbine rotor through the air separator.

2.2.2.3 Results

The objectives described above are addressed by the casing/manifold cooling scheme shown in figure 2-2. Approximately 2 percent of the compressor inlet flow is ducted from the 11th-stage bleed port (13th stage in the baseline engine) to cool the 3rd-stage turbine disk rim. No air is drawn from the 4th-stage bleed (6th stage in the baseline engine) because the pressure is too low to use for cooling. Another 6 percent of the compressor inlet flow passes through metering holes in the combustor casing baffle/manifold support. Two-thirds of this air (4 percent of compressor flow) is ducted between the outer diameter of the manifold and the inner diameter of the casing, cooling both the manifold and the casing by convection. The other third of the cooling air (2 percent of the compressor flow) passes between the compressor diffuser and the inner diameter of the manifold. We estimate that the average wall temperature in the manifold will be approximately 1450°F. The inner surface of the casing is heated by radiation from the manifold and cooled by convection as described above. The outer surface of the casing is covered by an insulation blanket to protect personnel. Heat is conducted through the insulation blanket and transferred to the ambient air by convection. We estimate the average metal temperature of the casing to be 800°F. This estimate does not include any radiation shielding or internal insulation near the manifold, either of which could be used if necessary to reduce the casing temperature.

Figure 2-2. Casing Manifold Cooling Scheme.

3. CASING ARRANGEMENTS FOR INTERFACE WITH OFF-ENGINE COMBUSTOR

3.1 CONCENTRIC INLET AND OUTLET ANALYSIS

3.1.1 Introduction

An initial design configuration for a W251B12 combustion turbine using concentric inlet/outlet piping local to the combustion turbine was provided. See figure A-1 in appendix A. This configuration showed the hot (1600°F) return pipe, which runs from the PFBC filters to the turbine inlet, located concentrically within the relatively cool (670°F) outlet pipe. Once clear of the combustion turbine, the compressor outlet exits from the concentric pipe assembly laterally through the branch of a reducing tee. The turbine inlet pipe enters the end of the concentric pipe through a concentric reducing fitting attached to the run of the reducing tee.

3.1.2 Analysis

A preliminary review of the structural aspects of this concentric configuration raised the following technical concerns:

- The ability to reinforce the combustion turbine casing adequately for such a large penetration
- The ability of the 1600°F return pipe (inside the compressor discharge pipe) and its distribution plenum (inside the combustion turbine casing) to withstand a 20- to 25-psi external pressure differential; that is, the pressure drop between the compressor outlet pressure and the turbine return pressure.

The above concerns led to the design of a more conventional piping configuration with separate inlet and outlet pipes.

3.1.3 Results

A review of the allowable stresses at 1600°F and the creep data with regard to percent creep for stress levels that could be achieved with reasonable thicknesses for these configurations at these temperatures led us to conclude that buckling would occur with fewer than 1,000 hours of operation.

3.2 SEPARATE INLET AND OUTLET ANALYSIS

3.2.1 Arrangement

3.2.1.1 Introduction

The development of the conceptual design of the separate combustion turbine inlet and outlets was an iterative process. Several design decisions were made in the process of selecting the conceptual design presented in the next section. The design process and the alternatives considered are discussed in appendix A.

3.2.1.2 Results

The conceptual design of the turbine casing is shown on figure 3-1. The length of the casing and the flanges on each end are identical to those of the W251B12. The casing diameter and the wall thickness have been increased to accommodate a toroidal manifold up to 36 inches in diameter. A 36-inch inside diameter (id) round opening in the casing accommodates the 32-inch-diameter inlet pipe liner and cooling air for the liner and pipe walls.

Both the inlet and the outlet pipes connect to openings in the lower right-hand side (looking with flow) of the casing. These openings were located so that they would minimize interference with maintenance and keep the existing turbine bedplate design. The openings are reinforced in accordance with the area replacement method of the ASME code, as discussed in the next section. Beyond the inlet flange, the 1600°F air from the PFBC is ducted through a special lined and insulated pipe. The inside diameter of the outlet pipe is 24-inches. Because it carries compressed air at 670°F, the pipe requires no liner or special cooling, but it must be shielded to protect personnel. Anchored expansion joints between the casing and the main piping runs protect the casing from external piping loads.

The configuration inside the casing has been modified as follows:

- Internal combustors have been relocated to the external startup combustor
- A toroidal manifold has been incorporated to separate the cooler (670°F) compressor air from the hot (1600°F) air returning from the PFBC.

The inlet sleeve and toroidal manifold should be designed to withstand a steady external pressure. A small pressure drop across the manifold support diaphragm will reduce the pressure drop across the manifold itself. The feature that limits the life of this design is the creep bending strength of the manifold at elevated temperatures. To improve the manifold life, 670°F compressor discharge air is routed between the manifold and the casing. This air cools both the external surface of the manifold and the inner surface of the casing, and is removed from the forward end of the casing through a set of bleed pipes. This air is cooled externally and returned to cool the rotor. Joints between the inlet pipe, manifold, and transition ducts are fitted to allow some relative motion, and are expected to leak. This

Figure 3-1. Turbine Casing

leakage is beneficial to the joint, since relatively cool air leaks into the hot gas path, but the leakage must be minimized to avoid performance degradation.

The casing should be maintained at the temperature of the W251B12 baseline casing, approximately 800°F. The casing will be covered by an external insulation blanket to protect personnel, so the cooling flow must be adequate to remove most of the radiation heating of the casing. A radiation shield may be required pending further analysis to be done during the detailed design phase.

Because the manifold will be designed conservatively against creep bending failure, there is a substantial margin against higher transient pressure drops resulting from upset conditions. For transient pressure drops, the limiting condition is the elastic buckling strength of the manifold. A detailed analysis of the cooling system, creep bending analysis, and elastic stability analysis are beyond the scope of the task, but our preliminary assessment is that such a design is feasible.

A structural diaphragm (figure 3-1) is shown within the casing to support the toroidal manifold and the transition ducts. It also separates the toroidal manifold region from the compressor outlet region. This diaphragm is used to limit the bypass cooling flow and reduce the differential pressure across the toroidal manifold wall to protect the toroidal manifold. A comprehensive finite element stress analysis of the manifold and its support structure will be required before the design is completed.

3.2.2 Flow Analysis

3.2.2.1 Introduction

We used traditional handbook methods to conservatively estimate pressure losses from the compressor discharge diffuser to the discharge pipe and from the return pipe to the transition duct exit plane. Most of our calculations were based on methods presented in *The Handbook of Hydraulic Resistance*, 2nd edition, by I. E. Idelchik. Since traditional methods can't accurately predict the distribution of airflow into the transition ducts from the combustor manifold, we built a finite element model of the manifold system to use in a computational fluid dynamics (CFD) analysis of that system.

3.2.2.2 Analysis

Almost all the pressure loss between the compressor discharge and the discharge pipe is from two sources: the expansion/turning loss from the compressor diffuser and the pipe entrance loss. Additional losses are from flow swirling around in the chamber and from air flowing between the transition ducts, but, those losses are small enough to be ignored.

If the air leaving the compressor diffuser were simply dumped into a large cavity, the pressure loss would be approximately equal to the velocity head at the diffuser exit plane. In the PFBC unit, however, the combination of the shaft on one side and the confluence of the transition ducts downstream of the diffuser will force some (if not most) of the flow to turn

back on itself. Although the flow probably will not turn a full 180 degrees and the velocity will be a little lower because of some diffusion, we assumed complete turning to calculate the 1.96-psi (two velocity heads) pressure loss. This is conservative, because the actual loss will be somewhere between one and two velocity heads.

To calculate the pipe entrance loss, we assumed that the flow enters a pipe that has a baffle facing it. We further assumed that all the flow entering the pipe travels between the baffle and the chamber wall, so no flow enters the pipe through gaps between the transition ducts just upstream of the pipe. This conservative approach gives a 0.98-psi pipe entrance loss.

If the air leaving the compressor flowed uniformly through the spaces between the transition ducts, the pressure loss would be about 0.05 psi. We ignored this loss because it was small compared to the probable overestimation of the other losses.

The combustor plenum system comprises the return pipe segment, the combustor plenum, and eight transition ducts. We broke down the pressure losses in the system into three categories: return pipe exit loss, transition duct entrance loss, and transition duct internal loss. For the first category we calculated a loss of 0.57 psi for airflow entering a tee, slightly higher than it would be for a simple sudden expansion loss. To calculate the transition duct entrance loss we assumed that air from a passing airstream enters tubes mounted flush with a wall, and we progressively lowered the airstream velocity as air was removed by each transition duct. The pressure loss ranges from 0.36 psi for the transition duct nearest the return pipe to 0.39 for the farthest.

To calculate the transition duct loss we assumed the duct could be represented by an S-shaped gooseneck bend with 35-degree angles at each end. The calculation procedure detailed in Idelchik is straightforward and includes friction losses. This pressure loss is about 0.43 psi.

We built a finite element model of the toroidal manifold and the transition ducts (figure 3-2) for a CFD analysis of that system. To save time we simplified some features of the manifold system in our model that should not affect the accuracy of the results significantly. The model return pipe is not offset 5 degrees with respect to one of the transition ducts as it is in the hardware, for example. The maximum airflow through any transition duct is limited to the choked flow of the turbine stator vanes downstream of the duct, so we added a contraction downstream of each transition duct (figure 3-3) in our model that limits the flow by choking it at the appropriate flow rate.

3.2.2.3 Results

The total pressure loss from the compressor diffuser exit to the pipe leading to the fluidized-bed combustor is approximately 2.9 psi, which compares with a 2.2-psi loss to the combustor inlet for the unmodified gas turbine without the fluidized-bed combustor. This loss could be reduced by enlarging the chamber discharge pipe, increasing flow diffusion from the compressor, or reorienting the diffuser.

Figure 3-2. Combustor Plenum Model with Transition Ducts

Figure 3-3. Segment of CFD Model Showing Key Features

We calculated 1.4-psi manifold system pressure loss. If the manifold and transition ducts must remain unchanged, the loss could be reduced by using multiple return pipes. Otherwise, a larger manifold and combination manifold/transition duct could be used.

3.2.3 Stress Analysis

3.2.3.1 Introduction

The concerns developed as we evaluated the concentric/eccentric penetrations led us to design a more conventional piping configuration with separate inlet and outlet pipes. The separate pipe casing penetrations were located to minimize interference with existing gas turbine piping and support structures.

3.2.3.2 Analysis

Once the location and size of the separate casing penetrations were established, we performed an area reinforcement analysis in accordance with the ASME Boiler and Pressure Vessel Code, Section VIII, Division 1.

3.2.3.3 Results

A W251B12 gas turbine casing incorporating a 42-inch od/36-inch id inlet penetration and a 28-inch od/24-inch id outlet penetration in the same quadrant represents a feasible design. From a structural standpoint, the best way to achieve this would be to make a forging that includes both pipe nozzle segments out to their first pipe flange, incorporating the requisite reinforcement. This forging would then be inset into the casing using a 100-percent efficient butt weld.

The initial analysis was reevaluated in order to minimize the distance between the centerlines of the two pipe nozzles. Application of the ASME Section VIII reinforcement rules, together with the listed modifications in assumptions, was found to permit a minimum separation distance of 7.25 inches between nozzle outer surfaces and a minimum centerline separation of 42.25 inches while still satisfying the Section VIII rules.

4. TURBINE SYSTEMS

4.1 CONTROL SYSTEM ANALYSIS

4.1.1 Introduction

The PFBC combustion turbine with an off-engine startup combustor will require the standard W251B12 control system to be modified to include control and sequencing of the external valves and startup combustor. Under this task we looked at the current Westinghouse Distributed Processing Family (WDPF) control system used to control the standard W251B12 combustion turbine generator (CTG) to identify all necessary control system input and output signals and determine the adequacy of the system for this application.

4.1.2 Analysis

As part of our analysis we

- Reviewed a set of 46 drawings of the existing WDPF control system for the W251B12 CTG. From these drawings we identified all necessary control system input and output signals.
- Reviewed interface requirements for the off-engine combustion turbine startup combustor and identified all the control system input and output signals this interface will require.
- Analyzed the adequacy of the current WDPF control system and determined whether it has the capacity to meet the additional requirements imposed by the baseline plant PFBC interface.

To determine the adequacy of the WDPF for the PFBC application, we developed some basic operating startup and shutdown sequences and reviewed the normal operation and emergency shutdown procedures.

4.1.3 Results

From the WDPF drawings of the W251B12 CTG, we prepared Lotus spreadsheets listing all the W251B12 combustion turbine engine control signals required to work with the WDPF control system. An analysis of cold startup, shutdown and hot startup, and normal operation was performed and sequences developed. Proprietary data were used, and results are not presented.

Figure 4-1 shows the sample flow panel that was conceptually designed for the entire PFBC compressed air and hot gas flow piping layout including the PFBC startup combustor. Figure 4-2 shows the conceptual design of the CT off-engine startup combustor control panel.

Figure 4-2. CT Startup Combustor Control Panel

4.2 STARTUP FUEL SYSTEM

4.2.1 Introduction

The combustion turbine startup combustor must operate on either fuel oil or natural gas. In addition, emission requirements may require steam injection with the fuel. Westinghouse has provided skid-mounted systems for each of these fuels and supplies them for conventional combustion turbine applications. Modified versions of these fuel systems will be required for the PFBC plant.

4.2.2 Analysis

The Fuel Gas and Steam Injection Skid contains the valves, strainers, vents, and controls necessary to control and regulate the fuel gas and steam, which is injected into the turbine startup combustor. A review of the skid components indicates that the design is adequate for use with the turbine startup combustor. The only potential changes are reduced trim in the control valves to match the lower fuel and steam flow rates required by the PFBC turbine. Valve trim requirements will be defined during detail design when fuel flow rates are finalized.

The Fuel Oil Injection Skid is similar to the Fuel Gas and Steam Injection Skid except it contains the components and controls for handling the fuel oil and may have to have its control valves and fuel pump resized for the lower fuel flow rates of the PFBC turbine.

The turbine startup combustor assembly will contain the fuel manifolds and control valves normally associated with the turbine assembly of a conventional combustion turbine. Since the number of combustors will be less than on a standard turbine, the number of fuel nozzles will be similarly reduced. Also, the need for the accurate subdivision of the fuel oil to the several nozzles will not be required and the flow divider system and all of its components will be replaced by a simple fuel oil manifold, similar in function to the fuel gas manifold. This simplification is possible because the combustor outlet gasses will be thoroughly mixed prior to being introduced into the turbine. Any minor variation in fuel flow to the combustors (and resultant variation in combustor outlet temperature) will disappear well before the gas gets to the turbine.

4.2.3 Results

Detailed specifications for fuel system components will be developed during detail design, when fuel flow requirements have been finalized.

4.3 INLET AND EXHAUST SYSTEM

4.3.1 Introduction

The inlet and exhaust system for this W251B12 application will be slightly different from the typical ECONOPAC offering, which is shown in Figures 4-3 and 4-4. The inlet system of a typical ECONOPAC is comprised of a two-stage inlet air filter, an absorptive panel silencer, ducting and expansion joints. This inlet system would be standard for a PFBC application that uses the normal 19-stage compressor. This PFBC application, however, requires a modified 17-stage compressor instead of the standard 19-stages. This modification reduces the flexibility to operate at a wide range of the inlet air temperatures and therefore requires temperature control. To achieve this controlled ambient temperature operating range, a combination heating and cooling system must be installed in the intake system. The necessary heating and cooling coils are shown in Figure 6-2 and are located between the inlet filter and compressor casing.

The typical ECONOPAC exhaust system is comprised of an exhaust silencer, expansion joints, ducting, and exhaust stack. The exhaust system for a PFBC application will contain a heat recovery steam generator. Both the heat recovery steam generator and the inlet and exhaust components are beyond the scope of this study.

4.3.2 Results

On the basis of the design detailed by the Pyropower reference plant, the inlet and exhaust systems will have the following design criteria:

Inlet System:

- Ambient Air
 - Temperature Range -20 to 120°F
 - Pressure 14.3 psia
- Inlet Pressure Drop 3 in. H₂O
- Inlet to Compressor
 - Temperature Range 55°F minimum
(17-stage compressor design limit)

Exhaust System

- Turbine Outlet Exhaust Gas
 - Temperature 801° to 821°F
 - Pressure 14.3 psi
- Exhaust Pressure Drop 8.0 in. H₂O

One of the other design considerations for the reference plant is the location of the inlet duct as it enters the turbine casing. Typical ECONOPAC installations have the inlet installed horizontally; but, because of inlet air temperature control equipment location constraints on this PFBC application, the reference design of the combustion turbine includes a 45-degree installation of the inlet system. This will provide the necessary clearance to install the other plant equipment. See figure 6-3 for an elevation view of the plant arrangement.

Figure 4-3. Typical W251B12 ECONOPAC General Arrangement

Figure 4-4. Typical W251B12 ECONOPAC Section Views

5. STARTUP COMBUSTOR

5.1 INTRODUCTION

To reduce the internal pressure losses of the hot gas inlet section of the combustion turbine, an off-engine startup combustor was proposed as part of the PFBC reference design. The off-engine combustor is installed in the system to provide enough combustion to heat the compressed air from the compressor during initial startup of the combustion turbine. An additional design requirement is that of operating at reduced flow, since once the combustion turbine has warmed up some of the compressed air will be routed to the PFBC bed for bed startup. Other design criteria such as outside wall temperature, pressure vessel design, and minimized pressure losses were considered as part of the conceptual design. The following sections detail the conceptual design of the off-engine startup combustor.

5.2 ANALYSIS

The conceptual off-engine startup combustor is located in a crossover connection, shown in figures 6-2 and 6-3 in section 6. The inlet and outlet flanges and pipes were sized to match the crossover pipe dimensions. Heat balance was calculated to ensure that adequate energy is added to the system through the use of an off-engine startup combustor. Through this analysis we identified the number of combustors required and the exhaust products. The environmental requirements that the startup combustor will be required to meet have not been established. The standard W251B12 combustor baskets used in the conceptual design may be replaced with dry low-NO_x combustors or steam injection as required by emerging needs.

5.3 RESULTS

The final conceptual design for the off-engine combustor is shown in figure 5-1. Table 5-1 details the associated bill of materials for the off-engine combustor. The combustor was designed to have flow characteristics similar to those in the W251B12 engine. Compressed air enters through a single axial pipe and circulates into the combustor area. This compressed air circulation helps to cool the pressure vessel walls before it enters one of the five combustor basket assemblies. Once it has entered the combustor basket, the compressed air is mixed with either a natural gas or a liquid fuel and combustion occurs as in a normal combustion turbine. The combustion gases are then routed through straight transition tubes into the clean hot gas pipe inlet to the combustion turbine.

Once the combustion turbine has warmed up, some of the compressed air is bypassed to the PFBC. The startup combustor will continue to fire until the PFBC bed is on line and returning 1200°F combustion air to the combustion turbine. When this occurs, the startup combustor will be shut down.

Table 5-1. Bill of Materials for Conceptual Off-Engine Startup Combustor

Item No.	Description	Qty Req'd.
1	Welded cone	1
2	Torospherical head	1
3	Inlet diffuser tube	1
4	Inlet flow baffle/outlet transition assembly	1
5	48-inch weldneck flange	1
6	30-inch weldneck flange	1
7	30-inch penetration ring for sphere	1
8	20-inch penetration fittings for combustors	5
9	6-inch penetration fittings for igniters	5
10	Combustor assemblies, including ignition system	5
11	Long, straight transition assemblies	5
12	View port assemblies (if required)	5
13	Front centerline support (rigid)	1
14	Rear centerline support (flexplate)	1

6. INTERCONNECTING PIPING SYSTEM

6.1 INTRODUCTION

This task involves the design of the piping out of and into the combustion turbine. It includes the piping, valves, and expansion joints required to transmit the compressed air from the compressor to the PFBC system and to return the heated gas from the PFBC to the turbine. It also includes the crossover pipe and valve to the turbine startup combustor.

6.2 PIPING DESIGN

The design of a piping system required a definition of the plant cycle arrangement and the fluid state points of that system. The state point diagram is in figure 6-1.

Preliminary design condition state points were obtained from Pyropower by telephone and were used as the basis for the initial piping design. As compressor and turbine performance analysis results became available, these data were updated and are presented in table 6-1. State points for the several operating modes during startup are shown in tables 6-2 through 6-4.

6.2.1 Pressure Drop Analysis

A proprietary Lotus 1-2-3 spreadsheet, generated to analyze heating, ventilating, and air-conditioning (HVAC) duct systems, was modified and used to verify the pipe sizes and calculate the pressure drops of the hot and cold legs of the piping system defined on figures 6-2 and 6-3. The flow rates, temperatures, and pressures of tables 6-1 through 6-4 were used in the analysis.

The spreadsheet was modified to account for the nonstandard density of the gas in the pipes. For HVAC analysis a standard density of 0.075 lb/ft^3 is used. Because of the increased pressure and temperature, the density in the cold leg is approximately 0.4 lb/ft^3 coming out of the compressor and the density in the hot leg is approximately 0.2 lb/ft^3 . These densities were calculated using the basic $Pv = RT$ formula.

where

P = pressure, psia

v = specific volume ft^3/lb .

R = gas constant, 53.5

T = absolute temperature, $^{\circ}\text{R}$

Figure 6-1. State Point Diagram

Table 6-1. PFBC Air System State Points-Base Case Design Flow*

Point	Description	Flow rate lb/sec	Temperature, °F	Pressure, psia
1	Inlet air	229	---	14.3
2	Compressor inlet	229	59	14.2
3	Compressor outlet	211	657	162.3
3a	Bleed flow	18	657	162.3
4	CT startup combustor	0	---	139.5
5	PFBC system inlet	211	657	162.3
5a	Purge bypass	0	657	162.3
6	PFBC system outlet	249	1600	139.5
7	Turbine inlet	249	1600	139.5
8	Turbine outlet	268	836	14.3
9	PFBC system vent	0	---	14.3

* 2/10/93

Table 6-2. PFBC Air System State Points-Turbine Startup Flow*

Point	Description	Flow rate lb/sec	Temperature, °F	Pressure, psia
1	Inlet air	229	---	14.3
2	Compressor inlet	229	59	14.3
3	Compressor outlet	211	657	162.3
3a	Bleed flow	18	657	162.3
4	CT startup combustor	211	1182	139.5
5	PFBC system inlet	0	---	162.3
5a	Purge bypass	0	---	162.3
6	PFBC system outlet	0	---	14.3
7	Turbine inlet	211+	1182	139.5
8	Turbine outlet	229+	TBD	14.3
9	PFBC system vent	0	---	14.3

*2/10/93

Table 6-3. PFBC Air System State Points-PFBC Startup (End of Warmup)*

Point	Description	Flow rate lb/sec	Temperature, °F	Pressure, psia
1	Inlet air	229	---	14.3
2	Compressor inlet	229	59	14.3
3	Compressor outlet	211	657	162.3
3a	Bleed flow	18	657	162.3
4	CT startup combustor	158	1633	139.5
5	PFBC system inlet	53	657	162.3
5a	Purge bypass	0	---	162.3
6	PFBC system outlet	0	---	139.5
7	Turbine inlet	211+	1406	139.5
8	Turbine outlet	229+	TBD	14.3
9	PFBC system vent	53	1200	14.3

*2/10/93

Table 6-4. PFBC Air System State Points-PFBC Purge*

Point	Description	Flow rate lb/sec	Temperature, °F	Pressure, psia
1	Inlet air	229	---	14.3
2	Compressor inlet	229	59	14.3
3	Compressor outlet	211	657	162.3
3a	Bleed flow	18	657	162.3
4	CT startup combustor	179	1406	139.5
5	PFBC system inlet	0	---	162.3
5a	Purge bypass	32	657	162.3
6	PFBC system outlet	0	---	14.3
7	Turbine inlet	211+	1406	139.5
8	Turbine outlet	229+	TBD	14.3
9	PFBC system vent	32	Amb+	14.3

*2/10/93

Figure 6-2. Plant Arrangement - Plan View

Figure 6-3. Plant Arrangement - Elevation View

The revised density was used first to convert the mass flow rate, lb/sec, to volumetric flow rate, cfm. It was also used to correct the ΔP calculation by multiplying the conversion factor by the square root of the density ratio. The revised conversion factor is given on the printout in appendix B. Printouts of the two cases are shown in appendix B, labeled table B-1 and table B-2 for the cold pipe and hot pipe respectively.

Referring to table B-1, the top of the first sheet lists the parameters used in the analysis and summarizes the results. A velocity limit of 12,000 fpm was assumed for this application and this limit is not exceeded in the current design except at the turbine inlet. The table starting in the middle of the first page is a component-by-component list of the piping model generated and the calculated pressure drop across each component. In the far right column is the static pressure in pounds per square inch, absolute.

Table B-2 is similar for the hot pipe. Note that in both tables the analysis extends only to the isolation valve adjacent to the crossover pipe. Piping to and from the PFBC pressure vessel is not included.

6.2.2 Thermal Insulation Design

Several insulation manufacturers were consulted for recommendations on insulation that would meet the requirements of the PFBC hot piping. Prior successful operating experience with refractory linings in the Westinghouse Synthetic Fuel Project, led us to select H-W Lightweight Castable 20 as the reference insulation. This product, produced by Harbison-Walker Refractories, Inc., is designed to be cast in place and anchored to the pipe wall. Its maximum service temperature is 2000°F, with a thermal conductivity of 1.50 Btu/hr-ft²-°F/in at the design temperature of 1600°F. The inner surface of the insulation is separated from the hot gas by a 1/16-inch thick stainless steel liner.

The spreadsheet calculated the heat loss and temperature profile through the insulated wall of the hot pipe. Insulation thicknesses and conductivities, gas and air temperatures, and pipe and liner thicknesses can be varied and the resulting temperature profile and heat transfer rate determined. A listing of the results for the two pipe sizes shown on figure 6-2 and 6-3 are given in Tables B-3 and B-4 in appendix B.

6.2.3 Pipe Thermal Insulation Results

- The pipe surface temperature for the 60-inch pipe with 7.5-inch insulation is approximately 200°F. For the 48-inch pipe with 5.5-inch insulation, the pipe surface temperature is approximately 225°F.
- To reduce the pipe surface temperature to 140°F would require an additional 9 inches of ceramic insulation inside the pipe or 2 inches of other insulation on the outside of the pipe. Adding insulation to the outside diameter can result in overtemperature of the pipe wall (temperature greater than 650°F) and is not recommended.

6.2.4 Pipe Wall Stress Analysis

An ASME B31.1 pipe wall stress analysis was performed to determine the required wall thickness of the PFBC outlet pipe. Several different pipe materials were considered, with their allowable stress values. The resulting required minimum wall thicknesses, t_m , for a 60-inch pipe at 200 psig and 650°F or less were:

- For A106, Grade A Seamless: $t_m = 0.496$ inch
- For A106, Grade B Seamless: $t_m = 0.397$ inch

For a 48-inch pipe, the same pressure and temperature:

- For A106, Grade A Seamless: $t_m = 0.397$ inch
- For A106, Grade B Seamless: $t_m = 0.318$ inch

6.2.5 Pipe Wall Stress Analysis Results

The following results were obtained from the pipe wall stress analysis we performed for the reference PFBC hot gas piping:

- We recommend A106, Grade B, carbon steel for this application.
- If A106, Grade B, pipe is used, the wall thickness of the 60-inch pipe should be 0.50 inch. The 48-inch pipe wall should be 0.375 inch (standard wall).

6.2.6 Special Clean Hot Gas Pipe Mechanical Design Requirements

Special design requirements for the hot pipe result from the cast-in-place refractory insulation:

- The maximum pipe section length for cast insulation is 10 feet.
- Insulated pipe sections can be welded together with proper joint design.
- Flanged joints every 20 to 30 feet provide access for inspection and repair.
- To permit axial thermal expansion the pipe liner must be free of insulation except at one support location. Slip joints must also be provided between liner sections.
- A gap must be provided between the liner and the insulation to permit radial thermal expansion. Flexible standoffs support the liner from the insulation.
- External insulation or lagging must be used with care. A thin layer (1/4 inch) of low-conductivity material could be used to protect personnel against contact burns.

6.2.7 Results

- The pipe sizes (figures 6-2 and 6-3) are reasonable for power plant piping. The velocities for most of the runs are in the 5,000 to 8,000 fpm range, which is reasonable for this application.
- The pressure drops through the pipes are not excessive. A total of 18.5 inches H₂O in the cold pipe and 11 inches H₂O in the hot pipe are acceptable.

6.3 VALVE DESIGN

6.3.1 Startup and Shutdown Sequence

The requirements for valves in the turbine/compressor system of the PFBC plant are, in part, determined by the startup and shutdown sequences and operating modes required by the various plant systems. The operating sequence developed for the project has been used to establish the arrangement and design requirements for the turbine/compressor valves of the reference PFBC plant. The valve arrangement is shown in Figure 6-1. Valve requirements should be considered preliminary pending a dynamic flow analysis of the system during the next design phase.

6.3.2 PFBC System Bypass Valve

This valve supplies compressor discharge flow to the turbine inlet during turbine and PFBC bed startup. During turbine startup, this valve will be wide open to provide 100 percent of compressor flow to the turbine. During PFBC bed startup and shutdown, this valve will throttle the bed bypass flow. The requirements are:

Type:	Butterfly
Size:	24 inch
Connections:	Flanged
Flow:	211 lb/sec during turbine startup 158-179 lb/sec during bed startup <0.5 lb/sec for power operation
Temperature:	700°F (max), 20°F (min)
Pressure:	200 psig
Shutoff:	1 percent of design flow at design pressure.
Throttle Range:	20 to 80 percent of rated flow

6.3.3 PFBC System Inlet Valve

This valve isolates the PFBC System from the compressor discharge. The requirements are:

Type:	Butterfly
Size:	36 inch
Connections:	Flanged
Flow:	211 lb/sec during power operation 32 to 53 lb/sec during bed startup 2.1 lb/sec (max) during turbine startup
Temperature:	700°F (max), 20°F (min)
Pressure:	200 psig
Shutoff:	1 percent of design flow at design pressure
Throttle Range:	20 to 30 percent of design flow

6.3.4 PFBC System Outlet Valve

This valve isolates the turbine inlet from the PFBC system during PFBC bed startup and plant shutdown. During power operation this valve is wide open. Throttling may be required during the changeover from startup to power operation and during shutdown. The requirements are:

Type:	Butterfly
Size:	44-inch id in a 60-inch flange
Connections:	Flanged (60 in)
Flow:	250 lb/sec during power operation 2.5 lb/sec (max) during bed startup
Temperature:	1600°F (max), 20°F (min)
Pressure:	200 psig
Shutoff:	1 percent of design flow at design pressure
Throttle Range:	20 to 80 percent of design flow

6.3.5 PFBC System Vent Valve

This valve vents the PFBC System bed. Since all the requirements for this valve are related to PFBC system operations, this valve is not a part of this report, nor are other valves that are part of the PFBC system.

6.3.6 PFBC System Purge Valve

This valve controls purge and pressurization air to the PFBC system during startup.

Type:	Globe
Size:	8 in.
Connections:	Flanged
Flow:	53 lb/sec 150 psi pressure differential
Temperature:	700°F (max), 20°F (min)
Pressure:	200 psig
Shutoff:	1 percent of design flow at design pressure
Open/Close Pressure Differential:	200 psig
Throttle Range:	50 - 100 percent of design flow

Alternative: A 12-inch butterfly valve located upstream of a multiple (6 - 10 stage) orifice flow restrictor. The valve/restrictor combination would drop the purge flow pressure from 160 psia to 14.3 psia in multiple stages so that no one stage would have a pressure ratio less than 0.53 over the entire operating range and sonic flow would be avoided across all stages.

6.4 PIPING LAYOUT

Figures 6-2 and 6-3 show the piping out of the compressor, through the turbine startup combustor, and back to the turbine. They also include the PFBC supply and return connections with isolation valves. This arrangement reflects the results of the preceding analyses together with several additional design requirements:

- Interface with turbine casing penetrations
- Isolate the turbine casing from pipe forces due to thermal expansion
- Minimize the air volume in the startup combustor circuit
- Minimize cost consistent with reasonable pressure drop

- Satisfy startup and shutdown airflow and turbine control requirements
- Interface with PFBC piping
- Incorporate manufacturing requirements for the insulated pipe (flanged joints, maximum assembly lengths)

In arranging the piping and components, the hot return pipe was considered first. The pressure balanced expansion elbow was placed as close as possible to the casing flange and connected by a reducing section to mate the pipe insulation liner diameter with the casing liner diameter. The expansion joint provides an anchor point for the main run of hot pipe to isolate its motion from the casing.

On top of the expansion joint is a flanged tee that connects to the PFBC system outlet valve and to the discharge of the turbine startup combustor. Above the system outlet valve the PFBC outlet pipe expands to 60 inches in diameter. The turbine startup combustor line contains a throttle valve and an expansion joint. The combined lengths of the components in this crossover pipe determined the location of the compressor discharge pipe.

The compressor discharge pipe starts with a reducer to connect the 24-inch casing flange with the 30-inch main pipe diameter. A 22-foot pipe section connects the reducer with a pressure balanced expansion elbow. The outlet of the elbow connects to an 8-foot long vertical pipe section, flanged to a tee leading to the turbine startup combustor. Straight through the tee is the PFBC system inlet valve. Bypassing the PFBC system inlet valve is the PFBC system purge valve.

The compressor discharge pipe is insulated externally, since the air temperature is approximately 670°F and carbon steel pipe performs well at this temperature. The PFBC outlet pipe, containing 1600°F clean hot gas, is lined with a ceramic insulation and stainless steel liner. The hot expansion joint contains a stainless steel liner and fibrous insulation.

7. CONCLUSIONS

The result of this successful study is a versatile conceptual design capable of meeting PFBC requirements.

7.1 TURBINE AND PLANT PERFORMANCE

Performance calculations were determined by varying the following parameters:

- Inlet temperature - Analysis shows that the necessary modifications to the W251B12 limit the lower inlet to approximately 55°F. The net power output expected, at a 59°F inlet temperature, is 20,250 kW.
- Pressure drop in the system - As the PFBC system pressure drop increases, power output level decreases. A detail design objective should be to design the bed, piping, valves, and turbine casing to minimize system pressure drop.
- Composition of vitiated air - As expected, the composition of the fuel affects the power output. A fuel with products of combustion higher in moisture content and less carbon dioxide produced more power.
- Vitiated air temperature into the turbine - From a net power perspective, the system should operate at the highest temperature it can accommodate, but is limited by the operating temperature of the ceramic filters.

7.2 BLADING AND COOLING ARRANGEMENTS

7.2.1 Compressor

For the selected PFBC application, removing the first two stages of compressor blading, reshaping the inlet scroll, modifying the blade design and repositioning the IGV is a feasible design approach to obtaining the reduced (from the standard W251B12 19-stage compressor) inlet mass flow. Analysis results show a reduction in mass flow from 375 lb/sec to 229 lb/sec, a reduction in pressure ratio from 15.5 to 11.6 and a revised compressor map and an increased sensitivity to inlet air temperatures, especially at the lower part of the range. If a compressor was required for a PFBC application with mass flow similar to the standard W251B12, these modifications would not be necessary.

7.2.2 Turbine

Analysis of the reduced flow PFBC turbine showed that the existing vane setting angle is approximately correct. The compressible flow function for the reduced flow PFBC is 77.8 compared to 81.1 for the baseline W251B12. Incidence angles were acceptable following current Westinghouse design practices. The match of incidence angles and turbine efficiency can be modestly improved by closing the row 1 vane by approximately 1 degree but is not recommended due to the additional manufacturing cost involved.

7.3 CASING ARRANGEMENT WITH OFF-ENGINE COMBUSTOR

Several alternative casing concepts were developed and analyzed. The concentric inlet/outlet design was eliminated because of the very large casing penetrations required and the potential for the inlet liner to creep or buckle at elevated temperatures when subjected to external pressure. The single inlet/outlet design with the piping located on the lower right-hand side of the casing was selected as the conceptual design choice because it used the existing bedplate design, allowed floor space on one side of the turbine for laydown, and kept the upper half of the turbine free of major piping connections for ease of maintenance.

The total pressure loss from the compressor diffuser exit to the pipe leading to the fluidized-bed combustor is approximately 2.9 psi, which compares favorably to the 2.2 psi loss to the combustor inlet for the unmodified gas turbine without the fluidized-bed combustor.

The separate pipe casing penetrations were located to minimize interference with existing gas turbine piping and support structures. An area reinforcement analysis, in accordance with the ASME Boiler and Pressure Vessel Code, was performed, and the inlet and outlet was located and sized to satisfy the rules.

7.4 TURBINE SYSTEM

7.4.1 Control System

In order to develop a control system conceptual design, normal operation and emergency shutdown sequences were developed. The current WDPF control system offered by Westinghouse for the W251F 12 is acceptable for the PFBC application.

7.4.2 Fuel System

The combustion turbine startup combustor must operate on either fuel oil or natural gas. Westinghouse has provided skid-mounted systems for each of these fuels and supplies them for conventional combustion turbine applications. Modified versions of these fuel skids will be required for the PFBC plant.

7.4.3 Inlet and Exhaust System

The inlet and exhaust system for this PFBC application will be slightly different from the typical Westinghouse ECONOPAC offering. The modified 17-stage compressor requires inlet air temperature control so a combination heating and cooling system must be installed in the intake system. The exhaust system will contain a heat recovery steam generator.

7.5 OFF-ENGINE STARTUP COMBUSTOR

Combustion performance calculations were performed and a conceptual design for an off-engine startup combustor was developed along with a detailed bill of materials. Many of the current combustor components can be used in this application, including the combustor basket, fuel nozzles, igniters, and steam or water injection system (if necessary).

7.6 INTERCONNECTING PIPING

Definition of the plant cycle arrangement and the fluid state points of that arrangement were developed in order to conceptually design the piping system. The pipe sizes defined are reasonable for power plant piping. The velocities for most of the runs are in the 5,000 to 8,000 fpm range, which is reasonable for this application. The pressure drops through the pipes are not excessive and are acceptable. A total of 18.5 inches H_2O in the cold pipe and 11 inches H_2O in the hot pipe. The cold pipe can be made of standard piping materials. The two sizes of clean hot gas pipes will require a design that includes an internal stainless steel liner, castable refractory inner-liner, and seamless carbon steel pipe. Piping and insulation material recommendations are based on temperature profile and pipe wall stress analyses and are acceptable based on prior successful operating experience and ASME B31-1. Flexible expansion joints are shown in both the compressed air and clean hot gas piping systems and turbine/compressor valve characteristics were developed and are specified.

APPENDIX A
DESIGN EVOLUTION

ILLUSTRATIONS

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APPENDIX A - DESIGN EVOLUTION

Westinghouse considered three concepts for the arrangement of inlet and outlet piping. Concept 1, (figure A-1), concentric/eccentric piping, was eliminated because of the very large casing penetrations required and the potential for the inlet liner to creep or buckle at elevated temperatures when subjected to external pressure. Concept 2, (figure A-2), with separate inlet/outlet pipes, avoided these problems but required off-engine startup combustors. Concept 3 (figure A-3) was similar to concept 2 but featured on-engine startup combustors, which eliminated the separate, independently pressurized turbine startup system and could operate in a topping mode.

Each of the three concepts was compatible with one or two sets of inlet and outlet pipes. System performance and mechanical design considerations favored dual inlet designs (see figure A-4). The baseline W251B12 uses a bedplate design that standardizes bleed air, lubrication, and fuel piping and facilitates shipping and installation. Use of the existing bedplate design has important cost advantages but restricts the location and size of interconnecting piping. We also wanted to keep the floor space on one side of the turbine clear and to keep the upper half of the turbine free of major piping connections for ease of maintenance. These factors favor one inlet and outlet pipe located on the lower right-hand side of the casing (figures A-5 and A-6).

Customer input during and after the progress review indicated little interest in a topping capability and the simplicity offered by on-engine combustors. To eliminate the additional pressure drop resulting from the presence of on-engine combustors during PFBC operation, an off-engine turbine startup combustor located between the inlet and outlet pipes (above the expansion joints) was selected. The resulting casing design is shown on figure A-7 and is compatible with the plant arrangement shown on figures 6-2 and 6-3.

One feature common to all these designs is the use of 670°F compressor outlet air to cool the manifold and casing walls that would otherwise be overheated by the 1600°F inlet air from the PFBC. With appropriate radiation shielding, each design can keep critical structural components acceptably cool to preclude creep, buckling, and corrosion problems. This cooling air can also be used to cool the rotor and support combustion with on-engine combustors. A patent disclosure has been filed covering these cooling concepts.

- SIMILAR TO ORIGINAL 1991 CONCEPT (CONCENTRIC INLET/OUTLET)
- TWO LARGE PENETRATIONS REQUIRED, ONE IN TOP COVER AND ONE IN BOTTOM
- MAXIMUM DIAMETER OF THE OUTLET PIPE LIMITED BY THE LENGTH OF THE COMPRESSOR/COMBUSTOR CASING
- POTENTIAL CREEP BUCKLING PROBLEM OF 1600°F PIPE WITH NET EXTERNAL ΔP OF 20 PSI

Figure A-1. Concept 1 - Concentric/Eccentric Inlet and Outlet Piping

- TWO SETS OF INLET AND OUTLET PIPES REQUIRED TO ACCOMMODATE THE AIRFLOW WITH ACCEPTABLE LOSSES
- FLEXPLATE CENTERLINE SUPPORT OF TRANSITION DUCTS
- MAXIMUM DIAMETER OF THE OUTLET PIPE LIMITED BY THE LENGTH OF THE COMPRESSOR/COMBUSTOR REMAINING AFTER INLET PENETRATION
- PRESERVES STRENGTH OF CASING

Figure A-2. Separate Inlet/Outlet Piping with Off-Engine Startup Combustors

- SIMILAR TO CONCEPT 2
- STANDARD (251) ON-ENGINE COMBUSTORS OR EXTERNAL COMBUSTORS CAN BE USED FOR STARTUP
 - ATTRACTIVE FOR PFBC STARTUP TIMES ON THE ORDER OF 1-2 DAYS
 - EXISTING COMBUSTORS WILL ACCOMMODATE REDUCED AIRFLOW FROM PFBC WITH ACCEPTABLE LOSSES
- ADAPTABLE TO TOPPING CYCLE USING COMPRESSOR DISCHARGE AIR FOR PRIMARY COMBUSTION

Figure A-3. Separate Inlet and Outlet Piping with On-Engine Startup Combustors

ADVANTAGES:

- Lower inlet velocities
- Better flow distribution
- Smaller inlet pipes
- More room for flanges
- No increase in casing width
- Simple cylindrical plenum sections

DISADVANTAGES:

- Two inlet pipes - top and bottom
- Piping arrangement requires more space
- Bypass cooling needed for plenum

Figure A-4. Dual Inlet Concept (Backup)

ADVANTAGES:

- One inlet and outlet on lower side (not top or bottom) improves access for maintenance
- Better external piping arrangement

DISADVANTAGES:

- Larger pipe
- Limited room for flange
- Higher inlet velocity
- Larger casing required (~ 144" wide)
- Plenum design to resist ΔP at high temperature

Figure A-5. Single Inlet Concept (Preferred)

Figure A-6. Section through Inlets

APPENDIX B

PIPING AND INSULATION SIZING CALCULATIONS

TABLES

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Table B-1. Cold Pipe Pressure Loss Analysis

15-Feb

Rev. 1

SYSTEM IDENTIFICATION: PFBC COLD PIPE

The spreadsheet is using the following parameters:

Velocity Limits:

Duct friction factor: ASHRAE
Duct material: STEEL
Insulation:
Liner: NONE
Air Density: 0.39182
Convergence criterion: 0.001

Main duct: 12,000
Branch duct: 12,000

Design Flow = 211 lb/sec
Design Flow = 12,660 lb/min
Dens.@657 & 162 = 0.391829 lb/cu ft
Design Flow 1 = 32,309.97 cfm
RFO_C = 1,752.203

ITEM BR NO. CH	DESCRIPTION	CFM	LOSS COEF.	VEL (fpm)	DELTA P (in. H ₂ O)	P STATIC (psia)
	Start model at compressor outlet.					162.30
1	Flange, 28"	32,309.97	0.00	11,198	0.00	162.30
2	Transition, 28-30"	32,309.97	0.16	7,044	8.60	161.99
3	Duct, 30"	32,309.97	0.09	7,044	1.50	161.93
4	Elbow, 30"	32,309.97	0.21	7,044	3.47	161.81
5	Duct, Round	32,309.97	0.05	7,044	0.75	161.78
6	Branch 1 Flow	1				161.78
7	Main Flow Now	32,309.97				161.78
8	Takeoff (Ac=As)	32,309.97	0	7,044	0.00	161.78
9	Duct, 30"	32,309.97	0.00	7,044	0.00	161.78
10	Valve, Butterfly	32,309.97	0.19	7,044	3.07	161.67
11	Transition, 30 x 36	32,309.97	0.24	4,571	0.74	161.64
12						
13	Total pressure drop:				18.14	0.66

CFM = volumetric flow rate, cu. ft/min
LOSS COEF. = loss coefficient, geometry dependent
VEL = local velocity, ft/min
DELTA P = pressure drop due to fitting or friction, inches of water
P STATIC = local static pressure downstream of fitting, lb/sq. in absolute

Table B-2. Hot Pipe Pressure Loss Analysis

15-Feb

Rev. 1

SYSTEM IDENTIFICATION: PFBC HOT PIPE

The spreadsheet is using the following parameters:

Velocity Limits:

Duct friction factor: ASHRAE
Duct material: STEEL
Insulation:
Liner: St. Std. STAINLESS
Air Density: 0.18360
Convergence criterion: N.A.

Main duct: 12,000
Branch duct: 12,000

Design Flow = 249 lb/sec
Design Flow = 14,490 lb/min
Dens. @1600 & 140 = 0.183609 lb/ft³
Design Flow 1 = 81,368 cfm

ITEM BR NO. CH	DESCRIPTION	CFM	LOSS COEF.	VEL (fpm)	DELTA P (in. H ₂ O)	P STATIC (psia)
	Start model at the transition into the PFBC hot isolation valve.					140.50
1	Duct, 44"	81,368	0.00	7,706	0.00	140.50
2	Transit ⁿ .on, 44-36"	81,368	0.10	11,511	0.91	140.47
3	Valve, 36"	81,368	0.19	11,511	3.84	140.33
4	Duct, 36" ID	81,368	0.02	11,511	0.48	140.31
5	Elbow, 36"	81,368	0.22	11,511	4.44	140.15
6	Duct, 36" ID	81,368	0.02	11,511	0.36	140.14
7	Transition, 36-32"	81,368	0.05	14,569	1.01	140.10
8						
9	Total Pressure Drop:				11.05	0.40

CFM = volumetric flow rate, cu. ft/min
LOSS COEF. = loss coefficient, geometry dependent
VEL = local velocity, ft/min
DELTA P = pressure drop due to fitting or friction, inches of water
P STATIC = local static pressure downstream of fitting, lb/sq. in absolute

Table B-3. 48-Inch Hot Pipe Wall Temperature Calculations

THERMAL CONDUCTIVITY CALCULATION
PFBC 48-INCH HOT PIPE w/ 5.5-INCH CERAMIC INSULATION

Inner Film	Liner	Ceramic Insulation	Pipe Wall	Outer Insulation	Outer Film
ID = 36	ID = 36.0	ID = 36.1	ID = 47.0	ID = 48.0	ID = 48.0
Thick = 0	Thick = 0.06	Thick = 5.48	Thick = 0.5	Thick = 0	Thick = 0
OD = 36	OD = 36.1	OD = 47.0	OD = 48.0	OD = 48.0	OD = 48.0
C = 0.0053	k = 13.4	k = 0.11	k = 25.9	k = 0.03	
Vel. = 11,511	OD/ID = 1.00	OD/ID = 1.30	OD/ID = 1.02	OD/ID = 1	
Rho = 0.1836	lnD/D = 0.00	lnD/D = 0.26	lnD/D = 0.02	lnD/D = 0	
G = 126,005	1/k = 0.07	1/k = 8.63	1/k = 0.03	1/k = 27.7	
G ^{0.8} = 12092	1/k*ln = 0.00	1/k*ln = 2.28	1/k*ln = 0.00	1/k*ln = 0	
D ^{0.2} = 1.2457					delta 146
h = 51.544					h = 2.03
A/ft = 9.4247					A/ft = 12.5
Ti = 1,600	Ti = 1,592	Ti = 1,592	Ti = 227	Ti = 266	Ti = 227
Q = 3,479		Q = 3,749.2	Q = 3,749.2		Q = 3,749 dT = 147
To = 1,592.2	To = 1,592	To = 227 U = 0.1541	To = 226	To = 226	To = 80

Table B-4. 60-Inch Hot Pipe Wall Temperature Calculations

THERMAL CONDUCTIVITY CALCULATION
PFBC 60-INCH HOT PIPE w/ 7.5-INCH CERAMIC INSULATION

Inner Film	Liner	Ceramic Insulation	Pipe Wall	Outer Insulation	Outer Film
ID = 44	ID = 44.0	ID = 44.1	ID = 59.0	ID = 60.0	ID = 60.0
Thick = 0	Thick = 0.06	Thick = 7.48	Thick = 0.5	Thick = 0	Thick = 0
OD = 44	OD = 44.1	OD = 59.0	OD = 60.0	OD = 60.0	OD = 60.0
C = 0.0053	k = 13.4	k = 0.11	k = 25.9	k = 0.03	
Vel. = 7,706	OD/ID = 1.00	OD/ID = 1.33	OD/ID = 1.01	OD/ID = 1	
Rho = 0.1836	lnD/D = 0.00	lnD/D = 0.29	lnD/D = 0.01	lnD/D = 0	
G = 84,889	1/k = 0.07	1/k = 8.63	1/k = 0.03	1/k = 27.7	
$G^{0.8} = 8771.6$	$1/k \cdot \ln = 0.00$	$1/k \cdot \ln = 2.52$	$1/k \cdot \ln = 0.00$	$1/k \cdot \ln = 0$	
$D^{0.2} = 1.2967$					delta 117
h = 35.918					h = 1.88
A/ft = 11.519					A/ft = 15.7
Ti = 1,600	Ti = 1,591	Ti = 1,591	Ti = 197	Ti = 196	Ti = 197
Q = 3,475		Q = 3,475.4	Q = 3,475.4		Q = 3,475 dT = 117
To = 1,591.6	To = 1,591	To = 197 U = 0.1429	To = 196	To = 196	To = 80

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